STOCHASTIC DISTURBANCES AND DYNAMICS OF THERMAL PROCESSES
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STOCHASTIC DISTURBANCES AND DYNAMICS OF THERMAL PROCESSES

with application to municipal solid waste combustion

PROEFSCHRIFT

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Summary

Thermal processes like combustion are important in helping to fulfil man’s energy need. In the total energy supply the role of sustainable energy is becoming more and more important. Waste incineration, biomass combustion and co-firing of biomass in power plants are examples of the implementation of sustainable energy in The Netherlands. However, compared with fossil fuels, waste and biomass have the disadvantage of a less consistent fuel quality. Stochastic disturbances in the fuel quality (the moisture content of the fuel, the composition of the combustible part and the density of the fuel) and the external requirements (low emissions, high energy efficiency and high availability) make the control of these combustion processes all the more important. At the start of this research, no tools were available for studying the influence of stochastic disturbances in the fuel on the process. For that reason a research programme was started into the stochastic disturbances and dynamics of thermal processes.

In this thesis, fundamental aspects of the combustion process on a grate are discussed and applied to Municipal Solid Waste Combustion (MSWC). New techniques and models are presented, that can be used to unravel the complex process of combustion of solid fuels on a grate. The presented tools are very valuable for the design of new control concepts and can contribute to the design of more efficient and reliable MSWC plants in the future.

A new method for the on-line determination of the calorific value of solid fuels is presented. Integration of a new hardware sensor, a mathematical model and physical knowledge of the conversion processes has led to a new calorific value sensor. Experiments with MSWC plants show that the sensor is capable of continuously monitoring the calorific value of the waste being converted. Besides that, the sensor generates other important process information, like for example the density and the water content of the waste. In this study, the data provided by the sensor are assessed as to their practical reliability and accuracy. The new developed method is compared with traditional methods and it is shown that the new method has several advantages. Furthermore, attention is given to the application of the on-line calorific value sensor.

A dynamic model is a very valuable tool for studying the dynamics of a thermal process. As there are no validated dynamic models available in literature for MSWC, a general dynamic model of thermal processes has been developed. Energy and mass
balances for a fuel consisting of one solid material (combustible) as well as balances for a fuel consisting of two solid materials (moisture and combustible) are derived. This general model is applied to MSWC, providing a completely new dynamic model description of the MSWC process.

As no information is available in literature about the validation of a process like MSWC, a new validation method is also presented. This method is applied to data obtained from the MSWC plant NV Huisvuilcentrale N-H in Alkmaar, The Netherlands. The data were gathered at two different operation points. The first case deals with a regular primary air temperature (120 °C), while the second case deals with a lower primary air temperature (70 °C). It is shown that the developed model describes the observed phenomena in the MSWC plant in a satisfying manner. Furthermore, it is shown that the reaction rate can be described by a classical mass transfer relation: in other words the reaction rate is mass transfer limited.

With the disturbance data provided by the on-line calorific value sensor and a validated dynamic model of the MSWC process, the theory from stochastic processes is applied to MSWC. This new application of stochastic concepts in the field of MSWC yields new results. It is shown that the irregularity in the calorific value waste can be described by an ergodic process. Furthermore, it is proven that the probability distribution of the calorific value, in kJ mol⁻¹, is Gaussian for the Dutch MSWC plants NV Huisvuilcentrale N-H and GAVI Essent Milieu Wijster. An analysis of data from two other Dutch MSWC plants (NV Afvalverbranding Zuid-Nederland in Moerdijk and Twence Afvalverwerking in Hengelo) shows that the probability distribution function of the calorific value, in kJ mol⁻¹, is not exactly Gaussian. These deviations from Gaussian behaviour are probably caused by the mixing process in the bunker. Regarding the spectral density of the calorific value it is concluded that due to the complexity of the process it is not possible to derive simple analytical relations for the spectral density. Furthermore, it is proven that the non-linear dynamic model describing the main processes in the furnace of a MSWC, can be linearized in a working point without losing essential information. In practice, the process will comprise several working points, which implies that the process can be described by a multiple linear system. This means that a complete system description will include a set of linear models, whereby the parameters of the linear models are dependent on the actual working point.

New linear transfer functions for thermal processes are derived from the linear model, which are used to carry out a sensitivity analysis in order to investigate the influence of process and design parameters on the standard deviation of the main output variables. It is concluded that decreasing the calorific value, and thus changing the working point, has a large negative impact on the standard deviation of the gas phase temperature. This implies that combustion of waste with lower-than-normal calorific values is more difficult to control than waste with higher calorific values.

The validation experiments performed at the MSWC plant NV Huisvuilcentrale N-H showed that the process behaviour at lower primary air temperatures (70 °C) is completely different than it is at higher temperatures (120 °C). This newly observed phenomenon has been studied in detail in pot furnace experiments. Extensive expe-
riment were carried out in order to study the influence of preheating of the primary air on the combustion behaviour of solid fuels on a grate. New results from these laboratory experiments with primary air preheating are presented.

The present theory for the combustion of solid fuels is extended in order to explain the observed phenomena with preheated air. The combustion process without preheated air can be described with the classical theory proposed by Rogers. The extension of this theory with the influence of preheated air provides an explanation for all observed phenomena. The main point of the theory is that preheating of the primary air acts as a catalyst for the ignition on a grate rather than merely drying the waste. A combination of a water-cooled grate with efficient primary air preheating can result in smaller and more compact grate systems.

An important outcome of this study is that pot furnace experiments have only a limited value in studying grate furnace combustion. The translation from the plug flow behaviour into a batch type reactor is limited by the fact that in the real plant the grate movements have a large impact on the combustion behaviour. This effect is not simulated in the pot furnace experiments. For that reason the translation of results of pot furnace experiments into practical solutions should be made with care.
Samenvatting

Thermische processen zoals verbranding zijn belangrijk in het vervullen van de menselijke energie behoefte. In de totale energievoorziening wordt het aandeel duurzame energie steeds belangrijker. Afvalverbranding, biomassa verbranding en meestal ook van biomassa in elektriciteitscentrales zijn voorbeelden van de implementatie van duurzame energie in Nederland. Echter, vergeleken met fossiele brandstoffen, hebben afval en biomassa het nadeel dat de brandstofkwaliteit minder constant is. Tengevolge van stochastische verstoringen in de brandstofkwaliteit (vochtgehalte van de brandstof, de samenstelling van het brandbare deel van de brandstof en de dichtheid van de brandstof) en de opgelegde externe eisen (lage emissies, hoge beschikbaarheid en hoog energetisch rendement) wordt het beheersen van zulke processen belangrijker. Bij de start van dit onderzoek waren voor thermische processen, en afvalverbranding in het bijzonder, geen gereedschappen beschikbaar om de invloed van brandstoffen variaties op het proces te onderzoeken. Om die reden is destijds een onderzoeksprogramma gestart naar de dynamica van thermische processen alsmede naar de invloed van stochastische verstoringen op thermische processen. De ontwikkelde technieken zijn toegespitst op afvalverbranding.

In dit proefschrift worden fundamentele aspecten van de verbranding van vaste brandstoffen op een rooster besproken. Nieuwe technieken worden gepresenteerd, waarmee het mogelijk is het complexe proces van verbranding op een rooster beter te begrijpen. De gepresenteerde methodieken zijn erg waardevol voor het ontwerpen van nieuwe regelconcepten en kunnen verder bijdragen aan het ontwerp van meer efficiënte afvalverbrandingsinstallaties in de toekomst.

Om de stookwaarde van afval te kunnen bepalen is een nieuwe methode ontwikkeld, waarmee de stookwaarde continu kan worden bepaald. Integratie van een nieuwe hardware sensor, een wiskundig model en fysische proceskennis heeft geleid tot een nieuwe stookwaarde sensor. Toepassing van deze sensor op het afvalverbrandingsproces laat zien dat de sensor in staat is om continu de stookwaarde van het verbrandende afval te bepalen. Daarnaast is de sensor ook in staat andere belangrijke procesdata te genereren, zoals bijvoorbeeld de dichtheid van het afval en de vochtigheid van het afval. In de praktijk is aangetoond dat de sensor betrouwbaar en nauwkeurige resultaten oplevert. De nieuw ontwikkelde methode is vergeleken met traditionele methoden en het is aangetoond dat de nieuwe methode verschillende voordelen biedt.

Verder is aandacht besteed aan de mogelijke toepassing van de stookwaarde sensor.
Een dynamisch procesmodel is een goed hulpmiddel bij het onderzoeken van het dynamisch gedrag van een thermisch proces. Omdat in de literatuur geen gevalideerde dynamische procesmodellen voor afvalverbranding bekend waren, is tijdens dit onderzoek een algemeen dynamisch model voor thermische processen ontwikkeld. Energie- en massabalen voor een brandstof die uit één of uit twee componenten bestaat zijn afgeleid. Dit algemene model is toegepast op afvalverbranding hetgeen heeft geresulteerd in een complete nieuwe beschrijving van de dynamica van het afvalverbrandingsproces.

Omdat in de literatuur geen informatie beschikbaar was voor de validatie van complexe dynamische processen zoals afvalverbranding, is binnen het onderzoek een nieuwe methode toegepast. Deze methode is toegepast op data verkregen bij de NV Huisvuilcentrale N-H in Alkmaar. De verzamelde data heeft betrekking op twee verschillende werkpunten. Het eerste werk punt heeft betrekking op een normale primaire luchttemperatuur van 120 °C, terwijl het tweede werk punt betrekking heeft op een lagere primaire luchtvoorverwarmingstemperatuur (120 °C). Het is aangetoond dat het ontwikkelde model in voldoende mate de geobserveerde verschijnselen kan verklaren. Verder is aangetoond dat de reactiesnelheid kan worden beschreven met een klassieke stofoverdrachtsvergelijking.

De theorie van stochastische processen is toegepast op afvalverbranding. Hiervoor is de door de on-line stookwaarde sensor verzamelde data en het gevalideerde procesmodel gebruikt. Deze nieuwe toepassing van stochastische concepten binnen afvalverbranding heeft nieuwe resultaten opgeleverd. Het is aangetoond dat de verstoringen in de stookwaarde van huishoudelijk afval ergodisch van aard zijn. Verder is bewezen dat de waarschijnlijkheidscoditionalverdeling van de stookwaarde in $kJ mol^{-1}$ Gaus- sisch is bij de installaties NV Huisvuilcentrale N-H en GAVI Essent Milieu Wijster in Wijster. Analyse van data afkomstig uit twee andere installaties (Twence Afvalver- werking in Hengelo en NV Afvalverbranding Zuid-Nederland in Moerdijk) toont aan dat de waarschijnlijkheidscoditionalverdeling van de stookwaarde (in $kJ mol^{-1}$) in hun installaties niet exact Gaussisch is. De afwijkingen van Gaussisch gedrag wordt waarschijnlijk veroorzaakt door het mengproces in de bunker. Om de stochastische analyse te complementeren is ook de spectrale dichtheid van de stookwaarde bepaald. Uit deze resultaten is geconcludeerd dat het vanwege de complexiteit van het proces niet mogelijk is om eenvoudige relaties voor de spectrale dichtheid af te leiden.

Verder is aangetoond dat het niet-lineaire dynamische model, welke de hoofdprocessen in de vuurhaard van een afvalverbrandingsinstallatie beschrijft, kan worden gelineariseerd in een werk punt zonder dat essentiële informatie verloren gaat. In de praktijk zal het proces een aantal werkpunten omvatten, hetgeen betekent dat het proces beschreven kan worden met een meervoudig lineair systeem. Dit houdt in dat een complete systeembeschrijving een set lineaire modellen zal omvatten, waarbij de parameters die in modellen voorkomen zullen afhangen van de actuele werk punten. Nieuwe lineaire overdrachtsfuncties voor thermische processen zijn afgeleid uit het lineaire model, welke zijn gebruikt om een gevoeligheidsanalyse uit te voeren naar de invloed van ontwerp- en procesparameters op de standaarddeviatie van de
belangrijkste uitgangsvariabelen. Hieruit is geconcludeerd dat een verlaging van de
stookwaarde, dus verandering van het werk punt, een sterke negatieve invloed heeft
op de standaarddeviatie van de gasfasetemperatuur. Dit impliceert dat verbranding
van afval met een lagere stookwaarde moeilijker is dan verbranding van afval met een
hogere stookwaarde.

De bij de NV Huisvuilkentrale N-H uitgevoerde validatie-experimenten laten zien
dat het procegsedrag bij lagere primaire luchttemperaturen (70 °C) verschilt van
het gedrag bij hogere (120 °C) temperaturen. Dit nieuwe verschijnsel is uitvoerig
besneden in zogenaamde potoven experimenten. Uitgebreide experimenten zijn uit-
gevoerd om de invloed van primaire luchtvoorverwarming op het verbrandingsgedrag
van vaste brandstoffen op een rooster in kaart te brengen. Nieuwe resultaten van de
laboratorium experimenten zijn gepresenteerd.

De huidige theorie voor verbranding van vaste brandstoffen op een rooster is uitge-
breid ten einde de waargenomen verschijnselen met luchtvoorverwarming te kunnen
verklaren. Het verbrandingsproces zonder luchtvoorverwarming kan worden beschre-
ven met de klassieke theorie van Rogers. De uitbreiding van deze theorie met de
invloed van luchtvoorverwarming levert een theorie die in staat is alle waargenomen
verschijnselen te verklaren. De kern van deze theorie is dat de primaire luchtvoorver-
warming katalyserend werkt op het ontstekingsproces, in plaats van alleen een dro-
gend effect. De combinatie van een watergekoeld rooster met een efficiënte inzet van
primaire luchtvoorverwarming kan leiden tot kleinere en meer compacte roostersyste-
men.

Een belangrijk resultaat van de potoven experimenten is dat deze experimenten naar
een beperkte waarde hebben bij het verklaren van verschijnselen in de praktijk. De
vertaling van het propstrooggedrag naar een pot reactor wordt beperkt doordat in
praktijkinstallaties roosterbewegingen een grote invloed op het verbrandingsproces
hebben. Dit effect wordt niet gesimuleerd in potoven experimenten. Dit is de voor-
naamste reden waarom de vertaling van resultaten uit experimenten naar de praktijk
voorzichtig moet worden uitgevoerd.
Voorwoord

Het is januari 1992 en ik start mijn loopbaan bij TNO in Apeldoorn. Mijn voornaamste opdracht bestond uit het ontwikkelen van een dynamisch model van een afvalverbrandingsinstallatie. Inmiddels is het medio 2003 en ik ben nog steeds betrokken bij de dynamica van het afvalverbrandingsproces, hetgeen geresulteerd heeft in onderliggend proefschrift. Afvalverbranding is voor mij de afgelopen jaren een boeiend onderwerp geweest waarin ook in de toekomst nog vele onderzeks vrienden blijven bestaan.

Als ik terugkijk op de afgelopen periode die dit onderzoek heeft geduurd zijn er twee momenten van grote invloed geweest op het uiteindelijke resultaat. Ten eerste was dat de ontwikkeling van de on-line stookwaarde sensor die het mogelijk maakte on-line data te verzamelen, waarop de theorie van de stochastiek kon worden toegepast. Ten tweede is dat de toepassing van systeemidentificatietechnieken in de praktijk, waar door de dynamica van het proces beschikbaar kwam en het mogelijk is geworden om de dynamische modellen te valideren.

Een promotieonderzoek kan alleen worden uitgevoerd als aan een aantal voorwaarden wordt voldaan.

Geen promotie zonder een goed onderwerp. In de eerste jaren van het onderzoek naar afvalverbranding bestond het werk bij TNO globaal uit twee routes. Naast het iets praktischer gerichte werk waar ik aan werkte, waren er meer fundamentele aspecten die Richard Gort in het kader van zijn promotieonderzoek verrichtte. Hij was het die mij in 1995 erop attentende dat het door mij ontwikkelde complete dynamische model van een afvalverbrandingsinstallatie de basis zou kunnen vormen van een promotieonderzoek. De verwachting dat dit model onderdeel zou worden van het uiteindelijke proefschrift, is echter niet uitgekomen, omdat de gekozen benadering in vergelijking met de andere beschreven onderwerpen in dit proefschrift, te toepassingsgericht is.

Geen promotie zonder promotor. Bert Brouwers zag in het tot dan toe door mij verrichte werk wetenschappelijke uitdagingen op het gebied van de stochastiek en zag analogieën met door hem eerder verricht werk.

Geen promotie zonder geld. Omdat ook TNO geïnteresseerd was in een promotieonderzoek hebben de verantwoordelijke managers Jan de Koning en later Sjaak van Loo een deel van de financiële middelen ter beschikking gesteld. Hierdoor is het voor mij gelukt om een goede balans te vinden tussen mijn eigen tijd en de door TNO beschikbaar gestelde tijd.

Tot slot wil ik Hilonka bedanken voor het kritisch doorlezen van het uiteindelijke manuscript. Als niet-technicus kijk jij naar andere zaken, hetgeen de leesbaarheid van het proefschrift zeker ten goede is gekomen. Verder wil ik je natuurlijk ook bedanken voor het vertrouwen dat jij steeds in mij hebt gesteld.

[Signature]

Robert
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Chapter 1

Introduction

Thermal processes like combustion are important to fulfil human's energy need. In the total energy supply the role of sustainable energy becomes more and more important. Waste incineration, biomass combustion and co-firing of biomass in power plants are examples of the implementation of sustainable energy in The Netherlands. However, compared with fossil fuels, waste and biomass have the disadvantage of a less constant fuel quality. Due to the variations in the fuel quality (the moisture content of the fuel, the composition of the combustible part and the density of the fuel) and the external requirements (low emission, high energy efficiency and high availability), the control of these processes becomes more important.

Up till now, the most common way to improve the control concept is by performing improvements in practice based upon practical experiences (trial and error). As combustion processes are multi-variable processes (multiple-input, multiple-output), which are often suffering from large disturbances in the fuel composition, it is difficult to understand the different output-input relations. Due to this complexity it is difficult to design or improve control systems by practical experiences. This trial and error method will lead to sub-optimal solutions. If one would understand the real process better and if the process could be described by mathematical functions, one could apply more modern techniques to develop or to improve control concepts. As control systems deal with dynamic process behaviour there is a need for dynamic process models, which describe the main dynamics of the process under study.

Unfortunately for thermal processes, general dynamic process models which are able to describe the process well, are not available in literature. Furthermore, especially in the case of municipal solid waste combustion there is no information available about validated models. Therefore the dynamics of thermal processes will be one of the main subjects of this thesis.

In general a control concept has two functions: set point tracking and disturbance rejection. In the case that the disturbances have a significant influence on the process it would be very useful to gather information about the disturbances for the design of control concepts. In this thesis a new on-line sensor will be discussed, which is capable to determine the disturbances acting on a thermal process. As thermal pro-
cesses are often suffering from large random variations in the fuel feed composition. The theory from stochastic processes can be applied. However, up till now the theory of stochastic processes has not been applied on thermal processes. In this thesis both the dynamics of thermal processes, and the application of the theory from stochastic processes to thermal processes are the main subjects. These subjects will be applied to municipal solid waste combustion, but can also be applied to other thermal processes like e.g. coal firing and wood burning. As the theory will be applied to municipal solid waste combustion the remaining part of this introduction will be used for a short introduction in the technology and mathematical modelling of municipal solid waste combustion.

1.1 Municipal solid waste combustion

1.1.1 Introduction

Since ancient times waste is burnt to get rid of waste and for producing local heat. For more than hundred years the grate furnace technology has been used for the combustion of waste, see figure 1.1 for an example of an old-fashioned municipal solid waste combustion plant. A lot of work has been done in optimising the system as burning a heterogeneous fuel like waste, is not an easy task. For this long period of time, the goals of waste combustion have continuously been changed and extended. Especially in the last decade a lot of extra requirements are governing waste combustion. The main goals for waste combustion are:

- volume reduction and inertisation
- low emissions
- reuse of solid residues

Figure 1.1: Diagram of an old-fashioned municipal solid waste combustion plant [1].
• energy production

• attractive economics

The main goal of waste combustion was, is and will be: volume reduction and inerti...

sions of dust, CO, HCl, SO₂, heavy metals, NOₓ and especially dioxins. Solutions
have been found in installing very complex flue gas cleaning systems. After reducing
the emissions, attention has been given to the quality of the flue-gas residues and
the bottom-ash. As modern MSWC plants can generate electricity from the heat
released during the combustion process, the improvement of the electrical efficiency
has become an important issue. However, the electrical efficiency is limited due to
the moderate steam pressure and temperature (mostly 400 °C and 40 bar). These are
limited due to the highly increased corrosion rate at higher steam temperatures. The
corrosion is caused by the chlorine and sulphur content of the waste in combination
with the deposition of ashes at the boiler surface. At the moment it can be stated that
as the environmental limits can be met, the economics of waste combustion becomes
the most important factor. As waste combustion seems to be proven technology, a lot
of suppliers are capable to build a modern waste combustion plant (as an example:
in the nineties five different suppliers have been build six new Municipal Solid Waste
Combustion (MSWC) plants in The Netherlands). So it becomes more and more

![Diagram of a municipal solid waste combustion plant.](image-url)

**Figure 1.2: Diagram of a municipal solid waste combustion plant.**

important to be able to build a modern plant that is economically attractive. This
means that a plant should have:

• a high availability,

• a high energy conversion ratio,

• a large flexibility for a wide variety of waste streams,
• a high quality of residues so that they can be reused.

The above mentioned issues can be realised by improving the technology. In reverse a technological improvement has to be judged with respect to the above given goals, e.g. what is the increase of the availability or the economic efficiency by implementing such a new technology?

Figure 1.2 gives a general overview of a typical modern MSWC plant. In the following subsections the parts of the MSWC plant which are important (waste feeding system, grate, furnace, boiler and control concept) for this thesis will be described. As the flue gas cleaning equipment is not relevant for the work described in this thesis no further description of the flue gas cleaning equipment will be given.

1.1.2 Waste feeding system

The waste feeding system is used to obtain a constant supply of waste to the combustion grate. Most suppliers use a feeding ram ('pusher') and a feeding table. The waste on the feeding table is pushed to the grate using the feeding ram. In general the ram moves forward with a speed comparable to the speed of the first grate section, while the backward movement of the ram has a much higher speed. In this way, the waste is almost continuously supplied on the grate.

1.1.3 Grate

The grate is one of the main components in a waste incinerator. Main objectives of the grate are: transport and mixing ('poking') of the waste. Mixing is important for a good burnout of the residues and for a good controllability of the process. Roughly, three types of grates can be distinguished:

• forward-acting reciprocating grate systems:
  parallel-acting
  counter-acting

• reverse-acting reciprocating grate systems

• roller grate systems

In both reciprocating grate systems the grate comprises of partially overlapping rows of grate beams. In the forward-acting and reverse-acting grate system the motion of the moving beams is respectively parallel to and opposite to the waste flow direction, see figure 1.3. In the forward-acting reciprocating grate systems two versions can be distinguished: parallel-acting and counter-acting. In the parallel-acting version the moving grate beams move in the same direction and in the counter-acting version the nearby moving grate beams move in the opposite direction. The latter version is supposed to have a larger mixing effect on the waste layer. Suppliers of reverse-acting reciprocating and forward-acting/counter-acting grate systems claim that their grate provides a constant recirculation of the waste on the grate. Because of this mixing, these grate
systems seem to be useful for waste with low and intermediate calorific values. The roller grate transports the waste by rotating large drums. On the drums itself almost no mixing of the waste takes place. Between two drums some mixing is possible.

1.1.4 Furnace

The furnace is used for a good gas mixing and post-combustion of unburnt flue gases with secondary air. The geometry of the furnace is very important for an optimum post-combustion with low emissions and a long lifetime of the furnace. Nowadays Computational Fluid Dynamics (CFD) software packages are used to optimise the geometry of the furnace and the configuration of the secondary air nozzles. In practice three main types of geometries are used (see figure 1.4):

1. a parallel flow or co-current furnace
2. a counter flow furnace
3. a cross or middle flow furnace

The above mentioned furnace names indicate the relationship between the main flow directions of the waste on the grate and the flue gas in the furnace. In the counter flow furnace the hot flue gases flow over the drying and devolatilising waste and support the ignition of the combustion process. The result is an intense heat contact between the hot flue gases and the cold waste. One problem with a counter flow furnace is that unburnt volatiles coming out of the waste on the first part of the grate, directly flow into the boiler section. Therefore, a good mixing of gases in the furnace is essential. Another problem of a counter flow furnace is that the fire could easily move to the beginning of the grate resulting in a fire of the waste in the chute. The cross flow furnace is comparable to the counter flow furnace, except for the fact that a cross flow furnace in general has a longer fire with a better burn-out. As in general, the
mixing of the gases in this furnace can be better controlled, compared to the counter flow type, the cross flow furnace is being preferred to the counter flow geometry. In a parallel flow furnace all the volatiles coming out of the waste, flow through the main fire. The hot gases also flow through the burnout zone and this improves the burnout of the bottom ash. In general parallel flow furnaces are run with a lower air factor. In practice the choice for a parallel furnace or a cross-flow furnace is determined by the preference of the client.

1.1.5 Boiler

In the boiler section, the heat of combustion is transformed into steam. The steam conditions (temperature and pressure) determine the electricity conversion efficiency. However, the steam conditions are limited compared to coal fired power plants. The reason is that the flue gases are much more corrosive compared to flue gases in conventional power plants. This is mainly caused by the relative high chlorine density in the waste, the low sulphur content in the waste and the high fly-ash load (which contains alkali and heavy metals). Therefore, in practice, steam temperatures from 350 to 425 °C are used. As a result of this the total nett electrical efficiency of present installations is limited to 18-23%.

During the last years, a lot of research has been carried out on the reduction of damages caused by corrosion [2]. Improvements of the boiler design, materials and the process control have decreased the corrosion problem. However, it is felt that corrosion will always be a major problem for waste combustion facilities. The steam temperature is not the only important parameter to reduce corrosion damages. Boiler design and process control should be optimised in such a way that the process conditions in the boiler are as uniform and constant as possible. A good burnout of the gases is also important to prevent corrosion in the boiler.
1.1.6 Control concept

The control concepts used in modern installations mostly have two main goals: achieving a constant steam production and oxygen concentration. To achieve this the four main control variables (primary air, secondary air, grate speed, and waste throughput) are directed by the control concept. However, due to the non-linearity of the process, the large fluctuations in the calorific value (waste composition) and the fact that all control variables have influences on all output variables (multiple-input multiple-output behaviour), in practice it is very difficult to elucidate the mutual connections. As a matter of fact, due to these problems, the present control systems look much like patchwork. For that reason during the last 10 years a lot of literature came available on advanced control concepts of MSW Combustion (MSWC). For example infrared cameras [3], [4] are used to determine the surface temperature of the waste layer, which is used into the control concept. Some suppliers combine this infrared-camera with fuzzy logic. Examples can be found in [5]. Others [6], [7] and [8] are using fuzzy logic to optimise their control concept. Also neural networks are used to optimise the processes, see [5] and [9]. However, all these techniques have only limited success due to the lack of understanding of the main phenomena occurring in the process. In order to increase the understanding of the combustion process mathematical modelling of the combustion process became popular during the nineties.

1.2 Mathematical modelling of MSWC

The use of mathematical models as a tool for better understanding and optimising the MSWC process has only yet started in the nineties. Due to the complexity and the inhomogeneity of the fuel, the research in this field has been scarce for a long time.

To study the conversion of waste, laboratory experiments in a packed bed with synthetic waste or wood have been carried out. Gort [10] gives an overview of the packed bed experiments which have been carried out for studying the combustion. Gort also derived a semi-analytical model, which can be used to explain the gathered results. Also, to be mentioned here is the work from Rogers [11], who was one of the first who carried out experiments with synthetic waste. The results from these batch experiments were translated in a theory for combustion of waste on a grate. Later in the nineties others performed research activities with pot furnaces [12],[13],[14],[15], which they combined with mathematical models to explain the results.

Due to the increased computing power during the last decade, the optimisation of fluid dynamics by applying Computational Fluid Dynamics (CFD) techniques has become the standard method. However, this technique describes only the gas phase behaviour, while the waste layer has a great influence on the gas phase behaviour and vice versa (e.g. release of gases, radiation waste layer to gas phase). To achieve more reliable results a waste layer model is used as a boundary condition to describe the composition and temperature of the gases which are released by the waste layer and the radiation of the waste layer. This boundary condition has to be an interactive one, because the gas phase radiates also to the waste layer. For that reason some authors [16],[17],[18],[19],[20] developed a stationary mathematical model of the waste layer,
which up till now has not been used as an interactive boundary layer.

About dynamic models on MSWC not much literature is published. Urban [21] and Boller [22] developed a model for a MSWC which is used as a simulator. In this model the incinerator is divided in only 5 zones. But, in spite of the rough split up, it is stated that it describes the observations in MSWC plants well, but no results are shown from these observations or from validation experiments. Furthermore, Rovaglio [23] and Kitami [24] proposed transfer functions for MSWC, where both used the transfer functions for control design. Also for controller design Becker [4] used open-loop step responses in order to estimate transfer functions with system identification techniques.

1.3 Scope of the thesis

Based upon the analysis given in this chapter the main topic of this thesis will be the research into the disturbances and dynamics of the MSWC process. As already said, the MSWC process suffers from large disturbances in the calorific value. At the start of this research it was obvious that for a good process analysis of the dynamics more information about the disturbances would be necessary. Therefore, a new on-line calorific value sensor was developed, which is described in chapter 2. The new on-line calorific value sensor makes it possible to monitor on-line important process variables like the calorific value and the water content of the fuel. The sensor is used to collect data from four different MSWC plants. Results from these MSWC plants will be presented. A comparison with traditional off-line methods and possible applications will be discussed as well.

After revealing the main disturbances of the process the study of the process dynamics can be performed. A mathematical dynamic model of the process is very useful for studying the dynamics of a process. Therefore, in chapter 3 a general model for the dynamics of thermal processes is derived. This general model is applied to MSWC, which yields a completely new model description of the MSWC process. However, a model has to be validated with practical data. Unfortunately, MSWC plants suffer from large disturbances, which makes a good validation complicated. As no good information for the validation of processes like MSWC was available in literature, new validation techniques have been applied to MSWC plants. The validation results will be presented. The results from the validation experiments will show that the combustion process in practice can become completely different when different primary air temperatures are used. Two situations with different primary air temperatures will be discussed in detail including the application of the derived dynamic model to explain the differences.

When the disturbances are measured with the on-line calorific value sensor from chapter 2 and a validated dynamic model of the process is available, the theory from stochastic processes can be applied to MSWC. This new application field of stochastics is discussed in chapter 4. The results obtained in chapter 2 will be used in this
analysis. Also new linear transfer functions for thermal processes will be given and applied to MSWC. Finally, applications of the new developed tools will be discussed. As already mentioned the validation experiments lead to the conclusion that the dynamics of the combustion process can change when the primary air temperature is changed. This was a new result, which has never been reported in literature before. For that reason during the research it was decided to start an extensive study into the influence of the primary air temperature on the combustion process. This has been performed by using laboratory experiments. In chapter 5 the results from this research will be presented. The existing theory for combustion of solid fuels is extended with a qualitative as well as a quantitative description of the influence primary air preheating. The new theory is used to explain observations from real plants and the results from system identification. Furthermore, the value of laboratory experiments for simulating the real combustion process on a grate is discussed.
Chapter 2

On-line determination of the calorific value of solid fuels

2.1 Introduction

In all kind of combustion systems, like e.g. grate fired systems and fluidised bed systems, solid fuels are converted into hot gases and these are in most situations used to produce heat and electricity by using steam boilers and steam/gas turbines. Maintaining a constant process operation is desirable from different points of view. The installed control concepts must be able to cope with the fluctuations in the solid fuel composition. In classic pulverised coal installations the quality of the coal changes only a little, due to small water content fluctuations. In biomass fuels the moisture concentration can change more, resulting in large disturbances of the biomass installations (fluidised bed, grate fired systems or under fired stokers).

Also fluctuations in the density of the fuel can lead to disturbances, for example in municipal solid waste combustion (MSWC) systems. As in such systems the input is volumetric, the amount of mass fed into the systems is continuously changing due to density fluctuations in the fuel. One of the problems is to determine on-line the calorific value of the municipal solid waste. If this calorific value is known continuously, it could be used within control concepts to reduce fluctuations in the output signals like steam production and oxygen concentration. Up till now there are no useful solutions available. MSWC installations mostly calculate the calorific value based on hourly or daily averages. The calculation of the calorific value is based upon solving the mass- and energy balance of the system. However, this method has several drawbacks; for the energy equation one needs information about temperatures, flows, and thermal efficiency. To determine the thermal efficiency heat losses are needed, which are difficult to measure on-line. Furthermore, the amount of mass fed into the system during this period is needed. Generally these values are obtained from the number of grabs that have been put in the chute during a certain period of time, assuming a constant mass per grab. Furthermore, the present used methods give average calorific values which are not very accurate and not on-line. In section 2.3.4
more attention to these methods will be given.
In the MSWC field there have been some attempts to determine the waste composition in the chute. One example is the installation of microwave equipment in the chute which should be able to determine the moisture fraction in the waste [25]. However, besides the fact that the results are not quite reliable, there are some other drawbacks which makes this application sub-optimal. Suppose that it is a good method to determine the moisture fraction. Since the composition of the combustible part of the fuel is still unknown, the calorific value can still not be calculated. Furthermore, the distribution of the waste in the chute is not homogeneous. With this method the moisture fraction is determined along a line through the waste in the chute. A grid of microwaves is needed in order to achieve a reliable value of the moisture fraction. Another disadvantage is that it is not exactly known at what time the measured waste is fed into the furnace. Furthermore, during the travel in the chute the density of the waste changes due to compression.

Recently another method has been introduced on the market [26]. Waste fed into the chute is analysed by a camera and quantified in different classes (e.g. color, dimensions, etc.). These interpretations are linked to the output signals by fuzzy logic. This method does not determine the calorific value, but it gives some information to predict the system behaviour.

However, based upon the aforementioned disadvantages of teh several methods it would be better to determine the composition and calorific value of the waste as soon as possible after entering in the furnace. For this purpose a new on-line calorific value sensor has been developed which will be treated in this chapter. This sensor has been applied for a patent [27].

2.2 On-line calorific value sensor

The main characteristic of the new developed sensor is the fact that it is only based on mass balances. No energy balance is used, as this would introduce additional uncertainties as already discussed in the introduction.

It is assumed that a fuel include three different fractions: moisture (\(X_{\text{moist}}\)), inert (\(X_{\text{inert}}\)) and combustibles (\(X_{\text{comb}}\)). The combustible part can be described by \(CH_yO_z\). The components in the flue gas are mainly \(CO_2\), \(H_2O\), \(N_2\), and \(O_2\). In normal operation there is good mixing of secondary air with the flue gases and the gas temperature is sufficient high (\(>800^\circ C\)). This results in a low concentration of \(CO(\leq 30 mg m^{-3})\) and the error made by ignoring the \(CO\) concentration is negligible.

If the wet volume fractions \(Y_{CO_2}\), \(Y_{H_2O}\) and \(Y_{O_2}\) in the flue gases are measured, they can be used to solve on-line the mass balances. For most processes there is a delay time between the combusting fuel and the analysers. But mostly this delay time (\(<1\text{ min}\)) is much shorter than the time constants of the system (10-60 minutes) and for that reason the time delay can be neglected.

The reaction equation for the combustion of \(CH_yO_z\) is straightforward:

\[
CH_yO_z + (\eta_C + \frac{1}{4}y - \frac{1}{2}z) O_2 \rightarrow (1 - \eta_C)C + \eta_C CO_2 + \frac{4}{2}H_2 O \tag{2.1}
\]
with \( \eta_C \) is the fraction of converted carbon. In MSWC this fraction is mostly between 0.9 and 1.0. The exact value is estimated from plant data. The influence of this value will be discussed in the next section where a sensitivity analysis is discussed. The total flue gas flow is calculated by the nitrogen molar balance:

\[
\phi_{f_g} = \frac{Y_{N_2,\text{air}}}{Y_{N_2}} (\phi_{\text{prim}} + \phi_{\text{sec}}) + \phi_{\text{recirc}} \tag{2.2}
\]

The molar flow of combustible material (\( \phi_{CH_2O_x} \)) follows from a carbon molar balance:

\[
\phi_{CH_2O_x} + Y_{CO_2} \phi_{\text{recirc}} = Y_{CO_2} \phi_{f_g} + (1 - \eta_C) \phi_{CH_2O_x} \tag{2.3}
\]

Combination of (2.2) and (2.3) leads to the molar flow of combustible material:

\[
\phi_{CH_2O_x} = \frac{Y_{CO_2} Y_{N_2,\text{air}}}{\eta_C Y_{N_2}} (\phi_{\text{prim}} + \phi_{\text{sec}}) \tag{2.4}
\]

The oxygen balance of the system yields the following equation:

\[
Y_{O_2,\text{air}} (\phi_{\text{prim}} + \phi_{\text{sec}}) + Y_{O_2} \phi_{\text{recirc}} = Y_{H_2O} \phi_{f_g} + \frac{1}{2} (2 \eta_C - 1) \phi_{CH_2O_x} \tag{2.5}
\]

Using the oxygen, carbon and nitrogen balance leads to the following relation for the \( z \)-value in \( CH_y O_z \):

\[
z = \frac{2 \eta_C}{1 - \frac{Y_{O_2,\text{air}}}{1-Y_{CH_2O_x,\text{air}}} \frac{Y_{O_2}}{1-Y_{CH_2O_x}}} (1 - \frac{Y_{CO_2}}{1-Y_{CH_2O_x}}) + \frac{1}{2} y \tag{2.6}
\]

with \( Y_{O_2,\text{air}} \) the wet volume fraction oxygen in the air supplied to the system and \( Y_{H_2O,\text{air}} \), the wet volume fraction water in the air supplied to the system. It has to be noted that the \( z \)-value is not a function of the amount of air, the waste fed to the system and the flue gas flow. This implies that it is also independent of the amount of leakage air. If the different gas concentrations are not measured at the same place, leakage air between the two measurement places can eventually occur (see section 2.2.1). As the variation of the leakage air as a function of time is mostly negligible, a stationary correction can be made in order to compensate the leakage air. The fact that the sensor is only dependent on fractions makes the determination of the calorific value more accurate, because of the difficulty in measuring mass flows accurate.

To use equation (2.6) the hydrogen molar ratio (\( y \)) has to be fixed. Therefore, an analysis of the waste composition was carried out to determine the \( y \)-value. On the basis of the standard composition of waste as used in the FACE model [28], an estimate of the variation in \( y \) and \( z \) has been made. In table 2.1 the composition of different components of combustible material is represented. Based upon data from table 2.1, the values of \( y \) and \( z \) can be calculated. These values are represented in table 2.2. The value of \( y \) therefore varies at a maximum between 1.6 and 1.8 and the value of \( z \) between 0.0 and 0.7. Based upon an average waste composition three different types of waste were selected, which are representing variations in the waste.
ON-LINE DETERMINATION OF THE CALORIFIC VALUE OF SOLID FUELS

<table>
<thead>
<tr>
<th>Component</th>
<th>C</th>
<th>H</th>
<th>O</th>
<th>Water</th>
<th>Inert</th>
</tr>
</thead>
<tbody>
<tr>
<td>Paper</td>
<td>0.3313</td>
<td>0.0173</td>
<td>0.3926</td>
<td>0.2364</td>
<td>0.0824</td>
</tr>
<tr>
<td>Plastic</td>
<td>0.0091</td>
<td>0.1030</td>
<td>0.0200</td>
<td>0.1000</td>
<td>0.0835</td>
</tr>
<tr>
<td>VFG</td>
<td>0.1800</td>
<td>0.0251</td>
<td>0.1304</td>
<td>0.0114</td>
<td>0.1381</td>
</tr>
</tbody>
</table>

Table 2.1: Standard composition of different combustible components (VFG= Vegetable, Fruit and Garden refuse.)

<table>
<thead>
<tr>
<th>Component</th>
<th>y</th>
<th>z</th>
</tr>
</thead>
<tbody>
<tr>
<td>Paper</td>
<td>1.713</td>
<td>0.005</td>
</tr>
<tr>
<td>Plastic</td>
<td>1.803</td>
<td>0.023</td>
</tr>
<tr>
<td>VFG</td>
<td>1.619</td>
<td>0.562</td>
</tr>
</tbody>
</table>

Table 2.2: y and z values of \( \text{CH}_y\text{O}_z \) for combustible components.

In Table 2.3 three different waste compositions are shown in which the plastic and the vegetables/fruit/garden refuse fractions are strongly varied. Referring to Table 2.3, the value of y is fairly constant for different waste compositions. A good estimate of y is 1.72. The sensitivity of y on the total result will be discussed in the next section. Finally, the heating value can be evaluated from the chemical composition of the fuel. E.g., with the well known formula of Michel (see for example [29]) the net calorific value \( (\text{MJ kg}^{-1}) \) of the combustible part can easily be calculated:

\[
H_{\text{CH}_y\text{O}_z} = \left(\frac{408.4 + 102.4y - 156.8z}{M_{\text{CH}_y\text{O}_z}}\right) \times 10^{-3} \tag{2.7}
\]

with \( M_{\text{CH}_y\text{O}_z} \) the molar mass of the fuel. So, the calorific value of the total fuel can be calculated if the moisture fraction and the inert fraction are known. First, the ash-free calorific value i.e. the inert fraction is not taken into account, is calculated. This is due because the inert fraction introduces an extra uncertainty in the calorific value, because the exact value is not known. For practical applications of an on-line sensor the amount of heat released is important. The ash-free calorific value is then the most important value. The moisture fraction of the ash-free fuel is defined as

<table>
<thead>
<tr>
<th>Component</th>
<th>Waste 1</th>
<th>Waste 2</th>
<th>Waste 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Paper</td>
<td>0.34</td>
<td>0.34</td>
<td>0.34</td>
</tr>
<tr>
<td>Plastic</td>
<td>0.11</td>
<td>0.23</td>
<td>0.05</td>
</tr>
<tr>
<td>VFG</td>
<td>0.37</td>
<td>0.23</td>
<td>0.33</td>
</tr>
<tr>
<td>Inert</td>
<td>0.18</td>
<td>0.18</td>
<td>0.18</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Total waste</th>
<th>Waste 1</th>
<th>Waste 2</th>
<th>Waste 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>y</td>
<td>1.71</td>
<td>1.75</td>
<td>1.69</td>
</tr>
<tr>
<td>z</td>
<td>0.46</td>
<td>0.32</td>
<td>0.54</td>
</tr>
<tr>
<td>Water</td>
<td>0.28</td>
<td>0.22</td>
<td>0.31</td>
</tr>
<tr>
<td>Inert</td>
<td>0.27</td>
<td>0.26</td>
<td>0.27</td>
</tr>
</tbody>
</table>

Table 2.3: Composition (mass fractions), y and z values for three different types of waste.
ON-LINE CALORIFIC VALUE SENSOR

follows:

\[ X_{H_2O_{ash-free}} = \frac{\Phi_{H_2O}}{\Phi_{H_2O} + \Phi_{CH_2O_2}} = \frac{\phi_{H_2O}M_{H_2O}}{\phi_{H_2O}M_{H_2O} + \phi_{CH_2O_2}M_{CH_2O_2}} \] (2.8)

assuming that \( Y_{N_2} = 1 - Y_{O_2} - Y_{CO_2} - Y_{H_2O} \). Furthermore, the molar flow of water \((\phi_{H_2O})\) follows from the water balance:

\[ \phi_{H_2O} + Y_{H_2O_{air}}(\phi_{prim} + \phi_{sec}) + Y_{H_2O}\phi_{rest} + \frac{1}{2}y\phi_{CH_2O_2} = Y_{H_2O}\phi_{fg} \] (2.9)

Substituting (2.4) and (2.2) in (2.9) leads to:

\[ \phi_{H_2O} = \left( \frac{Y_{H_2O}Y_{N_2_{air}}}{Y_{N_2}} - Y_{H_2O_{air}} - \frac{1}{2}Y_{CO_2}Y_{N_2_{air}}}{\eta_CY_{N_2}} \right)(\phi_{prim} + \phi_{sec}) \] (2.10)

The calculation of the ashfree moisture fraction is now straightforward by substituting (2.10) and (2.4) in (2.8):

\[ X_{H_2O_{ash-free}} = \frac{1}{1 + \frac{M_{CH_2O_2}}{M_{H_2O}}(\frac{Y_{N_2_{air}}}{\eta_CY_{N_2}}) - \frac{\phi_{CH_2O_2}M_{CH_2O_2}}{\phi_{H_2O}M_{H_2O} + \phi_{CH_2O_2}M_{CH_2O_2}})} \] (2.11)

The ash-free calorific value \( H_{ash-free} (MJkgashfree^{-1}) \) is then:

\[ H_{ash-free} = (1 - X_{H_2O_{ash-free}})H_{CH_2O_2} - X_{H_2O_{ash-free}}H_{evap} \] (2.12)

with \( H_{evap} \) the evaporation heat of water. As in the calculation of \( z \) it has to be noted that no flows are needed for the calculation of \( X_{H_2O_{ash-free}} \) and \( H_{ash-free} \).

The air factor, defined as the amount of air supplied divided by the amount of air needed for stoichiometric combustion of waste is:

\[ \lambda = \frac{\phi_{prim} + \phi_{sec}}{\phi_{air,comb}} = \frac{(\phi_{prim} + \phi_{sec})Y_{O_2_{air}}}{2(\eta_C + \frac{1}{2}y - z)\phi_{CH_2O_2}} \] (2.13)

Substituting (2.4) in (2.13) yields:

\[ \lambda = \frac{Y_{O_2_{air}}}{2(\eta_C + \frac{1}{2}y - z)(\frac{1}{\eta_C}Y_{N_2_{air}} + \frac{1}{\eta_C}Y_{O_2_{air}} - \frac{1}{\eta_C}Y_{CO_2})} \] (2.14)

It is noted that like the calorific value of the combustible fraction and the ash-free moisture content, the air factor is only dependent on concentrations. When also the inert fraction is known the moisture fraction of the waste and the net calorific value of the waste can be calculated:

\[ X_{H_2O} = (1 - X_{inert})X_{H_2O_{ash-free}} \] (2.15)

\[ H_{waste} = (1 - X_{inert} - X_{H_2O})H_{CH_2O_2} - X_{H_2O}H_{evap} \] (2.16)
The amount of converted material can be calculated if the air flow is known too:

\[
\Phi_{\text{waste}} = \frac{\Phi CH_4O + \Phi H_2O}{1 - X_{\text{inert}}} = \frac{Y_{CO2}Y_{N2,\text{air}} \cdot M_{CH4O}(\phi_{\text{prim}} + \phi_{\text{sec}})}{\eta \cdot Y_{N2} \cdot (1 - X_{H2O} - X_{\text{inert}})}
\]  \hspace{1cm} (2.17)

In case when the sensor is applied to a MSWC process, the density of the waste as a function of time can be calculated additionally if the feeder signal (dosage ram) is known on-line. This is calculated by dividing the total amount of waste that is converted by the volumetric waste input, as the ram doses volumetrically. The result is not the exact density, but it gives a good indication about the variation in the density.

In practical situations, the inert fraction can be taken constant. An error in the inert fraction has no influence on the ash-free calorific value. The inert part that leaves the combustion zone contains some convective energy and some unreleased chemical energy, but this is small compared to the total amount of released energy. Fluctuations in this part are negligible. Therefore, for calculating the systems energy balance the inert content will be neglected. The relation for the released thermal power is now easy to find if the air flow is measured:

\[
Q_{\text{thermal}} = (\Phi CH_4O + \Phi H_2O)H_{\text{ash-free}}
\]  \hspace{1cm} (2.18)

The amount of converted waste (ash-free) follows than directly from equation (2.4).

The flue gas flow is easy to find:

\[
\Phi_{fg} = \Phi_{\text{prim}} + \Phi_{\text{sec}} + \Phi_{CH_4O}
\]  \hspace{1cm} (2.19)

A complete energy balance is straightforward if the steam temperature, pressure, flow temperatures, the inert content and the radiation losses are known or assumed. The output of this energy balance are the steam production, the flame temperature and the overall efficiency.

**Summarising**

Based upon on-line measurements of the gas components CO₂, H₂O, O₂ and H₂O_{air}, from the mass balance the following quantities are calculated, see also figure 2.1:

- The net calorific value of the combustible part of the waste, i.e. without the inert fraction, also called the ash-free calorific value. This is the ash-free calorific value of the waste that burns at the grate at that moment, i.e. an average calorific value of the combustible fuel on the grate.

- The ashfree moisture fraction of the waste.

- The air factor

If the inert fraction is known, the net calorific value of integral (i.e. including combustible, water and inert) municipal solid waste, the total moisture content of the integral waste as also the carbon content in the bottom-ash can be calculated. If the primary, secondary and flue gas recirculation flow are measured on-line the following quantities can also be calculated:
Figure 2.1: Diagram of the on-line calorific value sensor

- The produced thermal power of the installation.
- A prediction for the steam production.
- The amount of waste that is converted into gas.
- The overall energy efficiency

The sensor can be validated in the following ways:

- Comparison of the predicted steam production with the actual steam production.
- Comparison of the calculated waste flow with the actual waste flow
• Comparison of the calorific value from the calorific value sensor with day averaged values from the incinerator. Most plants calculate day average calorific values.

The first way of validation provides the best picture of the quality of the sensor as, usually, direct on-line measurements of the steam production are readily available for a MSWC plant. Before discussing the validation results, first a sensitivity analysis will be discussed.

2.2.1 Sensitivity analysis

In a sensitivity analysis of the calorific value sensor static and dynamic sensitivity must be distinguished. Static sensitivity delivers information about the accuracy of the sensor when measured and calculated values are compared. Dynamic sensitivity is a measure for the accuracy of the sensor under fluctuating calorific values. In this case the actual calorific value is less important.

Static accuracy is influenced by measurement errors and eventually wrong assumptions. The influence of wrong assumptions on the results are depicted in figure 2.2 and 2.3. The results are based upon the aforementioned formulas. In the figures, the relative errors are based upon the difference between the assumed and actual value. The relative errors in the result are based upon the calorific value ($H_{\text{waste}}$), the amount of burning material ($\Phi_{\text{waste}}$) and the released thermal power ($Q_{\text{therm,al}}$).

Figure 2.2 shows that a relative error in the percentage unburnt material ($\eta_c$) or in

\[ Q_{\text{heat}} = \Phi_{\text{waste}} 

\]

the $H/C$ ($y$) ratio has only a slight influence on the calorific value. Figure 2.3 shows that an error in the inert percentage has some influence on the calorific value as well as on the amount of burning material. The calculated released amount of heat is not affected by an error in the inert percentage. This is caused by the fact that the
inert part has almost no thermal contribution to the process. Changes in the inert percentage eventually can be detected when the dosage ram signal is known. Figure 2.4 and 2.5 present the influence of measurement errors in the flue gas and the relative humidity of the used combustion air. The general conclusion is that it is necessary to measure the flue gas composition as accurate as possible. An error in the water percentage has the greatest influence on the calorific value as O₂ and CO₂ are measured on a wet basis. This results in an extra error in the CO₂ and O₂ concentrations on a dry basis. An error in the water content in the combustion air results in small, negligible errors. Figure 2.6 shows the influence of leakage air on the results. In general the calorific value is not affected by the leakage air. However the influence of the error in the leakage air on the amount of combustible material and thermal power is substantial. If the O₂ and the H₂O / CO₂ are not measured at the same place, leakage air between the two places can occur. In that case the error in the calorific value becomes unacceptably large. However, as the leakage air is usually a constant flow, this problem can be solved by correcting for the leakage air. The amount of leakage air is determined from the measured flue gas composition and their relative relations (see section 2.4). However, to overcome these problems it is recommended to measure the flue gas composition at one place.

**Systematic deviations and accuracy**

Based upon the above given analysis it follows that flue gas concentration measurements are causing the greatest error in the results. This means that these measurements needs to be very accurate. One of the possibilities to perform these measurements on-line is by an infrared analyser. Therefore, here the accuracy of the used infrared analyser is discussed.

The O₂ percentage is measured on a wet basis by a standard zirconium cell. The CO₂ and H₂O percentages are measured on a wet basis by an infrared meter (type:
Figure 2.4: Relative error (%) in the calorific value ($H_{\text{waste}}$), waste flow ($\Phi_{\text{waste}}$) and thermal power ($Q_{\text{thermal}}$) as a function of the relative error in the $O_2$ concentration in the flue gas (left) and the $CO_2$ concentration in the flue gas (right).

Figure 2.5: Relative error (%) in the calorific value ($H_{\text{waste}}$), waste flow ($\Phi_{\text{waste}}$) and thermal power ($Q_{\text{thermal}}$) as a function of the relative error in the $H_2O$ concentration in the flue gas (left) and the $H_2O$ concentration in the combustion air (right).

GM-500) from the company SICK [30]. The $O_2$ percentage is measured before the electrostatic filter and the $CO_2$ and $H_2O$ percentages are measured after the filter. The newest version of the equipment (type: GM-35) can also be implemented before the filter [31]. The inaccuracy of the used analyser (GM500) for the $CO_2$ and $H_2O$ measurements is 10% at the highest. The GM35 is more accurate with an inaccuracy of 5%. This is a systematic deviation and will lead to static deviations. This deviation doesn’t effect the dynamic behaviour of the sensor. Furthermore, this
infrared analyser has an internal test (cuvettes with calibration gas) to detect drifting. When drifting occurs an error message is produced. The reproducibility of the analyser is 100%.

Systematic deviations, caused by the concentration measurement errors can be found by analyzing the ratio of the CO₂ and O₂ concentrations. Based upon the mass balances only a limited number of combinations are possible. In section 2.4 this will be discussed in more detail.

Other systematic deviations, e.g. by wrong assumptions, can be detected by comparing the measured steam production with the predicted steam production. In the overall energy balance the energy losses have to be assumed. In this factor all systematic deviations are gathered, unless the factor becomes physically unrealistic. Then the comparison of the predicted and calculated steam production is a measure of the accuracy of the calorific value sensor. With this method the achieved accuracy of the calorific value sensor is about the same as the accuracy in the measurement of the steam production.

2.3 Results in practice

The on-line calorific value sensor has been tested at four different MSWC plants in The Netherlands. The first test was performed at the installation of NV Afvalverbranding Zuid-Nederland (AZN) in Moerdijk in the period November 1999 till February 2000. The second test was performed at the GAVI Essent Milieu Wijster (Essent) installation at Wijster in the period August 2000 till November 2000. The third test was performed at the NV Huisselcentrale N-H (HVC) installation in Alkmaar in the period December 2000 till February 2001. The last test was performed at the MSWC
Twence Afvalwerking (Twence) in Hengelo from March 2001 till June 2001. The results of the first three tests are discussed in this section. As the data from MSWC Twence does not generate new results, in section 2.3.4 the data from the MSWC Twence is only used for comparison with other methods to calculate the calorific value.

Besides the necessary information for the test, additional signals were collected with a time interval of 20 seconds. The registered signals are date, time, flue gas composition (O₂, CO₂, and H₂O), flame temperature, flow and temperature of primary and secondary air, flue gas recirculation, flue gas flow, temperature and pressure of steam, temperature and relative humidity of ambient air, and finally the dosage ram frequency. The actual amount of waste that is supplied by the grab is registered off-line for validation. The average inert fraction of the waste and the percentage unburnt are estimated based on data supplied by the MSWC plants.

2.3.1 NV Afvalverbranding Zuid-Nederland Moerdijk

The flue gas composition had to be adjusted once in order to compensate for the leakage air in the electrostatic filter. This adjustment does not affect the dynamics of the calorific value. Figure 2.7 gives the adjusted flue gas composition as a function of time over a period of 80 days. Changes in the flue gas composition are caused by changes in the calorific value of the waste or by changes in the air to fuel ratio. The data in the following graphs are filtered with a first order filter with a time constant of 133 minutes in order to achieve clearly visible trends. The calorific value of the total waste, the moisture fraction of the total waste and the calorific value of the combustible part of the waste are given in figure 2.8 as a function of time. As mentioned before it is possible to calculate the density of the waste with the measured dosage ram frequency and with the amount of waste that is converted on the grate. Figure 2.9 shows the results. The average value of the calculated density is only an indication.

![Figure 2.7: Flue gas composition as a function of time at AZN Moerdijk.](image)
Figure 2.8: Calorific value ($H_{\text{waste}}$), moisture fraction ($X_{\text{moi,wa}}$), calorific value of combustible part ($H_{\text{comb}}$) as a function of time at AZN Moerdijk.

Figure 2.9: Dosage mm frequency ($F_{\text{fan}}$), amount of waste that is burnt ($\Phi_{\text{waste}}$), waste density ($\rho_{\text{waste}}$) as a function of time at AZN Moerdijk.

The air factor is defined as the ratio of the volume of air supplied per volume of fuel to the volume of air theoretically required for complete combustion. Although the air factor is determined by the actual air to fuel ratio, the calorific sensor can only calculate the air factor from the flue gas composition. Figure 2.10 shows the waste flow, the airflow and the air factor.

Figure 2.8 shows clearly that the calorific value of the total waste is influenced by both the calorific value of the combustible part of the waste and the moisture fraction.
Some instabilities are caused by process stops and support burning with natural gas.

One method to validate the calorific sensor is comparing the measured steam production with the calculated steam production. The steam production is calculated by means of an energy balance. The main term of this balance is the product of the calculated calorific value and the calculated amount of waste that is burnt. Figure 2.11 shows that the measured and calculated steam production correlate very well.
Figure 2.12: Measured and calculated steam production as a function of time (1 day), AZN Moerdijk.

Figure 2.13: Average calorific value \( (H_{\text{waste}}) \) and density \( (\rho_{\text{waste}}) \) at AZN Moerdijk for a week’s period.

As the boiler system has a lot of mass, the boiler system has a time constant. The calculated steam production however is based upon the flue gas composition and has no time constant. For a good comparison the calculated steam production in figure 2.11 is filtered with a time constant comparable to the boiler system time constant. It has been found that the calculated steam production was 2.5 minutes ahead of the measured steam production, indicating that the time constant of the steam system is 2.5 minutes. The calculated amount of burning waste can be compared with the actual amount of waste that is supplied by the grab. The amount of waste that is supplied in the chute
by the grab is weighed and the average for every four-hour time frame is calculated for 45 days. Furthermore, figure 2.11 shows that the measured and calculated waste flow correlate strongly. Given the fact that the calculated steam production and the calculated amount of burning waste correlate strongly with the measured values and given the fact that the calorific value of the waste is proportional to the quotient of the thermal power (steam production) and the amount of burning waste, it can be concluded that the calculated calorific value is in good agreement with the actual calorific value. The dynamics of the measured and calculated steam production for one day is given in figure 2.12. This figure shows that the measured and calculated steam production correlate also at high frequencies.

Finally, all data are used to calculate an average calorific value for a week period. This is done to reveal periodicities in the waste composition due to the different waste supply during the day or night and during weekends. Figure 2.13 shows the results for the calorific value and the density. Day 0 corresponds to a Sunday (0:00 hour). During normal weekdays, the calorific values during the night are about 0.5 $MJkg^{-1}$ lower, compared with the day values. The density of the waste is in the beginning of the week slightly higher than at the end of the week.

### 2.3.2 GAVI Essent Milieu Wijster

The calorific value sensor is tested for 90 days at the GAVI Essent Milieu Wijster installation in Wijster. The sensor worked without noticeable disturbances. Adjustment of the flue gas composition was not necessary. This means that air leakage of the electrical filter did not occur. The CO$_2$ percentage in the flue gas showed a low frequent decreasing trend, while the H$_2$O percentage in the flue gas showed a low frequent increasing trend. These trends were caused by an increasing trend in the moisture fraction of the waste and by an increasing trend in the calorific value of the combustible part of the waste. The total calorific value of the waste did not change a lot because the trends compensated each other. The validation based on the comparison of the measured and calculated steam production was complicated by the fact that the primary and secondary air flow measurements were not reliable.

One interesting result of the sensor test is showed in figure 2.14. This figure shows an average calorific value and density for a week's period. The average calorific value shows the same tendency as at AZN Moerdijk. In normal weekdays during the night, the calorific values are lower than during the day. Here the difference is about 1 $MJkg^{-1}$. The density of the waste shows no clear tendency. Without data obtained over a longer period it is not possible to explain this phenomenon.

### 2.3.3 NV Huisvuilcentrale N-H Alkmaar

The calorific value sensor has been tested for 75 days at the NV Huisvuilcentrale N-H (HVC) located in Alkmaar. The sensor worked without noticeable disturbances. Because of the presence of leakage air the flue gas composition had to be adjusted. The validation showed that the leakage of air in the E-filter was not constant. The high frequent dynamic behaviour of the calorific value sensor was not influenced by this leakage changes. The calculated steam production could be used to predict the
Figure 2.14: Average calorific value \( (H_{\text{water}}) \) and density \( (\rho_{\text{water}}) \) at Essent for a week’s period.

actual steam production (about 4.5 minutes in advance). The most striking result of the sensor test is showed in figure 2.15. This figure shows an average calorific value and density for a week’s period. The average calorific value shows no clear tendency. The density of the waste during the weekends is a factor two higher than during the weekday. This is caused by the fact that during the weekends old waste (approximately 2 year) from the lower part of the bunker was burnt, which is more compact and therefore has a higher density.

Figure 2.15: Average calorific value \( (H_{\text{water}}) \) and density \( (\rho_{\text{water}}) \) at the HVC plant for a week’s period.
2.3.4 Comparison with other methods

At present MSWC calculated the calorific value of the waste with several different methods which are all based upon energy balances. Here, four different methods will be compared with the results from the calorific value sensor. The first three are applied by HVC Alkmaar, while the last one is applied by MSWC Twente.

HVC Alkmaar

The installation HVC Alkmaar uses three methods to calculate the calorific value. The first is based upon an energy balance over the installation. The energy content of the flue gases is calculated and of course the amount of waste fed to the system is used. Furthermore, some corrections for the energy content of the combustion air, support burners, fly-ash, bottom-ash and radiation losses are introduced in the calculation:

\[
H_{\text{waste}} = \frac{2.7132 \Phi_{\text{CH}_4} + 4.177 - 35.170 \Phi_{\text{steam}}}{\Phi_{\text{waste}}} + 0.024 + 0.1898
\]  

with \( H_{\text{waste}} \) the calorific value in \( MJ/kg \), \( \Phi_{\text{CH}_4} \) the amount of fed natural gas in \( m^3/day \), \( \Phi_{\text{steam}} \) the produced amount of steam in \( tonnes/day \), and \( \Phi_{\text{waste}} \) the amount of waste fed to the furnace in \( tonnes/day \). This calorific value is calculated on a daily basis in order to have a more accurate value of the waste flow. Figure 2.16 shows the calorific value calculated with the aforementioned formula and the calorific value sensor. It can be concluded that during the second half of the period and the first 10 days the signals are in reasonable agreement. During the rest of the period the differences are substantial. This is probably caused by inaccuracies in the methods. Inaccuracies in the HVC method are mainly caused by the fact that the energy content of the combustion air and the flue gas are assumed to be constant and by the measurement error in the amount of waste fed to the system. The error in the calorific value sensor is caused by the assumption that the percentage inert is constant and the possible occurrence of leakage air. However, during permanent application of the calorific value sensor the \( \text{CO}_2 \), \( \text{H}_2\text{O} \) and \( \text{O}_2 \) shall be measured at the same place, resulting in excluding errors due to leakage air.

The second method, the steam/\( \text{H}_2\text{O} \) method, is based upon the ratio of produced steam and the mass flow water in the flue gases. This mass flow water is calculated from the product of the moisture fraction in the flue gas, after the flue gas cleaning and the flue gas flow. The moisture fraction is corrected for the supplied amount of water in the quench and the spray dryer. These amounts are deduced from the changes in temperature difference over the different apparatus.

The third method, the steam/quench method, is based upon the ratio of the produced steam and the product of the temperature after the quench minus 60 °C and the flue gas flow. The steam/\( \text{H}_2\text{O} \) and the steam/quench method calculate the calorific value hourly. In figure 2.17 these methods are compared with the method of the on-line calorific value sensor. The steam/\( \text{H}_2\text{O} \) method corresponds good with the on-line sensor, while the steam/quench method clearly deviates. The dynamics of the second method is mainly determined by the flue gas temperature after the quench. Since the strong relation between the temperature after the quench and the calorific value of
Figure 2.16: Calorific value (filtered minute value) calculated with the sensor and HVC (daily average) as a function of time.

Figure 2.17: Calorific value calculated with the on-line calorific value sensor (filtered minute value) as a function of time and calculated with two different HVC methods (hourly average) as function of time.

The waste is uncertain and as the dynamics of the steam/quench method differs much from the steam/H2O method and the on-line calorific value sensor, the accuracy of the steam/quench method is considered to be low. Figure 2.18 shows a comparison between the steam/H2O and the on-line sensor method for a period of two days. The figure shows clearly that the two methods are in reasonable agreement.
ON-LINE DETERMINATION OF THE CALORIFIC VALUE OF SOLID FUELS

7.2
7.4
7.6
7.8
8
8.2
8.4
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Figure 2.18: Comparison of the calorific value calculated with the on-line calorific value sensor (minute value) and the steam/H₂O method used by HVC (hourly average) for a period of two days.

MSWC Twence

The calorific value sensor is tested for 55 days at the Twence Afvalverwerking. MSWC Twence calculates the calorific value with a regression formula given by Reimann [32]:

\[ H_{\text{waste}} = 1.133 \frac{\Phi_{\text{steam}}}{\Phi_{\text{waste}}} H_{\text{steam}} + 8T_f g - 801 \]  

(2.21)

with \( H_{\text{waste}} \) the calorific value in MJ kg\(^{-1}\), \( \Phi_{\text{steam}} \) the produced amount of steam in tonne s day\(^{-1}\), \( \Phi_{\text{waste}} \) the amount of waste fed to the furnace in tonne s day\(^{-1}\), \( H_{\text{steam}} \) the steam enthalpy in MJ kg\(^{-1}\) and \( T_f g \) the flue gas temperature after the boiler section. The regression formula is based upon the energy balance of the installation. Figure 2.19 shows a comparison of the on-line calorific sensor and the Reimann method. It can be noticed that after 35 days the values are in good agreement. The first 35 days there is a deviation between the two methods. The explanation for this deviation can be found in the measurement in the steam production. It was found that the measured steam production was not unambiguous. The peak at day 28 was caused by a temporary process stop.

Conclusions

The Reimann method and the steam/quench method give generally spoken reasonable results for the calorific value as daily averages. The on-line calorific value sensor has however clear advantages compared to the other methods. This method is based upon less assumptions and is on-line, which is important when this signal is used in a control loop. Furthermore, it generates more information than only the calorific value. It also calculates the moisture fraction, the amount of converted material, the
2.4 Possible applications of the calorific value sensor

In this section different applications of the on-line calorific value sensor will be discussed. Two main categories of application can be distinguished: the on-line calorific value sensor as a diagnostic tool, which has no direct influence on the process and integrating the on-line calorific value sensor in the control concept for reducing process fluctuations.

2.4.1 The on-line calorific value sensor as a diagnostic tool

The combination of the process parameters with the results from the on-line sensor give indications about the performance of the MSWC plant. Several process parameters can be verified on-line.

Composition flue gas

The O₂, CO₂ and H₂O concentrations in the flue gas are related to each other and only a limited number of combinations are possible. The possible concentrations of O₂ and CO₂ on a dry basis are restricted as is shown in figure 2.20. In this figure the linear relations between O₂ and CO₂ are given for average waste (waste is in this case comprised of biomass, waste and non-biomass), biomass and non-biomass (e.g. plastic, fuel from fossil origin). The assumption for this figure is the complete
burn-out of the fuel. When waste with a constant composition is burnt, the CO₂ and O₂ concentrations are located on that appropriate line. Shifting on the line is caused by changes in the air factor. Deviations from the line are caused by changes in the waste composition. However, the points have to be within the lines from biomass and non-biomass. Points located outside these lines are caused by measurement errors or too high concentrations of unburnt material in the bottom-ash. However, with normal waste this will occur only when the percentage unburnt material in the bottom-ash exceeds 50%. So, when the points are outside the restricted area, most probably this is caused by measurement errors or by leakage air between the measurement places in the case they are not measured at the same place.

Also the H₂O concentration has to be within limited values. For example, high percentages of H₂O can point to leakage from the steam system to the flue gas system. So, in this application early detection of leakage in the system will be possible.

Air and flue gas flow

When the total flue gas flow is measured in the plant, integration with the concentration measurement, combustion air flow measurements and a simple nitrogen balance can be used to check the flow measurements. Deviations in the nitrogen balance can be caused by measurement errors or by leakage air.

Thermal efficiency

When the right information is available, an on-line energy balance can be set up. In this way the thermal efficiency can be calculated on-line.
Air factor

With the flue gas composition it is possible to calculate on-line the air factor of the process. A high air factor has a negative influence on the boiler efficiency and the NOx emission. The on-line calorific value sensor can be used to run the installation with an optimal air factor.

Density waste

When the ram feeder frequency in the MSWC installation is known on-line, it is possible to receive a good indicator for the density of the waste. As the ram feeder doses volumetrically an indication of the density is received. This is important as old waste from the lower part of the bunker is more compact and has a great influence on the combustion performance.

2.4.2 Integration of the on-line calorific value sensor in the control system

The on-line calorific value sensor determines a number of important variables. If these variables are used within a control loop, it might be possible to reduce fluctuations in the process. Reduction of the fluctuations in the process can for example increase the throughput and the thermal efficiency. Furthermore, it can also be used for predicting the steam production and automatic preheating of the primary air. These possible applications will be discussed below.

Influence fluctuation reduction on waste throughput

As the gate fee of the waste is the main income for MSWC installations, it is financially attractive to maximise the waste throughput. The waste throughput in a MSWC installation is mostly thermally limited due to the fluctuations in the process. The fluctuation in the waste composition causes fluctuations in the gas phase temperature and consequently also in the boiler system. The maximum allowed temperature in the system, determines the maximum throughput. When the average capacity of the system is nearly equal to the maximum capacity, the thermal limit will often be exceeded. When the fluctuations are minimised it is possible to increase the average capacity without increasing the amount of excesses of the maximum capacity. When use is made of an advanced control concept in which the on-line calorific value sensor is integrated, it is possible to better cope with the fluctuations in the waste composition. Due to the fact that the sensor observes the causes of the changes and based on that information, a proper control action can be taken (feed forward control). Preliminary calculations showed that an increase of 1-5% should be possible depending on the specific plant situation.

Influence fluctuation reduction on energy efficiency

Due to the process fluctuations, the turbine in a MSWC installation is often not operating at the most optimal point. In general it can be stated that, if due to
reduction of the fluctuations the capacity of the MSWC installation increases, the efficiency of the turbine also increases. Calculations have shown that improvements of about 1%-point electrical efficiency can be reached.

**Using predicted steam production in the control loop**

The on-line calorific value sensor predicts the steam production. As in practice the boiler systems are suffering from time constants (4-8 minutes depending on the type of system) due to the accumulation in the boiler system, it is possible to use the predicted steam production signal in the control system. In this way an faster reaction on disturbances, caused by changes in the waste composition, is possible.

**Using the on-line calorific value sensor for automatic preheating of the primary air**

Primary air preheating is often used in MSWC’s. From an energy point of view the amount of energy in the primary air is negligible compared to the amount of energy in the waste fed in the system. However, air preheating has a clear influence on the process behaviour. In chapter 5, the influence of primary air preheating is presented in detail. Here it is shown that the preheating of primary air is an important variable to control the process. It depends on the local circumstances if primary air preheating has to be used or not. As the calorific value is the main indicator for the need of primary air preheating, the on-line calorific value sensor can be used as a direct indication for the amount of preheating. So, the calorific value sensor can be integrated in a control loop in order to control automatically the primary air preheating. In this way large fluctuations in the waste composition can be absorbed more easily by the system. If it is not controlled well, the furnace can be overfilled due to the reaction of the control concept on large negative fluctuations in the waste composition. A structural lower calorific value results in less steam than the desired set point. As a reaction, the system will increase the amount of waste, but the set point will not be reached due to the low calorific value of the waste. As a result the system will be overloaded with waste, which has a low calorific value. In the end, the amount of mass will become too high and the fire will extinguish.

Using primary air preheating has several process advantages, which will be discussed in chapter 5, but it has also the disadvantage that the waste throughput is reduced due to the energy content of the primary air. So optimal using of primary air preheating can result in a more flexible system, with an optimised waste throughput.

### 2.5 Conclusions

In order to be able to determine the calorific value, a new method for the on-line determination of the calorific value of solid fuels has been presented. Integration of a new hardware sensor, a mathematical model and physical knowledge of the conversion processes has led to a new calorific value sensor. Experiments with Municipal Solid Waste Combustion (MSWC) showed that the sensor is capable of continuously monitoring the calorific value of the waste being converted. Besides that, the sensor
generates other important process information, like for example the density and the water content of the waste. In this study, the data provided by the sensor are assessed as to their practical reliability and accuracy. The new developed method has been compared with traditional methods and it has been shown that the new method has several advantages. Furthermore, attention has been given to the application of the on-line calorific value sensor.
Chapter 3

Dynamics of thermal processes with application to MSWC

3.1 Introduction

Dynamic models of thermal processes can be a valuable tool to investigate control concepts and possible disturbance rejection capacities of the thermal process. In literature there are no dynamic models of thermal processes available, which have been extensively validated with data from real plants. For that reason, in this chapter a new model for the dynamics of thermal processes is presented, which is validated with data from real MSWC plants. First, based upon classical mass- and energy balances, general instationary equations will be presented. These can be used to study the dynamic phenomena which occur in different thermal processes e.g. combustion and gasification and different reactor environments like (grate-fired) ovens, bubbling fluidised beds, and circulating fluidised beds. Secondly, as validation of developed dynamic models is important in order to be able to use the model in practice, in section 3.3 validation of dynamic models by system identification will be discussed. Following that the theory of this chapter will be made suitable to apply on the MSWC process. Then two different cases of the MSWC HVC Alkmaar will be discussed in section 3.5. Some conclusions will be presented in the last section.

3.2 Instationary balances for thermal processes

A thermal process normally includes a furnace and a boiler section. In the next two sections the instationary balances for these two processes are discussed.
3.2.1 Instationary balances for the furnace

As in thermal processes for solid fuels a gas and a solid phase are present, the description of the process will contain mass balances and energy balances for the solid as well as the gas phase. First, the case when the fuel, which includes normally more substances, is considered as one component, will be discussed. Next the fuels with two components, i.e. water and combustible matter, will be discussed.

Balances for fuels including one solid material

Assuming that the furnace part of the thermal process can be regarded as the well known continuously stirred reactor (CSTR, see figure 3.1), the instationary mass and energy balance of the solid phase can be described by the following equations:

\[ \frac{dM}{dt} = \Phi_{in} - \Phi_{out} - RaM = (1 - X_{inert})\Phi_{in} - RaM \]  
\[ C_p \frac{d(M(T_s - T_0))}{dt} = \Phi_{in}C_p(T_{in} - T_0) - \Phi_{out}C_p(T_s - T_0) - RaMC_p(T_s - T_0) - \Phi_{prim}C_{pT}(T_s - T_{prim}) + RaM\Delta H + A\alpha(\epsilon gT_g - \alpha gT_s) \]  

Figure 3.1: Diagram of the furnace section of a thermal process section.
In equation (3.1) \( \Phi_{in} \) is a further to be defined fuel flow with specific composition (inert, water, and combustible). \( \Phi_{out} \) \((kg \, s^{-1})\) is equal to the amount of inert material \( X_{inert} \) \( \Phi_{in} \) that leaves the reactor unconverted. Furthermore, \( \Phi_{prim} \) is the amount of primary air fed to the system \((kg \, s^{-1})\), \( T_0 \) is a reference temperature \((298 \, K)\), \( T_{in} \) the temperature of the incoming fuel, \( T_s \) the temperature of the solid fuel in the reactor, and \( T_g \) the temperature of the gas phase in the reactor. All temperatures are in \( K \).

Finally, \( C_p \) and \( C_{p, s} \) are the heat capacities of the solid material and the flue gas in \( J \, kg^{-1} \, K^{-1} \). Some assumptions have been made in the derivation of the balance.

First, the heat capacity of the solid material is chosen to be the average of the heat capacities of the ingoing and outgoing material.

Secondly, it is assumed that the amount of unconverted material remaining in the ashes is negligible. Thirdly, it is assumed that the specific heat capacity of the air equals the specific heat capacity of the flue gas, which is not completely true as their values differ for about 10%. But as the air fed into the system, contains only a small part of the total energy released in the system, the error becomes less than 1%.

In working out the balance a term \( Ra \, M \, (T - T_0) \,(C_p - C_{p,s}) \) enters the balance. As \( C_p \) and \( C_{p,s} \) have about the same value, the mentioned term is negligible compared to \( Ra \, M \, X_{\Delta H} \, \Delta H \). Therefore this term is neglected.

The energy released from the reaction is described by \( Ra \, M \, X_{\Delta H} \, \Delta H \), where \( R \) the reaction rate \((kg \, m^{-2} \, s^{-1})\), \( \alpha \) the interfacial area \((m^2 \, kg^{-1})\) and \( M \) is the total mass in the system \((kg)\). \( \Delta H \) is the lower heating value of the fuel (i.e. the formed water during combustion remains in vapour phase). When combusting solid materials, the oxygen in the air, needed for combustion, will diffuse towards the surface of the solid material and will react with the solid material resulting into gases. Due to the often non-uniform distributions of the particles or due to a bad mixing of the air (for example channelling) not all the oxygen can reach the solid surface where the reactions will take place. In that case, gasification may occur at the surface of the solid particles. As in the direct surroundings of the particle no oxygen is available, this fuel gas will not be converted towards CO\(_2\) and H\(_2\)O. In general, above the bed there will be a good mixing of the gases due to e.g. secondary air injection. This good mixing in the gas phase will provide a good burn-out. As a result of the aforementioned phenomena, part of the combustion energy will become available in the gas phase instead of the solid phase. The factor \( X_{\Delta H} \) takes this phenomenon into account.

The last term indicates the energy transfer between the gas and the solid phase. It is written in its most general form so that it can be used for pure convective heat transfer, as well as in the case with radiative heat transfer as the dominant mechanism. \( \epsilon_g \) and \( \alpha_g \) are related to the emissivity and the absorptivity of the gas. In appendix A it is shown that if radiative heat transfer is the dominant heat transfer mechanism, which is normally described with \( T^4 \) terms, the transfer term can also be written as the last term of (3.3). It’s obvious that in case of pure convective heat transfer \( \epsilon_g = \alpha_g = 1 \) and that \( \alpha \) is determined by the classical Nusselt relationships.

Working out the derivative term in the energy balance and substitution of equation (3.1) in (3.2) for the energy balance:

\[
C_p \, M \, \frac{dT_s}{dt} = \Phi_{in} C_p (T_{in} - T_s) - \Phi_{prim} C_{p,s} (T_s - T_{prim}) + Ra \, M \, X_{\Delta H} \, \Delta H
\]
\[ + A \alpha (e_g T_g - \alpha_g T_s) \] (3.3)

The total mass balance of the gas phase is straightforward and is integrated in the energy balance of the gas phase:

\[ \rho_g C_p \rho \frac{dT_g}{dt} = \Phi_{prim} C_{p,T} (T_s - T_g) + \Phi_{sec} C_{p,T} (T_{sec} - T_g) + RaM(1 - X_{\Delta H}) \Delta H - A \alpha (e_g T_g - \alpha_g T_s) + RaM C_{p,T} (T_s - T_g) \] (3.4)

The first, second and the last term on the right-hand side describe the contribution of the convective heat from the primary, secondary air and the amount of solid material converted into gas, that enters and leaves the gas phase respectively. The third term describes the heat of reaction that is released in the gas phase, while the fourth term represents the exchange by radiation.

The mass balances for individual waste components are not taken into account. For oxygen, a stationary mass balance equation will be given below. However, first, the use of the reaction term \(RaM\) will be discussed. Gort [10] gives a summary of the use of various approaches to describe the chemical reaction rate term in heterogeneous reaction systems. One simple, but generally used description, is the formula for a first order surface reaction rate with externally limited mass transfer:

\[ \text{Reaction rate} = RaM = \frac{1}{\kappa_o} + \frac{1}{k_o \varepsilon_{\text{g},v} r_e} \frac{[O_2] M_{O_2} \alpha M}{\nu_{O_2}} \] (3.5)

here the reaction rate has the dimension \( (kg\text{ waste s}^{-1}) \) and \( \nu_{O_2} \) \((kg_{O_2} kgs^{-1})\) is the amount oxygen needed to convert stoichiometrically 1 kg of waste. When the composition of the waste is known, \( \nu_{O_2} \) can be calculated, otherwise the value has to be derived from plant data. The oxygen consumption \((kg_{O_2} s^{-1})\) follows direct from equation (3.5):

\[ R_{O_2} aM = \frac{1}{\kappa_o} + \frac{1}{k_o \varepsilon_{\text{g},v} r_e} \frac{[O_2] M_{O_2} \alpha M}{\nu_{O_2}} \] (3.6)

The oxygen balance is a gas phase balance where, compared to the solid fuel balance the accumulation term can be neglected. For that reason the oxygen balance is straightforward:

\[ \frac{(\Phi_{prim} + \Phi_{sec})}{\rho_{air}}[O_2]_{in} - \frac{R_{O_2} aM}{M_{O2}} = \frac{\Phi_{f_g} \alpha}{\rho_{f_g}} [O_2]_{out} \] (3.7)

Here the following relation for the flue gas flow is used:

\[ \Phi_{f_g} = \Phi_{prim} + \Phi_{sec} + RaM \] (3.8)

Neglecting the second order effect of the oxygen concentration in the flue gas calculation \((RaM)\) and as for a CSTR \([O_2]_{out} = [O_2]\) the oxygen concentration can easily
be found:

\[ [O_2] = \frac{\left(\frac{\Phi_{\text{prim}} + \Phi_{\text{sec}}}{\rho_{\text{air}}} \right) [O_2]_{\text{in}}}{1 + \frac{M}{\rho_{\text{fl}}}} \]  

(3.9)

In some processes like for example MSWC the oxygen at the place of reaction equals the oxygen concentration of the entrance. In that case \([O_2] = [O_2]_{\text{in}}\), the reaction rate becomes pseudo first order as the oxygen concentration in the reaction rate remains constant and the oxygen concentration follows directly from (3.7).

The mass transfer coefficient \(k_d\) has to be determined by classical Sherwood relations for mass transfer. However, as for now no Sherwood relation for waste combustion is available from literature, the following general Sherwood relation will be used:

\[ Sh = \frac{k_d d_p}{D} = C Re^{0.5} Sc^{0.33} \]  

(3.10)

where \(Re\) and \(Sc\) are the normal dimensionless Reynolds and Schmidt number, \(D\) the diffusion coefficient of oxygen in air and \(d_p\) the diameter of the particles in the solid phase. The following remark can be made about \(d_p\). In many solid fuel conversion processes no clear particles diameters can be distinguished. In that case, the used particle diameter is an indication for the interfacial area and is more or less an artificial diameter. In section 3.5.3 this point will be discussed in more detail after the model is validated with data from real plants.

Now, a set of three differential equations and one relation for the reaction rate has been derived. However, when the general time scales of the solid and gas process are compared for thermal processes the following inequality mostly holds:

\[ \tau_g = \frac{\rho_g C_{p_g} V_g}{(\Phi_{\text{prim}} + \Phi_{\text{sec}}) C_{p_{fl}} + A \alpha \epsilon_g} \ll \tau_s = \frac{C_{p_s} M}{\Phi_{in} C_{p_s} + \Phi_{\text{prim}} C_{p_{fl}} + A \alpha \epsilon_g} \]  

(3.11)

It implies that the gas phase can be described in a quasi-stationary way. The inertia of the solid phase is much larger, and the gas phase can be assumed to adapt instantaneously to changes in the solid phase.

This leads to the following set of basic equations:

Mass balance solid phase:

\[ \frac{dM}{dt} = \Phi_{\text{in}} - \Phi_{\text{out}} - Ra M \]  

(3.12)

Energy balance solid phase:

\[ C_{p_s} M \frac{dT_s}{dt} = \Phi_{in} C_{p_s}(T_{in} - T_s) - \Phi_{\text{prim}} C_{p_{fl}}(T_s - T_{\text{prim}}) + Ra M X_{\Delta H} \Delta H \] 

\[-A \alpha (\alpha_y T_s - \epsilon_g T_g) \]  

(3.13)

Energy balance gas phase (quasi-stationary):

\[ \Phi_{\text{prim}} C_{p_{fl}}(T_s - T_g) + \Phi_{\text{sec}} C_{p_{fl}}(T_{\text{sec}} - T_g) + Ra M (1 - X_{\Delta H}) \Delta H + A \alpha (\alpha_y T_s - \epsilon_g T_g) + Ra M C_{p_{fl}}(T_s - T_g) = 0 \]  

(3.14)
Reaction rate solid phase:

\[ RaM = \frac{1}{k_s} + \frac{1}{k_i \tau_{in}} \left[ \frac{[O_2]}{M_{O_2}} \right] \alpha M \]  

(3.15)

Equations (3.12)-(3.15) describe the main dynamics of the process and have to be solved simultaneously.

In thermal processes the oxygen concentration in the flue gas is often a main control variable. The gas temperature is mostly not a main control variable, as it is more difficult to measure it accurately than for example the steam production. This is one of the reasons that the steam production often serves as a main control variable in the control concept. Therefore, to use the model in practice, there is a need for relations describing the steam production. Before discussing the steam system, the model will be extended for fuels including two different solid materials.

**Balances for fuels including two solid materials**

In this section balances for a CSTR will be given for a fuel, including two solid materials. In this case the two solid materials are combustible matter including inert, referred to as 'comb', and moisture, referred to as 'mois'. The instationary mass balance for the solid phase can be described by the following equations:

\[ \frac{dM_{mois}}{dt} = X_{mois} \Phi_{in} - k_{evap} M_{mois} \]  

(3.16)

\[ \frac{dM_{comb}}{dt} = (1 - X_{mois}) \Phi_{in} - \Phi_{out} - RaM_{comb} = (1 - X_{mois} - X_{inert}) \Phi_{in} - RaM_{comb} \]  

(3.17)

In the first equation the kinetic rate for evaporation of the moisture is described by \( k_{evap} \). From the equations it is straightforward that if \( k_{evap} \) equals \( R \) both equations can be added resulting in equation (3.1).

For the energy balance of the solid phase the following equation can be derived:

\[ \left( C_{p,comb} M_{comb} + C_{p,mois} M_{mois} \right) \frac{dT_s}{dt} = \]  

\[ (1 - X_{mois}) \Phi_{in} C_{p,comb} (T_{in} - T_s) + X_{mois} \Phi_{in} C_{p,mois} (T_{in} - T_s) \]  

\[- \Phi_{prin} C_{p,s} (T_s - T_{prim}) + Ra M_{comb} X_{H} \Delta H_{comb} + k_{evap} M_{mois} \Delta H_{evap} \]  

\[ + \alpha (e_{gb} T_g - a_g T_s) \]  

(3.18)

with the straightforward definitions:

\[ \Phi_{comb} + \Phi_{mois} = \Phi_{in} \]  

\[ X_{comb-in} = \frac{\Phi_{comb}}{\Phi_{in}}, X_{mois-in} = \frac{\Phi_{mois}}{\Phi_{in}} \]  

\[ X_{comb} = \frac{M_{comb}}{M_{tot}}, X_{mois} = \frac{M_{mois}}{M_{tot}} \]  

\[ C_{p,s} = X_{comb} C_{p,comb} + X_{mois} C_{p,mois} \]  

\[ X_{H} \Delta H_{tot} = (X_{comb} \Delta H_{comb} + X_{mois} \Delta H_{evap}) M_{tot} \]  

(3.19)

it follows that when \( k_{evap} \) equals \( R \) equation (3.18) transforms into equation (3.3).
3.2.2 Instationary balances for the boiler

In thermal conversion processes the released heat is often converted into steam in a boiler system. The steam can be used for heating, but mostly it will be used to generate power by a turbine. In this section some simplified modeling approach will be described. This approach can well be used to model the heat transfer mechanisms in the boiler system.

In industry, natural circulation and forced circulation boilers are the well known type of boilers. As in thermal conversion systems usually circulation boilers are used, the emphasis will be put on them. For more information about boiler systems the reader is referred to [33]. Natural circulation boilers normally comprise three units: an economiser to heat up the feed water in the boiler system, the drum-evaporator section, and a superheater section where the steam is superheated. The drum-evaporator section is the central part of the boiler, where the water is evaporated into steam. On the flue gas side, the wall in the first stage of the boiler, i.e. the part after the secondary air injection, includes membrane walls. These membrane walls are part of the evaporator section. Then, there are different sequences possible. A normal sequence is that the flue gas flows successively through the evaporator, the superheater and the economiser bundles. Figure 3.2 gives a diagram of the boiler section for the flue gas side and the steam side. As the superheater is a normal heat exchanger which increases the temperature of the produced steam, it has no effect on the amount of produced steam. The economiser is also a normal heat exchanger which increases the temperature of the feed water. So also the economiser has no

![Diagram of the boiler section from the viewpoint of the flue gas in a MSWC plant and the steam system.](image-url)
direct effect on the amount of produced steam. However, if the temperature of the feed water leaving the economiser changes, it has an effect on the amount of produced steam in the evaporator boiler system. However, as in normal situations the variations in the temperature in the economiser section are low and as the heat transferred in the economiser section is only a small fraction compared to the evaporator-boiler section, the influence of the dynamics of the economiser on the amount of steam produced can be neglected. So, for describing the dynamics of the boiler system the central evaporator-drum system is the most important and has to be discussed in more detail. Figure 3.3 shows a diagram of the evaporator-drum section. The system comprises a drum, a downcomer section and a riser section. Due to the heat flux $Q$ transferred from the furnace side, water will evaporate in the riser section. Due to gravity forces and pressure differences this will result in a natural convective flow of steam/water which flows into the liquid of the drum. The presence of steam in the liquid part of the drum causes the shrink-and-swell phenomena which makes it difficult to describe the system by simple equations. Furthermore, these phenomena make it also difficult to control such systems. If all phenomena in this system should be studied extensively the system should be modeled as a distributed two phase system, which would result in partial differential equations describing the main variables as a function of time and place. In literature a lot of effort has been put in modeling boiler systems. See for example [33], [34], Aström [35] also did a lot of research in modeling boiler systems. He proved, in spite of the fact that the system is distributed, that the dynamics can be described by low-order non-distributed (i.e. only time dependent) models. The reason for this simplicity lies in the fact that the heat transfer in drum-boilers is very efficient due to boiling and condensation. For this reason all parts of the system in contact with the saturated liquid-vapor mixture will be in thermal equilibrium. When the pressure changes the stored energy in the liquid-vapour mixture is released or absorbed very rapidly. For the drum pressure Aström derived the following differential equation:

$$
\frac{dQ}{dt} = Q - \Phi_{f_d}(H_w - H_d) - \Phi_{steam}H_e
$$

(3.20)

With $Q$ the heat transferred from the gas phase to the boiler section, the second
term represents the increase of the enthalpy of the feed water entering the boiler section, and $H_w$ represents the enthalpy of the water in the boiler section, while $H_c$ represents the condensation enthalpy. The $e$ term results from the relation between internal energy and enthalpy. A good approximation of this term is:

$$e \approx \rho_w V_w \frac{\partial H_w}{\partial p} + mC_p \frac{\partial T_s}{\partial p}$$

(3.21)

with the subscript $w$ for water and $s$ for steam. $V_w$ represents the total water volume in the system, while $m$ is the total metal mass of the drum-boiler section and $C_p$ the specific heat capacity of the metal of the system. This equation shows that the dynamics of the evaporator-drum system is determined by the energy content of the water and the total metal mass in the system. Furthermore, $V_w$ and $m$ are known from construction data while the other terms are known or related to steam tables.

As the interest is in the amount of produced steam, equation (3.20) has to be connected with a hydrodynamical relation for the steam flow. The amount of required steam is determined by the user of the produced steam. The steam can be used for district heating or goes to a turbine where it is used to produce electricity. To simplify the system, the steam turbine can be regarded as a throttle valve. In valves, the amount of produced steam is dependent on the pressure, but it also depends on the extent of superheating. If the steam is saturated it behaves like water and if it is superheated for a great extent it behaves like an ideal gas. The Instrument Society of America [36] introduced the following relation for moderate superheating of steam:

$$\Phi_{steam} = C \rho^{(1-\alpha)} \rho^\beta$$

(3.22)

with $C$ a system dependent constant (valve area, adiabatic index $(c_p/c_v)$ amongst others), $\rho$ the steam density at the valve inlet and $p$ the steam pressure at the valve inlet. $\alpha$ is between 0 and 0.5 where $\alpha = 0$ holds for saturated steam and 0.5 for superheated steam. Equations (3.20) and (3.22) can be used directly to calculate the steam production. However, (3.22) can be linearised around the equilibrium value resulting in

$$\Phi_{steam} = C(1-\alpha) \rho_0 \rho^\beta (p - p_0)$$

(3.23)

which is allowed for most practical situations. Rearranging equation (3.23) and substituting in (3.20) results in the following first order differential equation for the steam production:

$$e \frac{1}{C(1-\alpha)} \left( \frac{p_0}{\rho} \right)^\beta \frac{d\Phi_{steam}}{dt} = Q - \Phi_{fd}(H_w - H_{fd}) - \Phi_{steam}H_c$$

(3.24)

If not much detailed information about the steam system is available, equation (3.24) can be used for simple calculations. The factor $\frac{1}{C(1-\alpha)} \left( \frac{p_0}{\rho} \right)^\beta$ is more or less a constant, which, when divided by $H_c$, equals the time constant $\tau$ of the steam system. The unknown factor can easily be calculated, as the time constant $\tau$ can be extracted from step responses. The amount of heat transferred from the gas phase to the boiler can be related to the gas phase temperature by a convective heat transfer relation:

$$Q = \eta \Phi_{fg} C_{p_f} (T_g - T_{out})$$

(3.25)
with $\eta$ an efficiency factor regarding the heat losses in the boiler. In view of an analytical solution of the system, the equation for the steam system makes the system more complex, as it changes from a second order system in a third order system. However, the temperature as a main observable will also give good results. This has the advantage that an analytical analysis can be performed more easily. As already mentioned before, for validation with plant data, the steam equation is necessary because in almost all thermal processes, the steam production measurement is more accurate than the gas temperature measurement and is together with the oxygen concentration the main control variable.

Summarising, the complete thermal boiler process is described by equations (3.12)-(3.15) and (3.24)-(3.25) with the following unknown parameters $X_{\Delta H}$, $a$, $k_d$, $E_a$, $k_0$, and $\nu_{C_2}$. These unknown values have to be fit to actual plant data. From plant data the stationary values of the quantities $\Delta H$, $\Phi_{\text{prim}}$, $\Phi_{\text{sec}}$, $\Phi_m$, $\Phi_w$, $\Phi_{fd}$, $\Phi_{\text{steam}}$, $[O_2]$, $p_0$, $H_w$, $H_{fd}$ are known. $\epsilon_g$ and $\alpha_g$ follow from literature [37], [38]. $\alpha$ is calculated with the relation given in Appendix A. The steam parameters determining the time constant of the boiler system, $e^{-1/(1-\alpha)}(\frac{p_0}{\rho})^\alpha$, can in practice be represented by a factor $A_{\text{boiler}}$, which can be fitted to practical data.

3.3 Validation of dynamic models

In order to use the developed dynamic models into practice (e.g. evaluation and design of controllers) the models have to be in good agreement with practice. Therefore validation of dynamic models is an important subject. In this section the use of system identification techniques for the validation of first-principles dynamic models is discussed. A more extensive treatment of the application of system identification on MSWC, including a theoretical background, can be found in Ljung [39]. More general information about system identification can be found in Ljung [40].

Validation by means of system identification

After the development of a model of a process, for example by means of first-principles (i.e. a model based upon physical/chemical laws, like the model in section 3.2, the most important question to be answered is: how realistic is this model? In order to answer this question, the model has to be compared with data obtained from the real process, showing that the model fulfills its requirements or revealing shortcomings of the model. A simple way of dynamical validating a model, is by applying the same step inputs to both the real process and the model and subsequently comparing the obtained step responses. This method, however, doesn’t work properly when large disturbances are acting on the real process because they distort the step responses strongly. This is the main reason why this method fails in case of MSWC processes, where due to the variation in waste composition large disturbances are acting on the process. In addition a disadvantage is the long experimentation time which is in particular the case for MIMO (multi input, multi output) processes with a large amount of transfer functions with dominant time constants. Furthermore, the step
response method is not very suitable for processes that are acting in closed-loop if one wants to validate the open-loop process. Validation by means of system identification can avoid all these disadvantages. The proposed way of validation by means of system identification techniques is as follows:

- Estimation of a black-box model based upon the measured data (a black-box model is a pure mathematical dynamic model without any physical meanings, that describes the input-output relations as a function of time) of the real process. (Note that this estimation procedure inherently contains a statistical validation procedure).

- Comparing the black-box and the to be validated white-box (first principles) model, dynamically by applying the same input signals to both models and subsequently comparing the responses. If the responses of both models differ too much the parameters of the white box model have to be adapted until the difference between the responses is sufficiently small.

This validation procedure is shown in figure 3.4. The estimated model is supposed to represent the dynamics of the real process to which the white-box model has to be compared and to be adapted dynamically. If the white-box model resembles the dynamics of the estimated black-box well, it is assumed that it also resembles the dynamics of the real process.

The white-box model can be used for controller evaluation and design. Why not use the black-box model directly for controller design? For sure, this is possible but only in one operating point as with the system identification techniques that are presently commercially available it is only possible to estimate models that are linear and time invariant and therefore can only be used in (a restricted area around) one operating point. For processes that operate over a wide operating range over which the dynamics vary significantly it is, for controller design, not sufficient to have a model of this process that is only valid in one operating point. This discussion discloses one disadvantage of the proposed validation method: it is possible to validate the white box model only in one operating point. This problem can be solved by validating the white-box model at several operating points; if the model gives a satisfactory resemblance with the dynamics of the real process at, for example, two operating points.

Figure 3.4: The validation procedure.
one can also assume a good resemblance for the dynamics lying in between these operating points. One could, of course, also resort to nonlinear system identification techniques but these are not yet readily commercially available and in addition may suffer from long estimation times due to the nonlinear and therefore recursive optimisation problem that has to be solved.

A useful alternative for the validation/adaptation method is to adapt the parameters of the white-box model directly to the experimentally obtained in- and output data \( u(t) \) and \( g(t) \). This is also referred to in literature as grey-box modeling. This will almost lead unavoidable to a nonlinear optimisation problem which might lead to long estimation times and problems with finding the global minimum.

Besides for validation purposes, system identification also gives valuable information about the process. It reveals relations between in- and output signals. This information can well be well used to analyse the process behaviour as will be shown in the next section. The rest of this section is devoted to a short introduction in the practical procedure of system identification.

**System identification in practice**

System identification techniques can be divided in parametric and non-parametric techniques. Here, only parametric system identification techniques are discussed. Basically, the way that is followed with these techniques is that a number of differential equations are postulated with a certain pre-specified order of which the unknown parameters are to be determined, i.e. estimated. This set of (parametrised) differential equations constitutes the black-box model. It describes the way in which certain variables of the process, which form the input variables of the resulting black-box model, are dynamically related to other variables of this process, which form the output variables of the black-box model.

The procedure of (parametric) system identification in practice is as follows:

1. **Experiments**
   Experiments are carried out in order to collect data from the real process, i.e. the process of which the dynamics is to be comprised in the black-box model. These experiments consist of exciting the process by means of user-defined signals like, for example, so called (P)RBS signals ((Pseudo) Random Binary Signals (see Ljung [40])). The variables that are excited are also the input variables of the resulting model and are often used to those variables that are used to control the process (the so called manipulating variables). At the same time a number of other variables are measured, which are often the variables that need to be controlled (the so called controlled variables). These latter variables are the output variables of the resulting black-box model. For example, in case of a MSWC process, a variable that could be excited is the fuel flow and the variable that simultaneously is measured is the steam production. The resulting black-box model then would have the fuel flow as an input variable and the steam production as an output. By means of this model the effect of variations in the fuel flow on the steam production could be simulated.

2. **Estimation of a black-box model**
During this phase the parameters of this model are fitted, in some optimal way, to the data that are obtained from the real process during the previous experimental phase. The way this is performed will be discussed later on.

3. Validation of the estimated black-box model

Statistical methods, like correlation tests are used to validate the estimated black-box model with real data, which are not used for the estimation of the model.

Usually, the experimental phase is the most critical phase due to a number of reasons. The most important reason is that no good model can be obtained when the experiments are badly performed. The estimation phase is performed by means of the so called prediction error method, which will not be discussed here. The interested reader is referred to literature, e.g. [39] and [40].

3.4 Application to municipal solid waste combustion plants

As already mentioned in the introduction, modeling of municipal solid waste combustion (MSWC) is a complicated task. In a grate-fired furnace the waste undergoes different thermal processes like drying, gasification, pyrolysis, burning and burn-out. Furthermore, there is interaction with the gas phase and the waste travelling along the grate. In the gas phase complex processes like radiation, mixing, and combustion of volatiles, which are released from the underlying waste layer, occur. To study the influence of different geometries and nozzles on the gas phase mixing, detailed Computational Fluid Dynamics (CFD) calculations are needed, see for example [16], [17], [18], and [19]. These programmes are in principle stationary and time-consuming. As the dynamic behaviour is the main goal in this work it is not necessary to study the gas phase in detail.

Gort [10] carried out a PhD study into the phenomena occurring during combustion in a packed bed. He studied this combustion process in a packed bed reactor experimentally as well as theoretically. Assuming there is no mixing of waste over the length of the grate, which holds very good in practice and is also confirmed by literature [41], a packed bed reactor can simulate the combustion process of burning waste that travels along a grate with constant speed (plug flow). Of course, the radiation interaction is not covered by this process, but it gives good understanding of the processes which occur in the waste layer. Gort found that systems with no primary air preheating incinerate like a moving reaction front with a distinctive layer between the hot burning zone and the cold zone. Recently other research groups found the same results [13], [15], [20], and [42]. The very small reaction front slowly moves down the waste pile during the transport of the waste along the grate. This means that only a fraction of the mass in the system burns. The rest of the mass is so to be said "inactive". However, some grate suppliers use primary air preheating. The function of air preheating is to evaporate the moisture in the "inactive" fresh part of the waste pile. Dependent on the calorific value of the waste and the amount of preheating, two combustion regions can be distinguished. When the waste is relatively wet, the
system will behave like a pipe-bowl process, as described by Gort. However, when preheated air is used, the process starts with the pipe-bowl process and evaporation of the moisture in the inactive part. Experiments in a packed-bed reactor have been carried out to study the phenomena. From these experiments a theory for combustion with primary air preheating is developed. This will extensively be discussed in chapter 5. The theory says that due to local circumstances, there can be a complete change of the combustion regime. At a certain moment there will be an almost instantaneous ignition of the remaining material. The reaction rate is significantly increased and the completion of the waste burning process will occur earlier in time, compared to the situation without the preheating of the primary air. It is obvious that with air preheating a smaller grate can be used. However, also in this situation there is an inactive, cold part and an active, hot part. The waste layer can be regarded as a number of "large particles". The reaction mainly will take place at the surface of the particles, resulting in a kind of shrinking core combustion. The released volatiles will react in the gas phase. The heat released in the gas phase will penetrate into the core and will help to heat up the cold inner part of the core.

Based upon the observed phenomena, the MSWC system will be made mathematically suitable for the theory of section 3.2. From the phenomena it is learnt that there are two distinctive solid phases, an active and an inactive part. During the process a part of the inactive passive (pas) part is "fed" to the active (ac) part of the mass continuously. This leads to the following general partial differential equations for the solid fuel on the grate in a MSWC (x is the place in the transport direction on the grate, L is the total length of the grate, \( i^x \) is the value of variable \( i \) on place \( x \)):

The mass balance solid phase:

\[
\frac{\partial M_x^x}{\partial t} = -v^x \frac{\partial M_x^x}{\partial x} + \Phi_{trans}^x - R^x \alpha^x M_x^{ac}
\]

\[
\frac{\partial M_{pas}^x}{\partial t} = -v^x \frac{\partial M_x^{pas}}{\partial x} - \Phi_{trans}^x
\]  

(3.26)

with the following boundary conditions:

\[
v^x \frac{\partial M_{pas}^x}{\partial x} \bigg|_{x=0} = 0
\]

\[
v^x \frac{\partial M_{pas}^x}{\partial x} \bigg|_{x=L} = \Phi_{out} = (1 - X_{inert}) \Phi_{in}
\]

\[
v^x \frac{\partial M_{pas}^x}{\partial x} \bigg|_{x=0} = \Phi_{in}
\]

\[
v^x \frac{\partial M_{pas}^x}{\partial x} \bigg|_{x=L} = 0
\]  

(3.27)

In the equations, \( \Phi_{trans}^x \) is the amount of mass transferred from the passive layer to the active layer at place \( x \). In the beginning of the furnace the active layer is formed due to radiation from the gas phase above. After a short time, still in the beginning of the first section of the grate, the burning rate of the active waste is high enough, becomes stable and can be described with the theory from Gort. This theory says that the burning rate is not dependent of the amount of "passive" mass. In equilibrium
therefore $\Phi_{trans}^x$ equals $R^x a^x M_{ac}^x$, where $R^x$ can be determined from Gort’s theory, but can also be described by relation (3.15). In the last relation the parameters have to be determined from practice.

If it is assumed that applying a step on the input ($\Phi_{in}$), the passive balances react almost instantaneously compared to the active mass balance, then the passive mass balance can be considered as quasi-stationary. Then integration along the length of the mass balances ($M_i = \int_{x=L}^{x=0} M_i^x dx$), using the boundary conditions, substitution of the passive balance in the active balance and the definition of the mass averaged reaction rate:

$$Ra_{ac} = \frac{\int_{x=0}^{x=L} R^x a^x M_{ac}^x \partial x}{\int_{x=0}^{x=L} M_{ac}^x \partial x}$$

yields:

$$\frac{\partial M_{ac}}{\partial t} = \Phi_{in} - \Phi_{out} - Ra_{ac} M_{ac}$$

which equals equation (3.12). The assumption of a quasi-stationary passive mass balance can be explained as follows from a physical point of view: the amount of passive mass has a direct influence on the amount of active mass, i.e. the thickness of the burning layer increases when the fuel flows are higher.

As the solid fuel layer is divided into two layers of active and passive mass the energy balances for both phases can be described by:

$$C_p, M_{ac}^x \frac{\partial T_{ac}^x}{\partial t} = -v^x M_{ac}^x C_p, \frac{\partial T_{ac}^x}{\partial x} + \Phi_{trans}^x C_p, T_m - \Phi_{prim}^x C_{p,prim} (T_{ac}^x - T_{prim})$$

$$+ R^x a^x M_{ac}^x v^x \Delta H^x + A^x a^x (v^x T_g^x - v^x T_{ac}^x)$$

$$C_p, M_{pas}^x \frac{\partial T_{pas}^x}{\partial t} = -v^x M_{pas}^x C_p, \frac{\partial T_{pas}^x}{\partial x} - \Phi_{trans}^x C_p, T_m$$

with the following boundary conditions:

$$v^x M_{ac}^x C_p, \frac{\partial T_{ac}^x}{\partial x} |_{x=0} = 0$$

$$v^x M_{ac}^x C_p, \frac{\partial T_{ac}^x}{\partial x} |_{x=L} = \Phi_{out} C_p, T_{out}$$

$$v^x M_{pas}^x C_p, \frac{\partial T_{pas}^x}{\partial x} |_{x=0} = \Phi_{in} C_p, T_{in}$$

$$v^x M_{pas}^x C_p, \frac{\partial T_{pas}^x}{\partial x} |_{x=L} = 0$$

It is obvious that the specific heat capacity, $C_{p, pas}$, is in general the same for the active and passive mass, but is different due to the different temperatures in the active and passive layers. However, in the elaboration of the energy equation, the passive mass will vanish and therefore the difference in $C_{p, pas}$ will not remain in the final equation. As the temperature of the passive layer normally does not change as a function of time the instantation term in the passive mass energy balance can be neglected. The same strategy as with the mass balance is followed; integration of the energy balance,
substitution of the passive energy balance, applying the boundary conditions and some straightforward definitions, yields the following energy equation:

\[ C_p, M_{ac} \frac{\partial T_{ac}}{\partial t} = \Phi_{in} C_p, T_{in} - \Phi_{out} C_p, T_{out} - \Phi_{prim} C_{prim} (T_{ac} - T_{prim}) + Ra M_{ac} X_{\Delta H} \Delta H + \alpha \left( c_s T_g - \alpha_s T_{ac} \right) \]  

(3.33)

This equation is similar to equation (3.3). In this situation all phenomena occurring in the waste layer are incorporated in the reaction rate term \( R \), where \( R \) equals (3.15). Furthermore, \( M_{ac} \) contains water, inert and combustibles. As already mentioned in section 3.2 the factor \( X_{\Delta H} \) indicates the distribution of the released energy between the solid and the gas phase. The equation for the gas phase is the same as derived in section 3.2. There it was already shown that the gas phase can be considered as stationary, resulting in (3.14).

For the oxygen concentration equation (3.7) is used instead of equation (3.9). This is because the oxygen concentration in the waste layer of the fuel bed equals the oxygen concentration of the primary air rather than the oxygen concentration in the gas phase.

Concluding, it can be shown that the general equations for a MSWC can be transformed into the equations (3.12)-(3.15). However, the integration implies that the calorific value \( \Delta H \) is the average calorific value from the waste that is converted on the total grate. So, this means that it is not the calorific value of the waste that enters the furnace. This seems to be a problem, but in chapter 2 it is shown that the calorific value sensor measures the integrated calorific value as a function of time. This fact will be used in the next chapter, where a stochastic analysis will be presented.

In the next sections validation of the dynamic model to the MSWC NV Huisvuilencentrale N-H (HVC) in Alkmaar is presented. In chapter 4 the implications of stochastic disturbances on a MSWC will be discussed, by using the linearised transfer functions to be derived in section 4.3.

### 3.5 Validation of the dynamic model

In this section the application of the system identification techniques at the NV Huisvuilencentrale N-H which is located in Alkmaar, The Netherlands will be discussed. Furthermore, the developed dynamic model from section 3.2 will be adapted to the circumstances at the HVC, and will be validated with the identification models.

Since 1995 the HVC installation (see figure 1.2 for a diagram of the installation) has operated and it comprises of three combustion lines each with a capacity of 18.5 ton h\(^{-1}\) waste with an average calorific value of 10 MJ kg\(^{-1}\). Each line has its own flue gas equipment in order to comply with the Dutch legislation on combustion. The used grate is a reciprocating grate, supplied by the Swiss firm Von Roll. It is a grate that moves discontinuously in the time, while the waste is more or less continuously fed into the system by a moving ram feeder. The four main input variables of the furnace section are the amount of primary air, the amount of secondary air, the grate speed (in fact the time between two movements), and the amount of waste fed in to the furnace (this depends on the velocity of the ram feeder). The closed-loop excitation
experiments are carried out by disturbing the four input variables simultaneously and gathering these variables together with the main output variables oxygen and steam production.

Because of the large time constants of the process, the duration of the experiments preferably should be a couple of days and around the same set point. In practice this is almost impossible, while during such long experiments the process never behaves in such a way that all data can be used. Therefore, a lot of experiments of a limited period (6-12 hours) have been performed, which together form the total experimental data set. To use these different data sets for identification some special software was used to merge these data sets [43].

Another important point to mention is the low frequency behaviour of the estimated identification models. The used ARX method has the disadvantage that the low-frequency behaviour of the models is not so well estimated, compared to the higher frequency part. If information about the static behaviour is known in advance, it is good to use this in the estimation. For this purpose software was used to enforce static gains on the estimated models [43]. The operating point values at which the identification experiments were performed are displayed in Table 3.1. With the above described system identification techniques black-box models were estimated from the obtained data. The input-output relations can be used to simulate the process and compare these simulations with simulations carried out with the physical-chemical model, derived in section 3.2. Therefore the physical-chemical model was adapted to the situation under study by fitting the different parameters of the model \( (E_a, k_0, k_d, a, X_{\Delta H}, \nu_{O_2}, \varepsilon_g) \). Experiments were carried out in two different working points. First, under normal conditions the primary air temperature is about 120 °C. Some experiments were carried out in the weekend with fresh relatively dry waste. In this case it was possible to use a lower primary air temperature \( (T_{\text{prim}} = 70 \degree C) \). All these identification results were compared with results from the physical-chemical model by using step response curves. First the lower primary air temperature will be discussed.

### Table 3.1: Operating point values of the MSWC plant HVC during the identification experiments.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Inputs</strong></td>
<td></td>
</tr>
<tr>
<td>Waste inlet flow</td>
<td>18.5 [ton/h]</td>
</tr>
<tr>
<td>Speed of grate</td>
<td>79.9 [%]</td>
</tr>
<tr>
<td>Primary air flow</td>
<td>52277 [m³/h]</td>
</tr>
<tr>
<td>Secondary air flow</td>
<td>10580 [m³/h]</td>
</tr>
<tr>
<td><strong>Outputs</strong></td>
<td></td>
</tr>
<tr>
<td>Steam production</td>
<td>16.3 [kg s⁻¹]</td>
</tr>
<tr>
<td>Oxygen</td>
<td>5.7 [%]</td>
</tr>
</tbody>
</table>

3.5.1 Results from identification at \( T_{\text{prim}} = 70 \degree C \)

During normal process operation HVC is using a primary air preheating temperature of 120 °C. However, during the experiments HVC had one weekend fresh, loose, open waste. In that weekend the primary air temperature was lowered to 70 °C. The presented dynamic model has been adapted to the present plant under study. The
parameters that have to be determined are: \(E_a, k_d, \alpha, k_0, X_{\Delta H}, \nu_{O2}, \epsilon_g\), and the coefficient for the boiler system \(A_{boiler}\). The mass transfer coefficient \(k_d\) in relation (3.15) can be determined by a Sherwood relation. Later in section 5.3 it is shown that due to channelling in the waste layer a channelling factor \(X_{air}\) is introduced. This factor is defined as the fraction of air that goes through the waste layer into the gas phase without having interaction with the solid material in the bed. This means that when using Sherwood relations for calculating \(k_d\), the superficial velocity has to be corrected with the factor \(X_{air}\).

The use of Sherwood relations has the disadvantage that more parameters have to be determined e.g. the particle diameter, the porosity and a constant linear factor \(C\) (\(Sh = CRe^{0.5}Sc^{0.33}\)). As most of these values are not known and the number of unknown variables is high, an infinite number of solutions are mathematically possible. Therefore a different way of fitting was used. First, a linearised version of the reaction rate (3.15) was used. It was found that it was impossible to find any possible physical acceptable solution when the reaction rate was described with equation (3.15). Physically acceptable solutions were only possible when the system was assumed to be fully mass transfer controlled. This means that the kinetic rate term in equation (3.15) is negligible compared to the mass transfer coefficient, in other words \(R = k_d\). The mass transfer coefficient is dependent on the velocity and temperature. An analysis with general Sherwood relations has shown that for the circumstances in the fuel bed of a MSWC plant, the mass transfer coefficient is linear in the temperature, while it is allowed to linearise the dependency of the mass transfer coefficient on the primary air velocity. For that reason, first in the calculations the mass transfer coefficient \(k_d\) is described with a linear temperature and velocity dependency. In section 3.5.3 the \(k_d\) values for the situations with different primary air temperatures will be discussed from a physical point of view.

In table 3.2 the most important values have been reported for the situation with \(T_{prim} = 70^\circ C\) as well as with \(120^\circ C\), which were found from the validation. In

<table>
<thead>
<tr>
<th>(\Delta H) (MJ kg(^{-1}))</th>
<th>(A(m^2))</th>
<th>(X_{inlet}) (kg kg(^{-1}))</th>
<th>(A_{inlet})</th>
<th>(\epsilon_g)</th>
</tr>
</thead>
<tbody>
<tr>
<td>8.6</td>
<td>39</td>
<td>0.22</td>
<td>2.08 \times 10^{-5}</td>
<td>0.88</td>
</tr>
</tbody>
</table>

*Table 3.2: Model parameters for both the two primary air temperatures situations.*

The results it was found that in the situation with \(T_{prim} = 70^\circ C\) the system can be described by the "one component" equations \((3.12)-(3.13))\), while at \(T_{prim} = 120^\circ C\) the system has to be described with the "two component" equations \((3.16)-(3.18))\). The values of the 120°C will be discussed in more detail in the next section. The differences between the two situations will be discussed in section 3.5.3. \(\Delta H, A,\) and the \(X_{inlet}\) are based upon information received from the HVC installation. The surface of the burning bed layer \(A\) is used to calculate the appropriate superficial primary air velocity. It has to be noticed that during these experiments (1998) the on-line calorific value sensor was not implemented on that time. This implies that the mentioned calorific values are received from global plant calculations. Especially the value for the lower temperature has to be taken with care as it was different
from the normal case as already discussed. The value was probably (slightly) higher than during normal operation. The fitting procedure has shown that, based upon the available information, the products \( k_{\text{a}1}/v_{\text{O}2} \) and \( k_{\text{evap}} \) are fitted. This means that the number of possible combinations is infinite, i.e. the physical parameters cannot be derived from these values. In section 3.5.3 this will be discussed in more detail.

As in Appendix A is shown that for solid fuel flames \( \alpha_3 = \epsilon_3 \), in the table only a value for \( \epsilon_3 \) is given. In figure 3.5 the responses for a step on the waste flow is given (changes

<table>
<thead>
<tr>
<th>( T_{\text{prim}}(\degree \text{C}) )</th>
<th>( M_{\text{comb}} ) (kg)</th>
<th>( M_{\text{moist}} ) (kg)</th>
<th>( k_{\text{a}1}/v_{\text{O}2} ) ( (m^3 \text{kg}^{-1} \text{s}^{-1}) )</th>
<th>( k_{\text{evap}} ) ( (kg m^{-2} \text{s}^{-1}) )</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>5047</td>
<td>0</td>
<td>2.9 ( \times 10^{-3} )</td>
<td>0</td>
</tr>
<tr>
<td>120</td>
<td>1006</td>
<td>1/4</td>
<td>2.5 ( \times 10^{-3} )</td>
<td>7.0 ( \times 10^{-3} )</td>
</tr>
</tbody>
</table>

Table 3.3: The model parameters that are different for the two primary air temperatures situations. Note: \( M_{\text{mois}} \) for \( T_{\text{prim}} = 70\degree \text{C} \) includes combustible, moisture and inert, while \( M_{\text{comb}} \) for \( T_{\text{prim}} = 120\degree \text{C} \) includes only combustible and inert.

![Figure 3.5: Steam production response from a step applied on the waste input. Comparison of the estimated black box model (---) with the first principle dynamic model (-); \( T_{\text{prim}} = 70\degree \text{C} \).](image)

are related to the working point. In all further figures in this chapter the responses are related to the working point. The figure shows a good similarity between the identification (black-box) model and the physical-chemical (white-box) model. Two striking facts can be noticed. The first is the overshoot and the second is the time lag before the steam production gradually increases. The latter is explained by the heating up of the extra waste that enters the furnace. The overshoot is explained in the following way: the additional amount of waste results in a lower air-to-fuel ratio. Consequently the temperature will rise and the total conversion rate will be higher. This accelerates to a higher temperature, but at the same time due to the higher reaction rate the total mass on the grate will decrease. Due to this decrease the conversion rate will decrease and in the end a new equilibrium state will be
established. Figure 3.6 shows the corresponding O₂ concentration for both models.

![Graph showing O₂ concentration response](image)

*Figure 3.6: Oxygen concentration response from a step applied on the waste input. Comparison of the estimated black box model (---) with the first principle dynamic model (---); T_{prim} = 70°C.*

In Figure 3.7 the response of the steam production on a step applied on the primary air flow is shown. It is seen that due to the increase of the primary air, the combustion becomes more intense and will result in a higher steam production. This will decrease the amount of mass in the system and consequently the steam production decreases again and stabilizes around the zero line. This is a logical process, as the amount of energy entering the system has not changed. In fact, in practice small deviations around zero are possible due to changes in the amount of unconverted material in the bottom-ash. In the end, the intensity of the burning will be higher, resulting in a shorter fire on the grate and less mass on the grate. Figure 3.8 shows the steam production as a function of time for a step on the grate speed. At a first glance there is no parameter for the grate speed available in the developed model. However it turned out that the influence of the grate speed can be described by the interfacial parameter $a$. The system under study has a discontinuously moving grate, i.e. the grate bars are only moving during a short period of 15 seconds. The time between the movements is determined by the control system. The influence of increasing the grate speed (more grate movements per unit time) in this system can be explained as a "poking" effect, which increases the interfacial area. In the model this is implemented by a linear relationship between the grate speed and the interfacial area. The shape of the response curve is similar to the one of the primary air step, but less effective as can be concluded from the height of the response. From a physical point of view this means that the fire is more intensive, the total burning mass will decrease and the fire length will shorten, while the stationary value will not change much.

Figure 3.9 shows the steam production as a function of time for a step on the secondary air. Compared to the previous figures the similarity between theory and experiments is less. The explanations for this deviation can be found in the influence of the
Figure 3.7: Steam production response from a step applied on the primary air. Comparison of the estimated black box model (---) with the first principle dynamic model (-); $T_{\text{prim}} = 70^\circ C$.

Figure 3.8: Steam production response from a step applied on the grate speed. Comparison of the estimated black box model (---) with the first principle dynamic model (-); $T_{\text{prim}} = 70^\circ C$.

secondary air on the dynamics of the process. The secondary air does not have a large influence on the dynamics of the process. The main task for the secondary air injection is to achieve a good mixing in order to comply with the emission legislation. So it has an influence on the gas phase, but the dynamics of the gas phase are very fast compared to the solid phase. However, due to radiation interaction a change in the gas phase has an influence on the waste layer, which will influence the steam production with a delay caused by the time constant of the fuel layer. However, this
influence is too weak to explain the curve. The original model gives a direct negative response of steam production when a step is applied to the amount of the secondary air. The only method to incorporate the temporarily response is by adding an extra linear influence of the secondary air dependency on the reaction rate \( k_d \). From a physical point of view, this means that the secondary air has an influence on the mass transfer in the waste layer, which cannot be explained for the moment.

On the other hand, the influence of the secondary air on the steam production is very weak. Due to this weak influence, it is very difficult to excite the secondary air in the plant experiments to such a level that its influence can be found in the output variables. As the secondary air is the less important control parameter for this moment, no special attention has been given to the secondary air response. Future research have to be carried out to investigate the real secondary air response.

The presented figures show the influence of the main input variables on the output variables. From these results a clear control strategy is found. When the set point of the steam production is changed, the system has to react as soon as possible. For the short term, this can be achieved by changing the amount of primary air or the grate speed. As these variables will not affect the stationary value of the steam production over a longer time, at the same time the waste inlet flow has to be increased. This combination of adapting input variables has the result that the system will react much faster, which is one of the main purposes of a control concept. Figure 3.10 shows the predictions of the steam production and the oxygen concentration when only the waste inlet flow is excited and the other input signals are kept constant, i.e. are zero in the case of the estimated model and at the operating point values in the case of the first-principles model. One can see that the predictions of both the steam production and oxygen concentration almost perfectly coincide.

A similar good agreement between the predictions made by the estimated model
and the first-principles model was found with the primary air and grate speed as input variables. These results will not be shown here. The results indicate that the estimated model and the first-principles model show a good similarity and can be used to describe the dynamics of the MSWC plant.

3.5.2 Results from identification at $T_{prim} = 120^\circ C$

The situation with $T_{prim} = 120^\circ C$ also starts with an active and a passive layer. However, also evaporation of moisture occurs in the passive zone. After a certain moment, ignition occurs and the complete layer is burning. At that moment the evaporation and combustion of the fuel occur almost at the same time, like in the combustion layer in the pipe-bowl regime. The combustion of the complete layer can be interpreted as an active outside layer and a cold passive inside layer, like a shrinking core model. So, this process can also be described with active and passive balances, but with a fuel including two components (moisture and combustible/inert). So, the system has to be described with the more general equations (3.16)-(3.18). However, simulations showed that it was not possible to receive a good agreement between the experimental and theoretical response. Especially the response on a step on the waste flow induces an inverse response in the steam production. As in the model radiation is already taken into account it would be expected that an increased evaporation results in a lower gas phase temperature and the subsequent lower radiation intensity to the waste layer will decrease the total reaction rate. However, this effect is not large enough to explain the obtained results. However, it was possible to achieve a good agreement between the experiments and the theory by extending the reaction
rate term with a linear relation in the amount of evaporated water:

\[
RaM_{comb} = k_d \frac{[O_2]_0 M_{O_2}}{\nu_{O_2}} \left( 1 + k_{evap} \frac{\Phi_{evap} - \Phi_{evap0}}{\Phi_{evap0}} \right) aM_{comb}
\]  

(3.34)

with

\[
\Phi_{evap} = k_{evap} aM_{ mois}
\]  

(3.35)

This indicates that a step on the moisture fraction has a more direct influence on the total reaction rate of the fuel layer. In chapter 5 the different behaviour of the combustion of waste at different primary air temperatures will be treated extensively. But for the moment a good physical fully accepted explanation is not available. More research is needed to elucidate this phenomenon.

In table 3.3 the relevant values for this situation were already presented. The steam production response of a step applied on the waste inlet flow shows the influence of the moisture equation. In the response from the system identification a clear inverse response was found. Figure 3.11 shows the prediction of the steam production for a step applied on the waste flow for the estimated model and the first-principles model. Clearly the inverse response is shown which can be explained by the extra differential equation in the model. Due to the step change more moisture enters the system. This moisture will first evaporate before the combustion will increase. In section 5.4 a theory for combustion of solid fuels on grates will be presented. In that section, the two primary air temperature situations will be treated more extensively from a physical point of view. The occurrence of an inverse response in MSWC is also described by Rovaglio [23], who explained this structure upon physical processes. He described the system with two first order systems, which was used to develop a control system.

Compared with the lower primary air temperature, there is no overshoot in the response, which is mainly caused by the moisture equation. Furthermore, in the first
part of the response the similarity is better than in the second part. The response obtained from system identification shows a somewhat strange bend. For the moment a physical explanation is not available. The used system identification method is supposed to be the major contribution to this deviation. As the used ARX identification method is inherent less equipped to estimate the low-frequency behaviour of processes [39], it is supposed that the shown results can be found under specific circumstances. Figure 3.12 shows the corresponding O$_2$ concentration for both models. Figure 3.13

![Graph showing O$_2$ concentration response](image)

**Figure 3.12:** Oxygen concentration response from a step applied on the waste input. Comparison of the estimated black box model (---) with the first principle dynamic model (---), $T_{prim} = 120^\circ$C.

shows the steam production as a function of time when a step is applied on the primary air. Here, the same deviation in the response from the physical model in the second part of the response is found as also is found with the step on the waste inlet flow. A further remark has to be made about the maximum of the response. This is about half of the maximum found in the $T_{prim} = 70^\circ$C situation. In the next section more attention will be given to this difference.

However, in spite of the less good similarity compared to the lower primary air temperature, both models show that the models found are a good indication of the dynamics of a MSWC. Figure 3.14 shows the oxygen concentration as a function of time when a step is applied on the primary air. The experimental obtained curves have the same shape as found by Behrendt [44], when he applied step response experiments in the Tamara pilot plant. Furthermore, both the step responses of the primary air and the waste flow have shapes which are similar to the ones obtained by Becker [4]. Becker performed open-loop process identification at a MSWC plant. However, in their presented results it was not possible to detect any inverse response. As their experiments were carried out in open-loop, it is difficult to obtain accurate results, due to the large disturbances acting on the process. As can be seen in figure 3.11 the inverse response is not quite significant, so probably together with the open-loop experiments this is
Figure 3.13: Steam production response from a step applied on the primary air. Comparison of the estimated black box model (— — —) with the first principle dynamic model (—); $T_{\text{prim}} = 120^\circ\text{C}$.

Figure 3.14: Oxygen concentration response from a step applied on the primary air. Comparison of the estimated black box model (— — —) with the first principle dynamic model (—); $T_{\text{prim}} = 120^\circ\text{C}$.

the reason that the inverse response was not found in the experiments of Becker.

3.5.3 Discussion results from both primary air temperature situations

In this section the results from both primary air temperature situations will be discussed. The difference found in the steam production after applying a step on the
waste flow has already been explained by the evaporation of water. Another main difference can be found in the primary air step responses. The maximum of the response, is about twice higher in the situation with $T_{\text{prim}} = 70^\circ C$ than in the situation with $T_{\text{prim}} = 120^\circ C$, which will be explained now. In section 3.5.1 it has already been mentioned that for the circumstances in a MSWC plant, the mass transfer coefficient is about proportional to the amount of supplied primary air. Besides that the $k_d$ is dependent on the temperature and local circumstances (porosity).

As already mentioned in section 3.5.1 the Sherwood relation were not used in the fitting procedure. Fitting of the model to the system identification results, results in the value of the products $k_d\alpha/\nu_{O2}$ and $k_{av,\text{O2}}$. Infinite different combinations of parameters would yield mathematical correct fits. Fortunately, it was possible to find some physical acceptable set of parameters.

It was found that $k_{d,70} < k_{d,120}$. This can be explained by the circumstances of the experiments. In the previous sections the circumstances of the different experiments have been described. In the situation of the lower primary air temperature, the waste was fresh and less dense. In the situation with $T_{\text{prim}} = 120^\circ C$ the mass transfer coefficient is higher. This implies that in situation with a higher primary air temperature, the mass transfer is less limited compared to the situation with a lower primary air temperature. This is mainly caused by differences in the bed temperature. The simulations have shown that the bed temperature in the situation with $T_{\text{prim}} = 120^\circ C$ was higher compared to the situation with $T_{\text{prim}} = 70^\circ C$, which means a higher mass transfer coefficient.

In conclusion, more primary air leads to a higher mass transfer coefficient $k_d$ and due to the higher temperature in the situation with $T_{\text{prim}} = 120^\circ C$, the influence of $k_d$ on the reaction rate is higher, compared to the situation with $T_{\text{prim}} = 70^\circ C$. This is the mechanism behind the higher maximum found in the response of the steam production on a step on the primary air.

More system identification data are needed to determine exactly the mass transfer parameters. That information, together with the validated model can be used for design and optimisation purposes.

### 3.6 Conclusions

In this chapter a new general model for the dynamics of thermal processes has been derived, which has been applied to MSWC plants. Balances for fuels comprising one solid material (combustible) as well as balances for fuels comprising two solid materials (moisture and combustible) have been given. This general model has been applied to MSWC, which has yielded a completely new dynamic model description of the MSWC process. As no information for validation of processes like MSWC is available in literature, a new validation method has been presented. The method has been applied to the MSWC plant HVC in Alkmaar, The Netherlands. Two situations have been discussed, one with a regular primary air temperature (120 $^\circ C$), and one with a lower primary air temperature (70 $^\circ C$). It has been found that the developed model describes the observed phenomena in the MSWC plant in a rather satisfying
manner. Furthermore, it has been shown that the reaction rate can be described by a classical mass transfer relation, in other words the reaction rate is mass transfer limited.

The validated dynamic model can be used for several applications like the development of new control concepts and the investigation into the influence of disturbances on the process. This application will be the subject of the next chapter.
Chapter 4

Stochastic characteristics of municipal solid waste

One of the objectives of this thesis is studying the influence of the irregularity of the fuel on the behaviour of thermal processes. In chapter 2 it was found that the disturbances acting on a MSWC installation have a stochastic nature. As it is important to learn more about the disturbances acting on the process, stochastic concepts can be applied to the data obtained from the on-line calorific value sensor. This new application of stochastic concepts in the field of MSWC will result in a better understanding of the influence of disturbances on the process.

In this chapter, first a short introduction into the theory of random (stochastic) processes will be given. Following that, the theory of stochastic processes, which is an outgrowth of probability theory, will be applied to the data obtained with the on-line calorific value sensor. It is shown that this new application of stochastic concepts in the field of MSWC will yield interesting new results.

Furthermore in chapter 3 it has been shown that the MSWC process can be described by a relatively simple non-linear dynamic model. It would be very useful if this dynamic model could be described by a linear transfer function model. This has the advantage that the theory for linear systems and the theory describing the transfer of disturbances by linear systems can be applied to MSWC plants. In section 4.3 it will be shown that the dynamic model from chapter 3 can be described by a linear transfer function model. Together with the stochastic properties of waste, this transfer function model will be used to study the implications of disturbances on the MSWC process in section 4.4.

4.1 Introduction to the theory of random processes

In this section, the theory which can be used to describe random signals is discussed. For a more rigorous introduction to the theory of random signals the interested reader is referred to the literature [45], [46], [47], and [48].

Disturbances acting on processes will be often random, which is the contrary of de-
terministic disturbances. Roughly speaking, a random variable is characterised by the fact that instead of knowing its precise value, we only know how to obtain various values of the random variable under certain experimental conditions which are fixed, stable and physically well-defined [47]. Random processes can be categorised as being stationary or non-stationary. To explain this in more detail the time history representing a random phenomenon has to be considered. One single time history of a random phenomenon is called a sample function. The collection of all possible sample functions, which the process might have produced, is called a random or stochastic process [48]. In case of one very long time record, this can be divided into a great number of sample functions. The collection of sample functions is often called an ensemble. From this ensemble, statistical characteristics can easily be determined. Suppose that the random process comprises $N$ sample functions of time length $t$. The random process is then denoted by $\{x_i(t)\}$, where the symbol $\{\}$ is used to denote the ensemble of $N$ sample functions. The mean value $\mu_x(t_1)$ and the autocorrelation function $R_x(t_1, t_1 + \tau)$ are given by

$$\mu_x(t_1) = \frac{1}{N} \sum x_k(t_1) \quad (4.1)$$

$$R_x(t_1, t_1 + \tau) = \frac{1}{N} \sum x_k(t_1)x_k(t_1 + \tau) \quad (4.2)$$

A process is said to be stationary if $\mu_x(t_1)$ and $R_x(t_1, t_1 + \tau)$ are independent of time $t_1$. On the other hand, if they are dependent of time $t_1$ the process is said to be non-stationary. A further refinement of stationarity is the ergodicity. A process is said to be ergodic if the mean and autocorrelation function do not differ when computed over different sample functions. Note that only stationary processes can be ergodic.

### 4.1.1 Basic properties of random data

Four main types of statistical functions are used to describe the basic properties of random data: probability density functions, moment values, autocorrelation functions and power spectral density functions. These will be outlined now. A simple first order probability density function at time $t_0$ is so defined that

$$\text{Prob}(x \leq x(t_0) \leq x + dx) = p(x, t_0)dx \quad (4.3)$$

and from the definition of the probability density function:

$$\int_{-\infty}^{\infty} p(x, t_0)dx = 1 \quad (4.4)$$

For the sake of clearness in the following formulas the time $t$ is omitted unless needed for clarity.

If the probability distribution function is known, some statistics of the random process can be calculated. The most are related to the moments of the variable. The $n$-order moment is defined as follows:

$$M_n = \int_{-\infty}^{\infty} x^n p(x)dx \quad (4.5)$$
The most important are the mean value, the mean square value, and the variance $\sigma^2$:

$$\text{mean value} = M_1 = E[x] = \int_{-\infty}^{\infty} x p(x) \, dx$$

$$\text{mean square value} = M_2 = E[x^2] = \int_{-\infty}^{\infty} x^2 p(x) \, dx$$

$$\sigma^2 = E[x^2] - (E[x])^2$$

One of the most well-known probability density function is the Gaussian distribution which has the following form [47]:

$$p(x) = \frac{1}{\sqrt{2\pi \sigma}} e^{-\frac{(x-m)^2}{2\sigma^2}}$$

Here $m$ is the mean of the process and $\sigma$ the standard deviation. Many in nature occurring random vibration processes can be described with this probability density function. A very important property of the Gaussian or normal process is its behaviour with respect to linear systems. When the input of a linear system is normal, the response (output) will be a very different random process in general, but it will still be normal distributed. The form of the probability density function is thus invariant with respect to linear operations.

The fact that many random processes in nature will be Gaussian is explained by the central limit theorem from the probability theory. Roughly speaking, the central limit theorem says that when a random process results from the summation of many infinite random events than this process will tend to have a Gaussian probability density function [46].

The aforementioned introduced values are calculated across the ensemble at time $t = t_1$. At every time $t$ these values can be calculated. As already mentioned, the important autocorrelation function can be calculated by using the second order probability function:

$$R[x_1(t_1)|x_2(t_2)] = \int_{-\infty}^{\infty} \int_{-\infty}^{\infty} (x_1 - \mu_1)(x_2 - \mu_2) p(x_1, t_1, x_2, t_2) \, dx_1 \, dx_2$$

(4.10)

The autocorrelation function can be calculated at different time intervals $\tau = t_1 - t_2$. From the definition of stationarity it follows that the autocorrelation function is a function of only $\tau$:

$$R[x(t)|x(t+\tau)] = R(\tau)$$

(4.11)

Note that in a process where $x$ has zero mean, $R(0) = \sigma^2$.

A frequency decomposition of the autocorrelation can be made in the following way:

$$R(\tau) = \int_{-\infty}^{\infty} S(\omega) e^{i\omega \tau} \, d\omega$$

(4.12)

where $S(\omega)$ is essentially the Fourier transform of $R(\tau)$:

$$S(\omega) = \frac{1}{2\pi} \int_{-\infty}^{\infty} R(\tau) e^{-i\omega \tau} \, d\tau$$

(4.13)
where \( S(\omega) \) is called the spectral density. An important property of \( S(\omega) \) becomes apparent when \( \tau = 0 \). Together with the definition of the autocorrelation function (4.12) for the mean square value of a stationary random process the following relation is found:

\[
E[x^2] = R(0) = \int_{-\infty}^{\infty} S(\omega) d\omega
\]  

(4.14)

The mean square value of a stationary random process is therefore given by the area under a graph of the spectral density \( S(\omega) \). The often in analytical investigations used white noise signal has a uniform spectral density \( S_0 \) for all frequencies.

In conclusion, to describe a random process it is essential to reveal the stationarity, the probability distribution, and the power spectral density. In the next sections an attempt is made to describe the signals from the on-line calorific value sensor within the aforementioned theory.

### 4.1.2 Transmission of disturbances in dynamic processes

When the stochastic properties of a random variable have been described one wants to use these to study the influence of the disturbances on the output variables of the process. The disturbances can be described by probability distributions, but how this probability distribution of an input signal is transferred in a dynamic system is the point of major interest. Even for a simple linear system the solution is not quite easy: there is no general method for obtaining the output probability distribution for a linear system, except for the special case that the probability functions of the input signals are Gaussian [46]. In that case the probability density function of the output signal turns out to be Gaussian too. As many processes in nature turn out to be Gaussian this result is very useful for engineering applications.

So, if the system is linear and the probability distribution of the input signal is Gaussian, the output distribution is Gaussian too. Then, the standard deviation of the output signal can easily be calculated by using the general notation \( H(\omega) \) for transfer functions of process models which connect the output (response) signals with the input (excitation) signals:

\[
\sigma_y^2 = \int_{-\infty}^{\infty} S_x(\omega)|H(\omega)|^2 d\omega = 2 \int_{0}^{\infty} S_x(\omega)|H(\omega)|^2 d\omega
\]  

(4.15)

However, if the process under study is not linear, the transmission of signals can be studied by the theory of Markov processes. Markov processes are a special class of random processes. Simply said: a Markov process is a process without memory effect, which implies that the next \((t+\delta)\) random occurrence only depends on the current state \((t)\). Although not all actual processes are exactly Markovian it sometimes is a good approximation to regard them as being Markovian. This makes it possible to obtain many concrete results by using the effective mathematical methods of Markov [47] and [49]. A special class of Markov processes can be described by the Fokker-Planck equation, which can be interpreted as the equation of probability conservation.

So, given an external disturbance and a (non-)linear process, it is possible to derive an
associated Fokker-Planck Equation (FPE) which describes the probability distribution as a function of time. For first order systems the FPE can easily be derived and the found diffusion-like equation can often be solved (semi-)analytically. For second order systems it becomes more difficult to find a (semi-)analytical solution for the FPE equation. The dynamic model presented in chapter 3 is second order in the gas phase temperature and even third order in the steam production. Therefore, for this moment no attempt is made to derive and solve the FPE equation. For more information concerning the FPE equation and stochastic processes the reader is referred to Stratonovich [47].

4.2 Stochastic properties of the calorific value

From the introduction section it was learnt that to characterise the calorific value data it is necessary to investigate three stochastic properties: stationarity, probability distribution and the spectral density. These will be treated in the subsections below. First, stationarity has to be determined, because the analytical procedures for non-stationary data are generally more complicated then those that are appropriate for non-stationary data. Secondly, the presence of periodicities in the data has to be investigated and then the probability functions can be determined. The reason for checking periodicities is that the presence of periodicities leads also to other analytical tools. Periodicities will be discussed further in section 4.2.3, where the spectral density section of the calorific value will be discussed.

4.2.1 Stationarity

First the stationarity of the process data will be investigated. As discussed in the beginning of section 4.1 a process is said to be stationary if the ensemble average \( \mu_x(t_1) \) and ensemble autocorrelation function \( R_x(t_1, t_1 + \tau) \) are independent of time \( t_1 \). There are several tests for testing stationarity. One of the methods [48] is based upon the fact that in practice it is highly unlikely that non-stationary data have a time varying autocorrelation function at any time displacement \( \tau \) without the value at \( \tau = 0 \) varying. From the definition of the autocorrelation function it follows that \( R(0) = \sigma_x^2 \); in other words \( R(0) \) equals the mean square value. Therefore stationarity is investigated with the following procedure:

- Divide the sample record in \( N \) equal time intervals \( x_i(t) \) where the data in each interval may be considered independent;
- Compute a mean square value for each interval;
- Test the sequence of mean square values for the presence of underlying trends, other than those due to expected sampling variations;
- Use the non-parametric reverse arrangement test, to test the stationarity.

In Appendix C the method is outlined in more detail and stationarity analyses are given for some artificially produced data, in order to achieve some feeling for the
Figure 4.1: Stationarity value of the mean value and standard deviation of the z-value as a function of different time frames for the MSWC AZN.

Figure 4.2: Stationarity value of the mean value and standard deviation of the z-value as a function of different time frames for the MSWC Essent.
concept of stationarity. There it can be seen that even for white noise, which should be perfect stationary, thus a stationarity value of 1, the stationarity value is only 0.8. Therefore, it is concluded that if the stationarity test gives a value of about 0.8 the process can be considered as stationary. In Appendix C it is shown that significant trends in the mean value of the standard deviation leads to very low (< 0.4) stationarity values.

In chapter 2 the results from the on-line caloric value sensor in four different MSWC installations were discussed. These data are used for the stochastic analysis, which will be discussed now. Figure 4.1 up till 4.3 show the stationarity value of the mean value and the variance as function of the used time-interval for the caloric value, the z-value (or O/C ratio, see equation (2.6)) and the H₂O-fraction for the MSWC plants AZN, Essent, and HVC. The figures for all three installations show that the stationarity ranges from 0.6 to 0.8. Referring to the figure concerning the MSWC Essent it was mentioned in chapter 2 that the moisture fraction in the waste slowly increased and the caloric value slowly decreased during the experimental period. These trends are not visible in the stationarity values. As white noise has values from 0.8 and higher, the processes are not stationary in a strict sense. However, as also the time interval has almost no influence on the results, it is acceptable to consider the processes as being stationary.
4.2.2 Probability distribution

Having shown within reasonable limits that the calorific value, the z-value and the H₂O-fraction can be considered as stationary random data, the next step is to investigate the probability density functions. First, in figure 4.4 the probability distributions of the calorific value for the four MSWC plants are presented. It can be seen that the distributions of HVC and Essent are fairly Gaussian, while the distributions of AZN and Twence are absolutely not Gaussian. To investigate this phenomenon in more detail the distributions for the z-value and the H₂O-fraction are depicted in figure 4.5 and 4.6. From the figures it can be seen that for Essent and HVC the water and z-distribution are almost perfect Gaussian. These distributions for the z-value and the H₂O-fraction resembles the theoretical Gaussian distribution even better than the distributions of the calorific value. The H₂O-distributions for MSWC AZN and

![Figure 4.4: Probability distribution for the calorific value (MJ kg⁻¹) compared with a Gaussian distribution for the installations AZN, Essent, HVC and Twence.](image)

Twence have both a higher maximum compared to the best fitted Gaussian distribution. However, the distribution of the z-value for both installations shows a tendency to have a higher peak shifted to higher z-values. Trying to explain this difference let us first explain the Gaussian property, which can be understood with the central limit theorem. If it is assumed that waste, which is collected from individual waste producers, is a random collection concerning the density and the calorific value, a great number of waste entities are collected in a truck, while a lot of trucks will fill the bunker. As in the bunker the waste is mixed as much as possible, from the central limit theorem it follows that the distribution of the calorific value and the density of the waste shall be Gaussian. This means that the probability distribution of these two quantities is Gaussian by the entering of the waste into the furnace. However, the calorific value sensor determines the calorific value of the waste that is converted at the grate. This means that it becomes a kind of average of the individual waste inputs. As this is a linear operation the probability distribution will remain Gaussian. As it was found that the probability functions for the MSWC Essent and HVC
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Figure 4.5: Probability distribution for the z-value compared with a Gaussian distribution for the installations AZN, Essent, HVC and Twence.

Figure 4.6: Probability distribution for the H₂O-fraction in the waste compared with a Gaussian distribution for the installations AZN, Essent, HVC and Twence.

are Gaussian, the transformation process has to be linear. As the AZN installation is of the same type as the HVC installation one would expect that the transformation process in the HVC-installation is also linear. Then the only explanation for the non-Gaussian property in the AZN case is that the stochastic properties of the waste, entering the furnace, are not Gaussian. One would expect that if the waste at the Essent and HVC installations is Gaussian at the entrance of the furnace, that it would also be Gaussian in the other installations (i.e. AZN and Twence). Based on that hypothesis the only explanation for the non-Gaussianity is found in the mixing process in the bunker. However, this explanation has to be verified in the future by
additional experiments. More attention to the transformation from the calorific value at the entrance to the calorific value at the grate will be given in subsection 4.2.3, which deals about the spectral density.

It was observed that at each installation during different time periods the same distribution of the z-value, H₂O-fraction and calorific value was found. This justifies the conclusion of stationarity, but it indicates also that the random process is ergodic. In table 4.1, the values for mean value and the standard deviation for the z-value, the H₂O-fraction in the waste and the calorific value are given. In the MSWC plants,

<table>
<thead>
<tr>
<th>MSWC</th>
<th>$z$</th>
<th>$X_{H₂O}$</th>
<th>$\Delta H$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$\mu$</td>
<td>$\sigma$</td>
<td>$\mu$</td>
</tr>
<tr>
<td>AZN</td>
<td>0.392</td>
<td>0.135 (34%)</td>
<td>0.320</td>
</tr>
<tr>
<td>Essent</td>
<td>0.179</td>
<td>0.158 (88%)</td>
<td>0.426</td>
</tr>
<tr>
<td>HVC</td>
<td>0.445</td>
<td>0.155 (35%)</td>
<td>0.321</td>
</tr>
<tr>
<td>Twence</td>
<td>0.422</td>
<td>0.159 (38%)</td>
<td>0.371</td>
</tr>
</tbody>
</table>

Table 4.1: Mean value ($\mu$) and standard deviation ($\sigma$) of z-value (mol mol⁻¹), H₂O-fraction (kg kg⁻¹) and calorific value (MJ kg⁻¹) for the MSWC plants AZN, Essent, HVC and Twence.

that process integral household waste (AZN, HVC, and Twence), the mean values of the z-values are in the same order of magnitude. The same holds for the standard deviation of the z-value. The mean value of the water content ranges for this installations from about 0.32 to 0.37 kg kg⁻¹. Remarkable is the relatively high standard deviation in the calorific value of the MSWC Twence compared to HVC and AZN.

At the moment there is no explanation available for this.

For the MSWC Essent the mean value of the O/C ratio (z-value) is lower and the mean value of the water content is slightly higher compared to the other MSWC plants. This is caused by the pre-treatment of the waste. By particle reduction and sieves the waste is sorted, which yields a separate low calorific fraction called the organic wet fraction. This separation decreases the z-value of the remaining waste, as it will contain relatively more plastic. As a result of this presorting the standard deviation of the z-value is relatively high as the absolute standard deviation of the z-value is about the same for the four MSWC plants. Furthermore, it is concluded that the presorting has an increasing effect on the water content. However, at present MSWC Essent adapted the pre-treatment slightly, by implementing a paper/plastics suction system. Based upon that it is expected that the current O/C ratio in the waste after the pre-treatment will be in the same range as the other MSWC plants.

Considering more in detail the probability functions of the z-value (see figure 4.5, it can be seen that the z-value can become negative. This seems to be incorrect. It can not be explained by systematic errors in the assumptions (unburnt fraction $\gamma_C$ or H/C ratio $y$), which form the basis of the calorific value sensor. However, a more detailed analysis leads to the conclusion that a negative z indicates that there are too many C-atoms available in the gas phase. This is the result from the fact that fixed carbon, which was accumulated in the solid waste layer, is converted due to process changes.
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In other words the carbon conversion changes temporarily. The present model for the calorific value sensor does not contain any dynamic behaviour, as the model is based upon a stationary gas phase mass balance. Fortunately this accumulating effect of the waste layer does not affect the results of the sensor, as the sensor is based upon Michel’s formula, which makes no difference in the origin of C-atoms. Furthermore, also for negative z values Michel’s formula is a continuous function.

Conclusion: The disturbances acting on the process are Gaussian in MJ kg\(^{-1}\) stationary and ergodic in the MSWC plants HVC and Essent. This is not the case for the MSWC AZN and Twence, probably caused by the mixing process in the bunker.

4.2.3 Spectral density

As the disturbances are stationary and Gaussian the last unknown characteristic is the frequency spectrum. Figures 4.7 and 4.8 show the measured spectral density of the calorific value at the four different installations. It has to be noticed that the experiments have been carried out during four different periods in the year. In the figures it can be seen that the power of the calorific value is decreasing with increasing frequency, with a more or less constant beginning. It has to be noticed that despite the long measurement periods (up to 90 days) the number of data representing the low frequencies (\(\leq 3 \times 10^{-5}\) rad s\(^{-1}\), which is about 2.4 days) is limited and consequently the accuracy of the spectrum for lower frequencies is limited. If the spectrum would be equal to a band limited white noise, the spectrum would be constant till a certain frequency \(f_{\text{band}}\), while above this frequency the power density would be zero. In the spectra from the MSWC installations the power decreases with the frequency, but in all four figures it is noticed that at certain frequencies the power shows positive peaks. The observed very sharp peaks are caused by periodicitics in the further stochastic signal [48]. The other, more smooth, peaks are indicated in the figure. The frequen-
cies of these peaks are of such a level that they are most probably caused by human or process interactions. These can be for example the actions of the crane driver, who is responsible for mixing the waste and filling the chute or the ram feeder.

The frequency spectra for the different installations are not the same. As already mentioned the low frequency behaviour is not very accurate, which can result in an overestimation of the power at low frequencies. Another explanation could be the following: the intensity of the fluctuations in the waste composition can result in a higher magnitude in the lower frequencies of the frequency spectrum, because a high fluctuating composition causes a high standard deviation, which is, according to formula (4.15), the double square root of the integral of the frequency spectrum.

The general question is how to describe the frequency spectra in a simple way. First, it has to be noticed that the calorific value is measured during the conversion of waste into gas. Therefore, to understand the system better, it is important to know how the waste layer influences the stochastic properties. Suppose that the calorific value at the entrance has a band-limited white noise spectrum with a Gaussian distribution. The question is how this spectrum is affected by the process on the waste layer. If the processes on the grate could be represented with a first order system with a time constant \( t \) lower than the time constant of the band limited white noise, then, according to the in section 4.1.2 described theory for transferring random data by linear systems, the frequency spectrum at the outlet would have the same shape as the linear first order transfer function. And as the first order system is linear and the input is a Gaussian signal then the output signal would also be Gaussian. Conversely, as the output signal is Gaussian, the input signal should be Gaussian too and the transformation will be linear. However, the frequency spectra at the outlet (figure 4.7 and 4.8) are absolutely not
a simple first order system or even a simple second order system (see for example [50] for Bode plots for classical (mass-spring system) second order transfer functions.) Especially the peaks in the higher frequencies make it almost impossible to describe the transfer process easily, because of the non-random properties (periodicities) of these sharp peaks and the possible by human operation caused smooth peaks. In chapter 5 it will be shown that the processes occurring on a grate in a MSWC plant are rather complex and it is not easy to deduce transformation rules for the calorific value entering the combustion system.

The analysis can be summarised as follows: the disturbances of the calorific value are stationary and ergodic and for MSWC HVC and Essent their probability density function can be considered as Gaussian. At this moment it is not possible to describe easily the exact frequency distribution of the waste. To study the influences of disturbances acting on MSWC plants, in the next sections an estimation of the frequency spectrum will be used. Future research must reveal more information about the frequency distribution.

4.3 Linear transfer functions for thermal processes

In section 3.2 a set of equations (3.12)-(3.15) and (3.24)-(3.25) was presented, which can be used to describe the main dynamics of a thermal process, where the fuel is considered to include one component. Such equations can easily be solved numerically. Due to the non-linearities, analytical solutions seem not possible. But, important transfer functions can be derived by linearising the equations. With linearised models the influence of disturbances acting on these processes can more easily be studied. For the application to MSWC plants this will be illustrated and discussed in more detail in chapter 4.4.

In this section, for the sake of generality, the linearised models will be presented, without discussing it in much detail. In the set of equations the steam production, oxygen concentration and gas temperature $T_g$ are the main observables. The aim is to investigate the influence of the main input variables on main output variables. However, as already mentioned in section 3.2, the steam production changes the mathematical system from a second to a third order system, which makes the semi-analytical solving of the problem more complex. For a semi-analytical treatment of the dynamics it is sufficient to consider the gas phase temperature as main observable output parameter, since the steam system has no influence on the gas phase temperature. The main input variables of the system are

- the amount of solid material that enters the system and that will be converted;
- the amount of primary and secondary air;
- the residence time of the solid material in the system.

The latter follows from the clear fact that the residence time has an obvious influence on the total mass of solid material in the system.
To investigate the influence of the main input variables on the main output variables, it is necessary to find the following transfer functions: \( \delta T_2/\delta \Phi_{in}, \delta T_3/\delta \Phi_{prim} \) and \( \delta T_4/\delta \Delta H \). The set of four equations (3.12)-(3.15) can be linearised, and after reducing it was found that all transfer functions have the following general form:

\[
\frac{bs_2 \tau^2 + bs_1 \tau + bs_0}{s_0 (\tau^2 \tau^2 + 2 \zeta \tau + 1)}
\]

(4.16)

This is a second order transfer system. When \( ts_2 = t_s = 0 \) the classical mass-spring system is obtained. After some tedious calculations, the natural period \( \tau \) and the damping factor \( \zeta \) are found:

\[
\tau = \sqrt{\frac{C_p \beta M}{s_0}}
\]

(4.17)

\[
\zeta = \frac{1}{2} \left( \frac{\beta (\theta + (1 - X_{in}) \Phi_{in} C_{p1} + A \alpha \varepsilon_g)}{\sqrt{C_p M \beta s_0}} \right)
\]

(4.18)

In a pure mass-spring system \( \zeta \) is a measure of the overshoot when a step function is applied at the input [46]. \( 0 < \zeta < 1 \) means that there is an overshoot (underdamped) in the transient response, while \( \zeta = 1 \) (critical damped) means that there is no overshoot and \( \zeta > 1 \) means an overdamped system. In fact, \( \zeta > 1 \) means that the second order system can be described by two serial first order processes.

The unknown value \( s_0 \) is given by the following expression:

\[
s_0 = (1 - X_{inert}) \Phi_{in} (\beta (C_{p1} \Phi_{in} + C_{p12} \Phi_{prim} + A \alpha \varepsilon_g))
\]

\[
- \alpha \varepsilon_g (1 + RE M \gamma / R)
\]

(4.19)

With \( \theta, \beta, \gamma \) defined as follows:

\[
\theta = C_{p1} \Phi_{in} + C_{p12} \Phi_{prim} - (1 - X_{inert}) \Phi_{in} \frac{RE}{R X_{\Delta H} \Delta H} + A \alpha \varepsilon_g
\]

\[
\beta = \frac{(\Phi_{prim} + \Phi_{sec}) C_{p12} + A \alpha \varepsilon_g + C_{p1} (1 - X_{inert}) \Phi_{in}}{\kappa}
\]

\[
\gamma = \frac{(1 - X_{inert}) \Phi_{in} (C_{p1} (T_x - T) - (1 - X_{\Delta H}) \Delta H)}{\kappa}
\]

and the denominator of the last two equations:

\[
\kappa = \Phi_{prim} C_{p1} + A \alpha \varepsilon_g + (1 - X_{inert}) \Phi_{in} ((1 - X_{\Delta H}) \Delta H RE / R + C_{p1} (1 + RE / R (T - T_x)))
\]

In the above equations \( RE \) is the temperature derivative of the reaction rate: \( RE = (\delta R/\delta T)|_{T - T_0} \). When the reaction rate is described with the first term of equation (3.5), the temperature dependence of the mass transfer coefficient \( k_d \) has to be discussed, before differentiating. The temperature dependency can easily be investigated when the mass transfer coefficient is determined with a general Sherwood relation (equation (3.10)). Working out the relation, by substituting the Reynolds and Schmidt number, leads to the following relation for the mass transfer coefficient:

\[
k_d = C \frac{\Phi_{d} d_p}{A} \frac{D_{\Phi}^2}{\eta d_p}
\]
where $\Phi$ is the amount of supplied air in $kg\cdot s^{-1}$ and $A$ the cross-sectional area with dimension $m^2$. Only the last fraction contains temperature dependency. However, it can be proven that for normal temperatures in solid fuel conversion systems (700-1200 $^\circ C$) the mass transfer coefficient is almost linear with temperature. For the linear analysis, this is a very important conclusion. The error made with such a linear assumption is less than 5%. Therefore at this stage $k_d$ is supposed to be $k_d = p + qT$, where $p$ and $q$ are to be determined coefficients from the Sherwood relation. Now, the temperature derivative from the reaction rate can be determined, which leads to the following relation for $RE = (\delta R/\delta T)|_{T=T_0}$:

$$RE = \frac{\delta R}{\delta T} = R^2 \left( \frac{q}{k_d^2} + \frac{E/RT^2}{k_0 e^{-\frac{E}{RT}} \tau} \right)$$

All variables are in the stationary state. For the sake of clearness the subscripts ‘O’ for the stationary state are omitted. The stationary values for the temperatures $T_g$ and $T_0$ can easily be found by solving the four equations (3.12)-(3.15) with $d/dt = 0$. Linearising, reducing and normalising for the different input-output relations result in the following transfer functions:

**Transfer function $\delta T_g/\delta \Phi_{in}$**

$$\frac{\delta T_g}{\delta \Phi_{in}} = \frac{ts_1 s + ts_0}{s_0(\tau^2 s^2 + 2 \zeta \tau s + 1)}$$

(4.20)

where the terms in the numerator are:

$$ts_1 = -C_{p,0}(1 - X_{inert})\gamma M + C_{p,0}(T - T_{in})$$

$$ts_0 = (-\gamma \theta + Ra(X_{\Delta H} \Delta H + C_{p,0}(T_{in} - T)(1 + RE/R\gamma M))(1 - X_{inert})$$

So, the gas temperature reacts on a change in the amount of fuel as a second order process with a differential action. The stationary value of this transfer function is $ts_0/s_0$ which is independent of the amount of mass in the system. This transfer function is not in accordance with the results obtained by Kitami [24]. Kitami performed step response tests in a MSWC. The result was that the transfer function from the step on the waste input to the furnace temperature could be described by a first-order system with a pure delay time. The above derived transfer function seems to be more general, as the proposed second order system can easily represent a slow start, which suggests the occurrence of a time delay. In other words a second order system can in some cases be described by a combination of a pure time delay and a first order system.

But there is some doubt concerning the use of a pure time delay in these systems. A time delay generates a discontinuity, which from a physical point of view is less realistic compared to the proposed second order transfer function.

**Transfer function $\delta T_g/\delta \Phi_{prim}$**

Regarding the transfer function $\delta T_g/\delta \Phi_{prim}$ attention has to be paid to the influence of the primary air on the reaction rate. Equation (3.15) shows that the reaction rate
may be dependent on the mass transfer coefficient $k_d$. For this coefficient the following Sherwood relation [51] can be used:

$$Sh = \frac{k_d l_p}{D} = Re \# Sc \#$$

(4.21)

This implies that $k_d$ is linear in the square root of the air velocity and thus the amount of primary air. Carrying out the same analysis results in the transfer function:

$$\frac{\delta T_g}{\delta \Phi_{prim}} = \frac{t_s g s^2 + t_s t s + t_s 0}{s_0 (s^2 + 2 \zeta \tau s + 1)}$$

(4.22)

with the unknown variables:

$$t_s g = -C_p M \eta$$

$$t_s t = (1 - X_{inert}) \Phi_m (1 + RE / R \gamma) - \nu C_p \gamma$$

$$t_s 0 = \frac{1}{M} \left( (1 - X_{inert}) \Phi_m (1 + RE / R \gamma) - \nu C_p \gamma \right)$$

$$\lambda = \left( \frac{C_p (T_a - T)}{M} - \frac{(1 - X_{inert}) \Phi_m \Delta H}{X_{inert} \Delta H} \right) \frac{T_a - T}{X_{inert} \Delta H} \frac{\nu \Delta H}{X_{inert} \Delta H}$$

$$\eta = \left( \frac{\omega}{1 - X_{inert}} \frac{\Phi_m \Delta H}{X_{inert} \Delta H} \frac{T_a - T}{X_{inert} \Delta H} \right) \frac{\nu \Delta H}{X_{inert} \Delta H}$$

$$\omega = \frac{1}{2} \frac{C_p M \eta}{C_p M \eta + \kappa X_{inert} \Delta H}$$

Transfer function $\delta T_g / \delta \Delta H$

The important relation between the gas phase temperature and the calorific value of the converting solid fuel has the following form:

$$\frac{\delta T_g}{\delta \Delta H} = \mu \frac{t_s g s^2 + t_s t s + t_s 0}{s_0 (s^2 + 2 \zeta \tau s + 1)}$$

(4.23)

with the coefficients:

$$t_s g = C_p M$$

$$t_s t = (1 - X_{inert}) \Phi_m \eta + \frac{\kappa X_{inert} \Delta H}{(1 - X_{inert}) \Phi_m}$$

$$t_s 0 = \frac{(1 - X_{inert}) \Phi_m \Delta H}{M} \frac{RE / R (1 - X_{inert}) \Phi_m}{(1 - X_{inert}) \Delta H}$$

$$\theta = \frac{\nu \Delta H}{X_{inert} \Delta H} \frac{\Phi_m \Delta H}{X_{inert} \Delta H} \frac{\nu \Delta H}{X_{inert} \Delta H}$$

Also in this case it can be seen that the stationary value $\mu t_s 0 / s_0$ is independent of the total mass $M$ in the system. The derived transfer functions will further be used in the next section.
4.3.1 Determining linearity of the system

The linearisation, described in section 4.3 is checked by comparing the numerical model solution with the semi-analytical solutions for the HVC case, presented in section 3.5. There it was shown that the situation with a primary air temperature of 120 °C is more complex (one extra differential equation) than the situation with a primary air temperature of 70 °C. The above discussed linear model can therefore only be applied to the case with the lower primary air temperature. Figure 4.9 shows a comparison of the numerical and the linear analytical model for steps on the waste flow and the calorific value. The values from table 3.1 have been used in the calculations. It can be seen that the agreement is very well. A remark has to be made about the shape of the response. In section 4.3 is shown that the linearised transfer function is second order in the denominator as well as in the numerator. In a simple second order system (mass-spring), overshoot in response occurs when $\zeta < 1$. In the present case it turned out that $\zeta \approx 0.6$, which indicates that the numerator is responsible for the overshoot.

So, with the use of equation (4.15) and the linearised transfer functions, the disturbances in the output signal can easily be calculated and compared with numerical calculations, when a white noise signal is used as the input signal for the calorific value. First the frequency spectrum of the gas phase temperature for the numerical and analytical method are compared in figure 4.10. In the figure it can be seen that the frequency spectra have a very good similarity. Calculations show that the ratio of the standard deviation of the gas phase temperature ($\sigma T_g$) and the standard deviation of the calorific value ($\sigma_{\Delta H}$) in the linear case is $6.49 \times 10^{-5}$ (K kg J$^{-1}$), while the non-linear (numerical) calculations give a value of $6.57 \times 10^{-5}$ (K kg J$^{-1}$). So, the difference between the linear and non-linear model is less than 1.5%, which also justifies the linearisation.

![Figure 4.9: Comparison analytical and numerical step response. Gas phase temperature as a function of time when a step on the waste flow and a step on the calorific value is applied.](image-url)
4.4 Implications of stochastic disturbances on MSWC processes

In section 4.2 it has been shown that for two plants the disturbances can be described with a Gaussian probability distribution with a certain frequency distribution. In the preceding section it has been demonstrated that the developed model from chapter 3 can be linearised and transfer functions can be derived. So, as it has been proven that the system is linear and the disturbances are ergodic and Gaussian, the classical theory for linear systems can easily be applied. For example peak distributions and extreme responses of output variables can now easily be calculated. In this section the influence of the design parameters on the output variables and the lifetime of superheaters are studied, while only the Gaussian situations will be studied because in section 4.2 it was concluded that the non-Gaussianess of the two other MSWC plants (Twence and AZN) is probably caused by non-random events.

First the Gaussian results from chapter 2 will be discussed from another viewpoint, based on the linearity of the system. Let us consider the probability distribution of the steam production from the HVC plant in figure 4.11. From the figure it follows that the distribution can be considered as Gaussian. In section 4.1.2 it was mentioned that it is almost impossible to predict the probability distribution of the output signal from a linear system, unless the input signal is Gaussian. In these circumstances the output signal will be Gaussian too. Although it is not proven here, this holds also for the reverse analysis: if for a linear system the output signal is Gaussian the input signal will be Gaussian too.

The obtained steam production is the controlled steam production i.e. the steam production can be changed by the control system. However, as the control system is linear, it will not affect the Gaussian characteristics of the steam production. This means that the steam production in the open-loop case will also be distributed Gaussian.

Due to complexity reasons, the linear system analysis has been performed with the gas
phase temperature instead of the steam production. However, the relation between the steam production and the gas phase temperature is a linear transfer function as easily can be seen in formulas (3.24) and (3.25). This indicates that the input signal of the system, i.e. the average z-composition and H2O-fraction on the grate have to be Gaussian. As the transformation process from the entrance to the average waste composition on the grate can be considered as a linear process (see section 4.2.2), extending this analysis leads to the conclusion that the waste entering the furnace must also have a Gaussian distribution in the z-value and the H2O-fraction. Of course besides the distribution, the initial values also have an influence on the response of a linear system. As mentioned before the Fokker-Planck equations can be used to describe the probability distribution as a function of time. In [47], it is shown that for stationary Gaussian processes the initial distribution of the Fokker-Planck equation has no influence on the stationary distribution.

### 4.4.1 Influence disturbances and design parameters on process behaviour

The influence of design parameters on the standard deviation of the main output parameters can be studied using (4.13). The transformation from the Laplace ($s$) to the frequency domain is straightforward by substituting $s$ in the transfer functions by $j\omega$. Here, it is chosen to study the influence of the main disturbances on the standard deviation of the gas phase temperature $T_g$. Therefore the transfer function $\delta T_g/\delta \Delta H$ for the main non-controllable variable $\Delta H$ is used. However, one problem arises, as the stochastic property (i.e. $S_g(\omega)$) of the input variable $\Delta H$ has to be known.

From the stochastic analysis performed in the first sections of this chapter, it is learnt that the frequency spectrum of the calorific value is the input value for the dynamic system. In other words the derived frequency distributions can be used as input values $S_p(\omega)$ for the $\delta T_g/\delta \Delta H$ transfer function. To use the found frequency spectrum in this analysis, the frequency spectrum was fitted into a transfer function. As discussed before it is almost impossible to fit the frequency spectrum exactly, but the estimation gives a good indication of the real spectrum. Here, the theory will be applied to the
HVC situation.

Although for two different primary air temperatures two models were found, which describe the found phenomena reasonably well, for the moment no model for intermediate primary air temperature is available. Besides that, at the moment only for the process at lower primary air temperature \( T_{\text{prim}} = 70^\circ\text{C} \) a complete semi-analytical model is available as presented in section 4.3. Those transfer functions will be used in the analysis. The used parameters can be found in section 3.5.1. The sensitivity of the standard deviation in the gas phase temperature has been calculated by changing the actual parameters by +25% and -25%. The influence of the most important parameters is studied: \( X_{\text{inert}}, \Delta H, X_{\text{air}}, A, \Phi_{\text{in}}, \Phi_{\text{prim}}, \) and \( k_d \). It is obvious that some values are very difficult to manipulate in practice. However, this analysis will give a good impression of the sensitivity of the standard deviation in the gas temperature for changes in the process. The sensitivity of the parameters \( \Phi_{\text{sec}}, T_{\text{prim}}, T_{\text{sec}}, \) and \( T_{\text{in}} \) is not investigated. This is because \( T_{\text{in}} \) has almost no influence on the process. Furthermore, the secondary air temperature has normally values around the ambient temperature, and the influence of changes in the secondary air temperature on the gas phase temperature is negligible, as was already concluded in chapter 3. There it was also shown that the influence of the secondary air flow is limited. The influence of the primary air temperature is high as has been discussed in chapter 3, but the model has not been validated for different primary air temperatures.

<table>
<thead>
<tr>
<th>( X_{\text{inert}} )</th>
<th>( \Delta H )</th>
<th>( X_{\text{air}} )</th>
<th>( A )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \Delta (%) )</td>
<td>+25</td>
<td>-25</td>
<td>+25</td>
</tr>
<tr>
<td>( \Delta \sigma_{T_g} (%) )</td>
<td>0.4</td>
<td>1.3</td>
<td>-9.4</td>
</tr>
<tr>
<td>( A (%) )</td>
<td>-6.6</td>
<td>+10.2</td>
<td>-3.8</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>( \alpha )</th>
<th>( \Phi_{\text{in}} )</th>
<th>( \Phi_{\text{prim}} )</th>
<th>( k_d )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \Delta (%) )</td>
<td>+25</td>
<td>-25</td>
<td>+25</td>
</tr>
<tr>
<td>( \Delta \sigma_{T_g} (%) )</td>
<td>2.0</td>
<td>-23</td>
<td>8.6</td>
</tr>
<tr>
<td>( A (%) )</td>
<td>9.8</td>
<td>0.3</td>
<td>7.5</td>
</tr>
</tbody>
</table>

Table 4.2: Sensitivity of the standard deviation of the gas phase temperature as a function of variations in the main process parameters.

It is chosen to present the results of the sensitivity analysis in two ways. Besides the usual presentation in table 4.2 also frequency plots of the transfer function have been used. In this way it can easily be seen if the system becomes unstable, as then smooth but high peaks will occur in the frequency spectrum. The standard deviation is also presented by using frequency plots. Formula (4.15) says that the integral over all frequencies is the mean square value (i.e. the square of the standard deviation). Therefore, instead of giving only the standard deviation as output, it is chosen to give the frequency plot to see the influence of different frequencies on the standard deviation. The cumulative magnitude of the standard deviation of the gas phase temperature is plotted as a function of the frequency. As the exact value of the standard deviation in the input variables is not known, the influence of the parameters on the output is presented only relatively to the standard situation, where the standard de-
IMPLICATIONS OF STOCHASTIC DISTURBANCES ON MSWC PROCESSES

Figure D.1 up till D.8 in Appendix D show the frequency plots for all eight parameters for the transfer function $\delta T_d/\delta \Delta H$. The frequency ranges from 0 till $5.2 \times 10^{-2}$ rad $s^{-1}$, which equals the Nyquist frequency, which is the reciprocal of twice the used sample time (60 s) in practice. With reference to this appendix and table 4.2 the main results will be discussed. The transfer function is multiplied by the frequency spectrum ($S_x(\omega)$) of the input variable $\Delta H$ as discussed above. Only the parameter $\Delta H$ has a significant influence on the transfer function. Especially the decrease of the absolute value of the stationary value from $\Delta H$ by -25% introduces a resonance peak in the transfer function, causing an increase of the standard deviation of $\sigma T_d$ with about 50%. This means that the control system probably must have other control parameters for the low calorific values, compared to the normal and higher calorific values.

The disturbance rejection capabilities of the system will not become worse by increasing the working point of the calorific value. Of course, increasing the calorific value can cause other problems in the furnace, like crossing the system’s thermal capacity or grate damaging.

Remarkable is that changing process parameters doesn’t have a large influence on the $\sigma T_d$. The conclusion of this analysis is that the system is more or less stable in the working point and further substantial disturbance rejection can not be achieved by manipulating process parameters. So, all disturbance rejection has to be achieved by the control system.

Suppose it would be possible to decrease the fluctuations in the calorific value, by a better mixing process in the bunker. With the derived relation this influence can easily be studied. Decreasing the standard deviation of the calorific value implies a lower $S_x(\omega)$. Here two situations are studied; first the absolute level of the frequency spectrum is lowered (case 1 in figure 4.12), while the shape of the spectrum remains the same. Secondly (case 2 in figure 4.12), the cut-off frequency has been set to lower frequencies, while the absolute level is not changed. In both cases the standard deviation of the calorific value has been lowered to 75% compared to the of the original value. Figure 4.13 shows the result for the gas phase temperature. In the first case the $\sigma T_d$ is -25%, while in the second case the $\sigma T_d$ is -16.4%. Therefore, it can be concluded that less power in the higher frequencies of the calorific value has a relative lower contribution compared to the lower frequencies.

The above performed analysis based upon parameter variations leads to the conclusion that only the stationary value of $\Delta H$ has a great influence on the standard deviation of $\sigma T_d$. Furthermore, if it is possible to lower the frequency content of the calorific value it would be preferred to lower the power of the lower frequencies $< 10^{-3}$ rad $s^{-1}$ (which equals to about 105 minutes in the time-domain).
4.4.2 A practical example: Lifetime of a superheater as a function of disturbances

Corrosion of superheaters is a major problem in MSWC installations. Lifetime of the superheaters is limited, due to corrosion damage. The mechanism behind the corrosion is fairly complex and not completely understood in detail yet. In general it can be said that it is the iron in the superheater which reacts with the oxygen present in the air and the chlorine originating from the waste. The corrosion rate increases with increasing temperature. This corrosion is the main reason for the limitation of the steam pressure and temperature, which has a major negative impact on the total ef-
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ficiency of the steam cycle. As substitution of superheaters is very expensive, mainly caused by the fact that the plant has to be shut down, it is attractive to increase the lifetime of superheaters. As the corrosion is influenced by the temperature of the superheater material and thus the flue gas temperature, up till now the following postulate is common: the reduction of flue gas temperature fluctuations will result in longer superheater life times. The idea behind this postulate is that higher temperatures cause relatively more damage than lower temperatures. The aforementioned theory is very useful to investigate this postulate.

From literature [52] it is known that, depending on the kind of material, the corrosion rate can be described by either a linear or a parabolic kinetic relation:

\[ t = \frac{x}{k} \quad \text{or} \quad t = \frac{x^2}{k} \quad (4.24) \]

with \( x \) the maximum allowable penetrating depth of the corrosion in the superheater material (mm), \( t \) the lifetime in hours (h) and \( k \) the corrosion rate in (mm h\(^{-1}\)) or (mm\(^2\) h\(^{-1}\)). In the case of a parabolic kinetic relation, the corrosion rate decreases as a function of time, due to already corroded material that forms a protecting layer on the surface of the material. The corrosion rate \( k \) depends on the temperature and can be described by an Arrhenius equation:

\[ k = k_0 e^{-\frac{E_a}{RT}} \quad (4.25) \]

with \( k_0 \) the pre-exponential factor (mm h\(^{-1}\)) or (mm\(^2\) h\(^{-1}\)), \( E_a \) the activation energy (J mol\(^{-1}\)) and \( R \) the gas constant in (J mol\(^{-1}\) K\(^{-1}\)).

The lifetime is inverse proportional to the averaged corrosion rate for both proposed kinetic relations. The average corrosion rate \( \overline{k} \) is determined by the temperature variations as function of time. This temperature variation can be described by a probability density function \( p(T) \). Then the average corrosion rate becomes:

\[ \overline{k} = \int k(T)p(T)dT \quad (4.26) \]

Information gathered from the on-line calorific value sensor showed that variations in temperature are mainly caused by variations in the calorific value. Furthermore, it turned out that the standard deviation of the gas phase temperature is about 3%.

Due to the more stable steam-side temperature, the standard deviation of the superheater surface temperature will be lower than 5%. Simple heat transfer calculations show that the standard deviation on the steam-side will be about 2%. Now, with formula (4.26) the lifetime of superheaters can be calculated if the kinetic constants are known. Rademakers [52] carried out corrosion experiments with different materials in real plants during long periods (up to one year). The material was exposed to temperatures between 410°C and 520°C. For three different materials (Steel 15Mo3, Alloy 28 and Alloy 625) kinetic constants were found, which are given in table 4.3. The values for Alloy 28 and 625 are in good agreement with other literature [53]. For Steel 15Mo3 the linear kinetic relation is valid, while for Alloy 28 and 625 the parabolic kinetics holds.

Calculations show that the lifetime for Steel 15Mo3 is about one year and for both
Alloys it is about hundred year! The latter is caused by the parabolic relation. As the extracted relation from practical data is only validated for one year, extrapolation of this relation should be done with care. Based upon recent practical experiences [54] it is expected that the life-time for the Alloy 28 and 625 is at least 5 years. With the kinetic parameter values and the equation (4.26) calculations have been performed in order to investigate the influence of the temperature level and the standard deviation on the lifetime of the superheater. The average temperature was chosen to be 450°C, the probability distribution was assumed to be Gaussian, and the standard deviation was assumed to be maximal 5% as this was the standard deviation of the temperature in the flue gases. However, the actual variations at the material surface will be lower as the steam side temperature is more constant. Figure 4.14 shows the relative lifetime of the three materials as a function of an increase of the average temperature of the superheater. Figure 4.15 shows the relative lifetime for the materials

<table>
<thead>
<tr>
<th>Type</th>
<th>$k_0$ (mm/h)</th>
<th>$E_a$ (kJ/mol)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel 15Mo3</td>
<td>0.127</td>
<td>35.5</td>
</tr>
<tr>
<td>Alloy 28</td>
<td>0.498</td>
<td>107.2</td>
</tr>
<tr>
<td>Alloy 625</td>
<td>1.18</td>
<td>101.2</td>
</tr>
</tbody>
</table>

Table 4.3: Pre-exponential factor and activation energy for Steel 15Mo3, Alloy 28, and Alloy 625.

Figure 4.14: Relative lifetime(%) for Steel 15Mo3, Alloy 28, and Alloy 625 as a function of an increase in the average temperature. 100% is 1 year for Steel 15Mo and 5 years for Alloy 28. Base temperature is 450°C, while the percentage is based upon Kelvin.

as a function of the magnitude of the standard deviation. The lifetime decreases with increasing standard deviations, due to the non-linearity of the corrosion constant. As Alloy 28 and Alloy 625 have a higher activation energy and so a higher non-linearity, the life time decreases faster compared to Steel 15Mo3. Some main conclusions can
Figure 4.15: Relative lifetime(%) for Steel 15Mo3, Alloy 28, and Alloy 625 as a function of an increase in the standard deviation of the temperature. 100% is 1 year for Steel 15Mo3 and 5 years for Alloy 28. Base temperature is 450°C, while the percentage is based upon Kelvin.

be drawn from these figures. First, Alloy 28 and Alloy 625 have a longer lifetime than Steel 15Mo3 and so is more suitable for using in MSWC plants. This has already been shown earlier [52]. For Alloy 28 and 625 variations in temperature and standard deviation have no significant influence on the lifetime, as the calculated lifetime is about hundred years. However, as above discussed, from recent practical experiences it is expected that the lifetime is about 5 years. Further research is needed to investigate if the used parabolic relation still holds for these long periods.

Secondly, lowering the average temperature has more impact than lowering the standard deviation for all three materials. This is contrary with common accepted theory [54]. The influence of the standard deviation is low because of the fact that the probability distribution is Gaussian and that the temperature range is in the linear part of the Arrhenius equation. A few remarks have to be made here. The above treated theory assumes that the temperature is uniform across the flue gas channel. If, due to mixing problems there is no Gaussian distribution locally extreme damage can occur. The same holds for the uniformity of the O₂ concentration. Furthermore, in this theory the corrosion rate is only dependent on the temperature. Recently, new concepts are developed, which say that the corrosion rate increases when high fluctuations occur. Due to the high fluctuations higher temperature differences between the gas temperature and the metal surface temperature occur. This implies higher gradients, which can induce more condensation of chlorides on the metal surfaces and a higher diffusion of chlorides into the metal surface. Future research is necessary to reveal the exact influence of process fluctuations on the lifetime of superheaters.
4.5 Conclusions

In this chapter the random data collected with the on-line calorific value sensor have been analysed with theory from stochastic processes. This new application of stochastic concepts in the field of MSWC yields interesting new results. First, it has been proven that the irregularity in the calorific value waste can be described by a Gaussian, ergodic process. Secondly, the probability distribution of the calorific value, in \(kJ\text{ mol}^{-1}\), is Gaussian for the MSWC plants (HVC and Essent). An analysis of data from two other plants (AZN and Twence) showed that the probability distribution of the calorific value, in \(kJ\text{ mol}^{-1}\), is not exactly Gaussian. These deviations from Gaussian behaviour are probably caused by the mixing process in the bunker. Regarding the spectral density it has been concluded that due to the complexity of the process it was not possible to derive simple analytical relations for the spectral density.

Furthermore, it has been proven that the non-linear dynamic model describing the main processes in the furnace of a MSWC can be linearised in a working point without losing essential information. In practice the process will comprise several working points, which implies that the process can be described by a multiple linear system. This means that a complete system description will comprise a set of linear models, where the parameters of the linear models are dependent on the actual working point. Furthermore, it is proven that the non-linear dynamic model describing the main processes in the furnace of a MSWC, can be linearised in a working point without losing essential information. In practice, the process will comprise several working points, which implies that the process can be described by a multiple linear system. This means that a complete system description will include a set of linear models, whereby the parameters of the linear models are dependent on the actual working point.

New linear transfer functions have been derived from the linear model, which were used to carry out a sensitivity analysis in order to investigate the influence of process and design parameters on the standard deviation of the main output variables. It has been concluded that decreasing the calorific value, and thus changing the working point, has a large negative impact on the standard deviation of the gas phase temperature. This implies that combustion of waste with lower-than-normal calorific values is more difficult to control than waste with higher calorific values.

The transfer functions have been used in a practical application. As, the corrosion rate of superheaters can be described by an Arrhenius equation, the developed theory can be used to predict superheater lifetimes. With this model it has been concluded that the lifetime of a superheater is more influenced by the absolute level of the temperature than the fluctuations in the temperature. This is caused by the linearity of the corrosion rate in the temperature region in which the plant is operated. However, this model does not contain condensation of chlorides as function of temperature. Therefore, more research is needed in order to reveal the exact influence of process fluctuations on the lifetime of superheaters.
Chapter 5

Refined analysis of the combustion of solid fuels on a grate

5.1 Introduction

The conversion of solid fuels on a grate is one of the main processes in Municipal Solid Waste Combustion (MSWC). Grate firing is not only used for MSWC but also for combustion of other solid fuels, like wood and other biofuels e.g. straw or chicken-manure, into heat and power. For process control and development of furnaces and grates it is very important to understand the processes occurring on the grate well. In the early 70’s Rogers [11] published a qualitative description of municipal solid fuel conversion on a grate. He described the conversion with an ignition front that propagates from the upper side of the fuel bed towards the grate surface. The upper layer of the fuel bed is heated by radiation from the flame in the gas phase. During this propagation phase, oxidation produces heat which is used to dry and devolatilise the raw material below the reaction zone. Due to the counter current flow of primary air the reaction zone is small and very distinctive. If there is a shortage of oxygen in the reaction zone, a char layer can be formed. From the moment that the reaction front reaches the grate, the reaction front goes up and converts the remaining char with oxygen into CO and CO₂ during the rest of the residence time of the fuel on the grate. In chapter 3 it was found that for different primary air temperatures, different phenomena were observed. The differences at different primary air temperatures cannot be explained by Rogers’ theory. This is important as different suppliers of MSWC plants use primary air preheating. Up till now, the general accepted explanation for the influence of the primary air temperature is the fact that the preheated air dries the fuel and the ignition will take place earlier on the grate. In this chapter it will be shown that it is impossible to dry the fuel with the preheated air during the residence time of the fuel in the first grate section. So, other phenomena in the waste layer have to be considered.

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Furthermore, Rogers did not present any quantitative description of the phenomena. Gort [10] studied this subject extensively. As real-scale installations are not very suitable for studying in detail the combustion behaviour, Gort [55] and [56] used in the early nineties a batch-type reactor (pot-furnace) for grate combustion experiments with solid fuels. Later in the nineties, a number of researchers used the same type of reactor to study the ignition phenomena of solid fuels [12],[13],[14], and [15]. Most of these authors used these experiments to validate models of solid fuel combustion on a grate. Most of these descriptions have the disadvantage that they are not translating the results from the laboratory to real MSWC plants with moving grates. Thunman [12] made a start by applying the results from laboratory experiments to real-scale furnaces. He found that it is not possible to apply the results directly to real-scale combustion plants of wood chips. A new theory was developed, in which it is proposed that the fuel bed is not ignited from the top by radiation, but at the grate surface.

Other authors [17],[18],[19], and [20] are focusing on the design of the furnace by using Computational Fluid Dynamics (CFD) calculations. As the solid fuel bed acts in CFD calculations as a boundary condition, these authors developed also solid fuel models of the waste layer. Laboratory experiments on solid fuel conversion were not used to validate the models. With measurements from real plants the complete CFD-model is validated, while the bed model is validated indirectly in this way. CFD-calculations are mainly used to calculate the stationary state of the furnace. With these calculations optimisation of the furnace can be performed, for example the injection of secondary air. However, the continuously changing composition of the waste makes the process instationary. Furthermore, some MSWC plant suppliers are using discontinuous grate systems, which makes the system inherent dynamic. For these reasons, in this study the observed dynamic behaviour will be used to develop a theory for combustion of solid fuels on a grate. First, based upon the results found in chapter 3, new pot furnace experiments with air preheating were performed, which will be described in the next section. The third section of this chapter will compare observations in real-scale plants with the pot furnace experiments. Here, also new important conclusions about the usefulness of pot furnace experiments for studying the combustion behaviour of a solid fuel on a grate will be given. In the fourth section a new extension of the well-known theory from Rogers is presented. In the fifth section the proposed theory will be applied to observations in real MSWC plants. Finally, conclusions will be given.

5.2 Experiments in pot furnaces

These section is divided into two parts. First, a summary will be given about experiments without primary air preheating. This is mainly based on previous work carried out by TNO. Secondly the experiments with preheated air will be discussed.
5.2.1 Solid fuel combustion without preheated air

This section will summarize the work on pot furnace experiments without preheated air carried out earlier at TNO, [10], [55], [57], [58]. In chapter 3, it has already been mentioned that Gort and Brim [55] were the first who carried out quantitative research in the mechanisms of solid waste combustion on a grate. They studied the combustion process in a packed bed experimentally as well as theoretically. Assuming that no mixing of waste occurs over the length of the grate, an assumption which holds very well in practice and is also confirmed by others [41], a batch packed bed reactor, also called a pot furnace reactor, can simulate the combustion process of a volume of burning waste that travels along the grate with constant speed. In real plants there is a feedback of radiation from the back-end of the fire to the beginning of the fire. This "radiation interaction" is not covered by this process, while the influence of the grate movement is also not incorporated. However, the pot furnace experiments give a good view of the processes which occur in the waste layer. Gort [10] carried out almost all experiments without preheated air and used wood blocks, coals, as well as shredded municipal solid waste. Furthermore, Ortmanns [57] also carried out experiments to investigate the processes occurring in the waste-pile. Municipal solid waste, with different compositions (moisture, combustible, and inert fractions) was used, which was composed manually from sorted municipal solid waste. In this way it was possible to determine more precisely the calorific value of the waste and the influence of it on the experiments. The used experimental setup is shown in figure 5.1. From beneath primary air is supplied while secondary air is supplied above the

![Diagram of the experimental set-up for packed-bed combustion.](image)

Figure 5.1: Diagram of the experimental set-up for packed-bed combustion.

waste layer to enable a further burn-out of the gases. The system is placed on a mass balance to monitor the decrease of the weight. Thermocouples are placed along the height (6 in total, 15 cm equidistant) and flue gas is sampled at the top of the bed in order to study the combustion behaviour. The process is started by using a heating element above the reactor. After ignition the heating element is removed and the process runs autonomously. The reactor has a height of about 150 cm, while
the internal diameter is 15 cm. The reactor can contain about 5-7 kg of fuel. The primary air temperature can be heated up to 500 °C. Every 20 seconds temperature and concentration signals are sent to a computer.

Gort [10], Ortmanns [57] and Koch [58] discovered that the combustion process behaves like a reaction front that moves slowly from the top to the bottom of the bed. There is a very clear distinction between a hot burning part (active) and a cold fresh (non-active) part. In figure 5.2 the temperature along the bed as a function of time shows clearly this sharp reaction front.

![Temperature as a function of height and time in an experiment with no primary air preheating.](image)

**Figure 5.2:** Temperatures as a function of height in the bed and time in an experiment with no primary air preheating. 0 cm corresponds to the bottom of the reactor.

Gort defined an ignition speed and a combustion rate. The ignition speed is defined as the speed of the reaction front in the packed bed, while the combustion rate is the amount of converted mass per second. The ignition speed is derived from the thermocouple measurements as shown in figure 5.2, while the combustion rate is calculated from the mass balance signal. The ignition speed multiplied with the density of the fuel gives the ignition rate. The units of the combustion rate and the ignition rate are both \( km^{-2} s^{-1} \).

To study the influence of the primary air supply, the experiments were carried out with different superficial air velocities. Figure 5.3 shows a typical ignition rate curve from the experiments found by Gort as well as by Koch. The general shape of the curve is explained by Gort's theory. First, oxygen will be rate limiting. More oxygen will result in a higher conversion rate and thus a higher ignition rate. At higher primary air flows the kinetic rate will become limiting and the primary air acts as a cooling agent which results in a decreasing curve. Besides the performed pot-furnace experiments Gort developed a mathematical theory to explain the curve from figure 5.3. He solved analytically the non-linear equations which describe the behaviour of the material in the bed. The interested reader is referred to [10]. The above shown results agree with the results of [11] and later studies [12], [13], [14],[20],[42],[59], and
Figure 5.3: Ignition rate as a function of superficial air velocity.

[60]. All authors found more or less the same results as Gort found analytically. However, the theory of Gort has the disadvantage that the model needs a lot of input data (kinetics, heat and mass transfer relations, bed porosity) which are difficult to determine. As a result Gort’s model can not easily be used in other models. The purpose of the performed research was to find a practical description of the combustion behaviour, which can be used in dynamic models describing the complete MSWC process. Based on all experiments carried out in the laboratories it was possible to derive an empirical relation for the ignition rate.

The empirical relation doesn’t contain any explicit temperature dependency, but a linear dependence on the cross-sectional area of the reactor. This is surprising because in chapter 2, equation (3.15) gives a general relation for the reaction rate, which was able to describe the global phenomena of the MSWC (see section 3.5). Implicitly the temperature dependency is included in the empirical relation by an influence of the moisture fraction. Furthermore, it has to be noticed that in this description the reaction rate is not dependent on the amount of mass. It has turned out that this proposed relation is a more convenient way for describing the reaction rate. However, in section 5.3 it will be shown that for real plants this relation is not valid. In section 5.4 attention will be given how to translate this result into a useful relation for a real scale plant. The presented relationship is only valid for non-preheated air situations. Therefore in the next section new experiments in which the influence of preheated air on the combustion behaviour is studied, will be discussed.

5.2.2 Influence of preheated primary air on solid fuel conversion

Up till now, the general opinion is that preheated primary air is responsible for the drying of the waste in the first zone of the grate. However, the primary air temperature is mostly maximised to about 170 °C. Given the amount of primary flow that is used
in practice in the first zone of the grate, it can be shown with some simple calculations that only a few percent of the moisture in the waste can evaporate in the first zone. So, it can be concluded that this opinion is not supported by physical laws. Therefore, it would be useful to know the exact influence of preheated primary air on the process behaviour. With this knowledge, it is for example possible to use the primary air temperature as an active control variable. The main goal is to reveal the influence of preheated primary air on the combustion process. Pot furnace experiments have been used to study these effects. Although in normal practice, inhomogeneous waste is burnt, in the performed experiments a more well-defined fuel (wood) has been used. The aim is not to achieve a complete detailed description of the effect of preheated primary air, but to present some general conclusions regarding preheated primary air, that can be translated in formulas to be used in combustion models.

Experimental observations

To study the process behaviour with primary air preheating, experiments have been carried out with the in section 5.2.1 described experimental set-up. General observations during the experiments with air preheating, showed that in the beginning there is an ignition front like in the experiments with no air preheating. Furthermore, as a result from the energy in the primary air, moisture is evaporated in the lower part of the reactor. Due to this evaporation, the temperature of the primary air will decrease. Depending on the amount of primary air and temperature the evaporated water can condense on the fuel more upstream in the reactor. Anyway, the gas that flows through the burning layer contains a higher content of water vapour compared to the situation without preheating. During the experiments the evaporation will continue and as a result an evaporation front will propagate along the height of the reactor. At the same time the combustion front propagates down the packed bed. At a specific time, the breakthrough time, \( t_{bw} \) these two fronts will meet each other. At that moment the ignition rate increases significantly and the reaction front reaches very quickly the bottom of the bed. As a result the total remaining mass burns, so the combustion regime has changed completely. The duration of the conversion process after the breakthrough is small (5-15 minutes) compared to the time before the breakthrough (120-180 minutes). So the breakthrough time is a good indication of the total reaction time. It was tried to distinguish between the mass converted before the breakthrough time and the remaining mass, but this was not possible because of the amount of char remaining after combustion.

In order to be able to explain the experiments, four quantities have been defined which can characterise the experiments. Besides the already mentioned ignition rate \( (U_I) \) and combustion rate \( (U_c) \) the evaporation rate \( (U_e) \) \((cm\ min^{-1})\), the breakthrough time of the reaction front \( (t_{bw}) \) \((min)\) and the acceleration \( (V) \) \((\cdot)\) are defined. The latter is defined as the ratio of the combustion rate before the breakthrough and the combustion rate after the breakthrough. The breakthrough time is defined as the time elapsed from the moment that the first thermocouple above the grate reaches a temperature of \( 400\ ^\circ C\). As the measured combustion rate \( (U_c) \) is disturbed by the amount of moisture that is evaporated, the combustion rate is corrected by the evaporation rate \( (U_e) \). This evaporation rate is measured by the front speed of the drying
front. The drying front is defined as the place where the gas temperature is equal to
the air temperature at the entrance of the reactor. To validate this value, a plug flow
model was developed which is capable to calculate the theoretical evaporation speed
($U_{v,t}$) based upon mass and heat transfer equations in a waste pile as a function of
time. In appendix B, the model is described. Input parameters for this model are the
primary air velocity $U_g$ in the fuel bed, the amount of moisture in the air $X_{f(air)}$, the
temperature $T_{fuel}$ of the fuel, the temperature $T_{prim}$ of the air, and the diameter
d$_{fuel}$ of the bed particles. The place where the breakthrough takes place is obtained
from the product of the evaporation speed and the breakthrough time.

Reference experiment

In all pot-furnace experiments the same well-defined fuels were used; cubic wood
blocks of 18 mm and a moisture fraction of 30 percent. First, a reference experiment
with no preheated primary air ($T_{prim} = 20{\degree}C$) was carried out. Table 5.1 shows
the results of this experiment. Although the preheated primary air is only 20{\degree}C in
this experiment, still evaporation takes place. After 181 minutes the ignition rate
accelerates with a factor 2. The values for $U_f$ and $U_v$ are in good agreement with
earlier experiments [10],[57],[58]. Exact comparison is not possible because the last
two authors used municipal solid waste. Instead of waste Gotz used wood blocks, but
his wood blocks had other dimensions (10 and 30 mm), while the moisture fractions
were 10% and 30% for the 10 mm wood cubes and 10% for the 30 mm ones.

<table>
<thead>
<tr>
<th>$U_g$</th>
<th>$U_f$</th>
<th>$U_v$</th>
<th>$t_{bg}$</th>
<th>$V$</th>
<th>$U_{v,t}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>(m s$^{-1}$)</td>
<td>(kg m$^{-2}$ s$^{-1}$)</td>
<td>(kg m$^{-2}$ s$^{-1}$)</td>
<td>(min)</td>
<td>(-)</td>
<td>(cm min$^{-1}$)</td>
</tr>
<tr>
<td>0.15</td>
<td>0.029</td>
<td>0.030</td>
<td>181</td>
<td>2.0</td>
<td>0.072</td>
</tr>
</tbody>
</table>

Table 5.1: Process numbers for the reference experiment with only wood cubes and
$T_{prim} = 20{\degree}C$. $U_f$ and $U_v$ are valid during the process till the breakthrough of
the process.

1) In this case it was not possible to determine a reliable value for the evaporation
rate.

Experiments with wood cubes only

In these experiments the superficial gas velocity of the primary air has been changed
at two different primary air temperatures, 140 and 170 {\degree}C. Tables 5.2 and 5.3
show the results for these experiments. One of the most striking observations is that
the combustion rate $U_v$ is higher than the ignition rate $U_f$. This is very strange as
from theory $U_v \leq U_f$. An explanation may be found in the calculation of $U_f$. In that
calculation use has been made of the initial density of the packed bed, that due to
several reasons can change during the experiments. A first reason could be that the
density of the fuel changes. Probably this is not the reason, because Gotz found that
$U_f$ was higher than $U_v$ (especially at higher $U_g$) in his experiments without preheated
air. Another possible reason could be that the density increased because of a redis-
tribution of the moisture which can condense on higher parts. This explanation is
supported by the fact that in the reference experiments there is almost no difference between \( U_f \) and \( U_v \). The small difference (3\%) can be explained by measurement errors. Calculations with the plug flow model support this explanation. It was found that at \( T_{prim} = 140^\circ C \) moisture condenses in the upper part of the bed. This results in a temporary increase of the percentage moisture of 4\%, which indicates a density change of about 4\% as extra moisture decreases only the porosity of the packed bed. In the situation of \( T_{prim} = 170^\circ C \) the moisture increases about 5\%, resulting in a density change of about 5\%.

Figure 5.4 shows the ignition rate \( U_f \) as a function of the superficial primary air gas velocity for different primary air temperatures. At first glance two facts can be noticed. First, at \( U_g = 0.15 \text{ m s}^{-1} \) the \( U_f \) is the highest for the situation without preheated primary air (\( T_{prim} = 20^\circ C \)). Although the differences are small, this can be explained by the evaporation of moisture in the lower part of the bed. As already discussed, moisture in the lower part of the bed will evaporate, which results in a decrease of the air temperature. Compared to the situation with no preheated air, the temperature of the primary air flow in the bed is only about 15 \(^\circ C\) higher and almost saturated. Moisture is condensed in higher regions of the waste pile, increasing the moisture content which results in a lower ignition rate. Due to the evaporation saturated air flows through the burning reaction front at the top of the bed, where the extra water vapour in the air has to be heated up. This energy is extracted from the burning layer, also contributing to a decreasing ignition rate.

The second remark to be made is about the shape of the curves. The curve for \( T_{prim} = 170^\circ C \) has a shape like the ones in former studies, but the maximum lies at a low \( U_g \) value (0.15 m s\(^{-1}\)), compared to the curve for \( T_{prim} = 140^\circ C \). There the curve increases with increasing \( U_f \). It is known from earlier experiments that the top of the curve shifts to higher \( U_g \) values at higher calorific values and lower moisture concentrations of the fuel. However, the difference at the top of the curve cannot

<table>
<thead>
<tr>
<th>( U_g ) (m s(^{-1}))</th>
<th>( U_f ) (kg m(^{-2}) s(^{-1}))</th>
<th>( U_v ) (cm min(^{-1}))</th>
<th>( \phi ) (min)</th>
<th>( V ) (-)</th>
<th>( U_{t,b} ) (cm min(^{-1}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.10</td>
<td>0.021</td>
<td>0.200</td>
<td>0.030</td>
<td>146</td>
<td>8.0</td>
</tr>
<tr>
<td>0.15</td>
<td>0.028</td>
<td>0.200</td>
<td>0.031</td>
<td>134</td>
<td>6.0</td>
</tr>
<tr>
<td>0.20</td>
<td>0.029</td>
<td>0.200</td>
<td>0.033</td>
<td>117</td>
<td>3.6</td>
</tr>
</tbody>
</table>

Table 5.2: Process numbers as a function of the superficial primary air velocity for experiments with only wood cubes and \( T_{prim} = 140^\circ C \). \( U_f \) and \( U_v \) are valid during the process till the breakthrough of the process.

<table>
<thead>
<tr>
<th>( U_g ) (m s(^{-1}))</th>
<th>( U_f ) (kg m(^{-2}) s(^{-1}))</th>
<th>( U_v ) (cm min(^{-1}))</th>
<th>( \phi ) (min)</th>
<th>( V ) (-)</th>
<th>( U_{t,b} ) (cm min(^{-1}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.10</td>
<td>0.021</td>
<td>0.250</td>
<td>0.059</td>
<td>122</td>
<td>2.1</td>
</tr>
<tr>
<td>0.15</td>
<td>0.028</td>
<td>0.400</td>
<td>0.090</td>
<td>120</td>
<td>10.0</td>
</tr>
<tr>
<td>0.20</td>
<td>0.029</td>
<td>0.470</td>
<td>0.095</td>
<td>123</td>
<td>2.0</td>
</tr>
</tbody>
</table>

Table 5.3: Process numbers as a function of the superficial primary air velocity for experiments with only wood cubes and \( T_{prim} = 170^\circ C \). \( U_f \) and \( U_v \) are valid during the process till the breakthrough of the process.
be explained with the moisture redistribution, because the temporary change of the moisture increase is 4% \( T_{prim} = 140^\circ C \) and 5% \( T_{prim} = 170^\circ C \). At \( T_{prim} = 140^\circ C \) no experiments with higher \( U_g \) values are carried out, but it is expected that the top of the curve lies at a higher \( U_g \) value. Additional experiments have to be carried out to study this behaviour.

![Graph of \( U_g \) vs. \( U_{g,prim} \)]

**Figure 5.4:** Ignition rate \( U_g \) as a function of the superficial primary air gas velocity for different primary air temperatures.

A point to be discussed is the acceleration factor \( V \). Due to the acceleration and the shorter breakthrough time, the total time needed for complete combustion is shortened in the case with preheated air. Exact values cannot be given as it was not possible to measure the total reaction time. Shortly after the breakthrough the experiments had to be finished due to the high temperatures caused by the remaining char oxidation.

At higher primary air temperatures, the acceleration factor is higher than at lower primary air temperatures. At higher primary air temperatures in the beginning more moisture is heated up, which results in lower temperatures in the burning layer. As a consequence, the combustion rate will be lower than with lower preheated temperatures. However, after the breakthrough the situation is reversed. At higher primary air temperatures, more energy is fed to the burning layer, which results in higher temperatures in the layer compared to the situation of lower primary air temperatures. As a logic consequence the acceleration factor \( (V) \) is higher with higher primary air temperatures. One of the hypotheses is that after the breakthrough time the reaction rate is proportional to the amount of remaining mass. Unfortunately, it was not possible to detect this relationship, due to the already formed char during the combustion. So, it was not possible to distinguish between the mass formed by char, that is formed before the breakthrough, and the remaining unconverted mass. Based upon these experiments, no conclusions can be given about the relation between the reaction rate and the amount of unconverted mass.
Comparison of the experimental evaporation rate values with the theoretical ones, shows that in all cases, except one ($T_{prim} = 170 \, ^\circ C$, $U_g = 0.15 \, m/s^{-1}$) the theoretical value is about 30 % higher. A sensitivity analysis has been carried out with the plug flow model and showed that the temperature and amount of the air have the greatest influence on the evaporation speed. It is noted that the particle size has not a large influence on the speed of the evaporation front. The reason is that the evaporation process is limited by the heat input of the primary air. The only reason for the deviation is to be find in heat losses in the reactor. During the experiment the grate at the bottom will be heated up, just like the reactor wall. Furthermore, the method to determine the experimental evaporation rate is not very accurate causing measurement errors. Probably this is the reason for the deviating experiment. As the comparison of the evaporation rates is only to support the experiments no additional experiment were carried out.

**Experiments with wood cubes mixed with plastics**

Research carried out at the TNO-laboratories showed that when waste is heated up with preheated air of about 180 °C, waste can ignite spontaneously. This phenomenon is also observed when waste is put into an electric oven and the temperature is increased up to 180 °C. Furthermore, from literature [61],[62],[63],[64], and [65] it is known that condensed-enhanced ignition can occur in piles of fuels. This theory says that when humid air flows along relative dry fuel, the moisture will condense and release its heat. As the time-scale for heat transfer is smaller than the time-scale for mass-transfer in some circumstances this condensing can locally result in a concentrated heat release, which largely increases the local temperature. Sometimes it can cause an ignition of the fuel. Another mechanism is that the saturated air cannot contain more moisture. So the released heat cannot be used for evaporation. In that case it will be transferred to the non-moisture parts which could ignite then. So, a critical heat loss rate should exist to avoid self-ignition. This is described by the classical theory of Frank-Kaminetskii [61].

Keeping both phenomena in mind, experiments with plastics were carried out. The idea was that plastics ignite more easily, compared to wet wood cubes. As in general, the initial plastic load is free of moisture, condensation of moisture on the plastics surface could cause a spontaneous ignition of the packed bed. Based upon this assumption poly-ethylene spheres (d=3 mm) were mixed with wood cubes, but with keeping the caloric value constant. This was achieved by increasing the moisture fraction of the wood up to 35 percent. Mixing 2.3 percent poly-ethylene with 97.7 percent wood results in a constant caloric value of 12.6 MJ kg$^{-1}$. Only 3 experiments with a poly-ethylene wood mixture have been carried out. Table 5.4 summarises the results of these experiments. Compared to the experiments with only wood cubes, the ignition rate is lower. This is expected because it follows from former experiments (e.g. [10]) that the influence of the moisture fraction is high. The evaporation rate $U_e$ was lower compared to the other experiments (see tables reftablepa140 and 5.3). It is obvious that this is a result of the wetter wood cubes. As a result the breakthrough time is higher. The acceleration factor is a little bit higher than in the case without plastics. This can be explained by the fact that the remaining dry wood cubes are
mixed with plastic and thus the calorific value has increased substantially. Plastic also has a lower ignition temperature. These two effects result in a higher acceleration rate. A higher acceleration has the disadvantage that the oxygen supply becomes rate limiting. In the case of high primary air temperature, after the breakthrough relative much unburnt gases (CO) were detected.

In these experiments the acceleration rate also indicates that the total combustion time will be shorter despite the more or less same breakthrough time. As already mentioned it was not possible to detect the total combustion time precisely.

The difference between the theoretical evaporation rate and the experimental evaporation rate is about the same level as in the experiments with only wood cubes. There a possible explanation has already been given.

In all three experiments ignition of the packed bed due to preheated air was not observed. Therefore, it is concluded that ignition induced by condensing of the moisture on plastic spheres will not occur. Also self-ignition, which from a theoretical point of view could have taken place in the dry lower parts of the packed bed, did not occur in the experiments. As the main objective was the investigation of ignition of the bed no more experiments with plastic-wood mixtures were carried out.

### 5.2.3 Summary of the pot furnace experiments

The influence of primary air preheating in pot furnace laboratory experiments can be summarised as follows:

- The combustion front on the top of the bed remains, but the ignition rate will in general be lower than in the case without primary air heating. This is because of the condensing moisture in higher parts of the packed bed. Also the heating of the evaporated water vapour up to about 900 °C in the reaction zone has a negative influence on the ignition rate.

- At the same time, an evaporation front propagates from the bottom to the top of the fuel bed. At higher primary air temperatures this propagation speed will be higher. After a certain time these two fronts meet each other. At that moment the remaining mass starts to burn almost instantaneously. At high primary air temperatures, the combustion rate is accelerated with about a factor 10. So, instead of one combustion regime, there are two different combustion regimes in the situation of preheated primary air.

<table>
<thead>
<tr>
<th>$T_{\text{prim}}$ (°C)</th>
<th>$U_f$ (kg m$^{-2}$ s$^{-1}$)</th>
<th>$U_e$ (cm min$^{-1}$)</th>
<th>$U_w$ (kg m$^{-3}$ s$^{-1}$)</th>
<th>$t_b$ (min)</th>
<th>$V$ (-)</th>
<th>$U_{e,\text{app}}$ (cm min$^{-1}$)</th>
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</thead>
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<tr>
<td>20</td>
<td>0.025</td>
<td>0.25</td>
<td>0.036</td>
<td>162</td>
<td>2.0</td>
<td>0.370</td>
</tr>
<tr>
<td>140</td>
<td>0.023</td>
<td>0.036</td>
<td>0.024</td>
<td>162</td>
<td>2.0</td>
<td>0.370</td>
</tr>
<tr>
<td>170</td>
<td>0.033</td>
<td>0.310</td>
<td>0.049</td>
<td>174</td>
<td>10.4</td>
<td>0.370</td>
</tr>
</tbody>
</table>

*Table 5.4: Process numbers as a function of the primary air temperature for experiments with wood cubes mixed with plastics and $U_g = 0.15$ m s$^{-1}$. $U_f$ and $U_e$ are valid during the process till the breakthrough of the process.*

1) In this case it was not possible to determine a reliable value for the evaporation rate.
• The total reaction time is reduced with about 40-60 minutes for the situation with \( U_g = 0.15 \text{ m s}^{-1} \). For the other two gas velocities, no experiments without preheated primary air have been performed. However, the difference in total reaction time between \( T_{prim} = 140^\circ C \) and \( T_{prim} = 170^\circ C \) is limited.

• No autonomous ignition or ignition induced by moisture condensation is observed when mixtures of wood and plastic are used.

5.3 Comparison of pot furnace experiments with MSWC plants

In this section the results from the laboratory experiments will be verified against observations in real-scale plants.

5.3.1 General observations in MSWC plants

To translate the laboratory results to real MSWC plants, first the observations from MSWC plants will be summarised. As already mentioned in the introduction some grate suppliers are using preheated primary air, while others do not. The application of preheated primary air depends mostly on the calorific value of the fuel. However in cases where some suppliers use preheated air, others do not use it. The difference of both philosophies can be seen in the dimensions of the grate and the grate damage. In general without preheated air the combustion behaves like a pipe-bowl combustion, the same as in the laboratory experiments. Based upon the laboratory experiments it is known that when air preheating is used two regimes will occur: pipe-bowl and total combustion. The advantage of the total combustion regime is that a smaller grate can be used as the average combustion rate is higher. Using no preheated air has the disadvantage of a larger, and therefore more expensive grate, but there is the advantage of less grate damage due to corrosion. In most cases in plants with air preheating in the vicinity of the end of the first section of the grate and the beginning of the second section, the corrosion damages are the highest. Thermocouples placed at these places in the grate indicate that the temperature at the end of the first grate section is about 500 \(^\circ C\), which becomes lower further downstream the grate. At biomass grate incinerators Thunman [12] found the same temperature level and distribution along the grate. Furthermore, Thunman had the possibility to look through a small window to the fire front, i.e. near the feeding system (normally it is only possible to look at the last part, near the grate end) and saw a lot of smoke coming from the bed in the first sections. He also found that volatiles are released from the first section up to the fourth section. The presence of air preheating results in an ignition of the total height of the waste layer at the end of the first zone of the grate. The first question that arises is to check whether it is possible to dry the waste completely in the first section of a grate. This can be calculated easily by an energy balance over the first section if the amount of primary air, the waste flow, the moisture content in the waste and the temperatures of the entering waste and primary air are known. Then it follows that for normal operating
conditions only a small part (5-15\%) of the total moisture in the waste can evaporate in the first section. Thuman [12] came to the same conclusion by measurements in wood-fired grate systems. Hence, the influence of the primary air temperature must be different to what is generally thought. From real-scale plants, combustion rates can be easily deduced as the grate surface area and the waste throughput are known. As the last part of the grate is mainly used to achieve a good burn-out about only 80\% of the grate area is important for the combustion. Urban [66] has given values for the mass load for different grate suppliers. If this is corrected with a factor 0.8 the values for the ignition rate range from 0.09 to 0.17 \( kg \, m^{-2} \, s^{-1} \). These values agree with values found in a lot of MSWC plants and wood-fired boilers. For installations that are not using primary air preheating the values are generally lower (0.07-0.09 \( kg \, m^{-2} \, s^{-1} \)).

Another remarkable issue is that the superficial velocities of the primary air are much higher (0.4-0.5 \( m \, s^{-1} \)) in the main part of the fire in MSWC furnaces, compared to the velocities that can be achieved in laboratory experiments. This can be explained by the fact that due to the well-defined fuels used in the laboratory experiments, the air distribution is almost optimal, while in practice the structure of the material is very complex, resulting in a less good air distribution. Besides that the waste in real plants is moving due to the grate movement, which can result in air redistributions. Furthermore, although the superficial velocities in some laboratory experiments are the same as in real MSWC plants, in practice the momentum of the air flow is much greater due to the small air nozzles in the grate bars at the bottom of the waste bed. Due to high local velocities not all air is well divided across the waste layer. To translate the results from the pot furnace experiments to MSWC plants these effects have to be taken into account. It is very difficult to quantify these effects based upon theoretical considerations. Therefore, an empirical factor \( X_{\text{air}} \) is defined to incorporate the effect of the bad distribution of the primary air. This factor is defined as the fraction of air that goes through the waste layer into the gas phase without having interaction with the solid material in the bed. This value \( X_{\text{air}} \) has to be empirically derived from real-plant data. Recently Yang [67] published results from a study where the effect of channelling in MSWC has been studied quantitatively.

It was already mentioned that from the pot-furnace experiments without air preheating an empirical correlation for the ignition rate was derived. This equation was originally used in the complete dynamic model that is described elsewhere [68], [69]. However, it turned out that it was impossible to achieve any physically reliable results without multiplying the experimental equation with a multiplication factor. In other words, the experimental observed ignition rates are much lower than would be expected based upon modelling work and practical data from the pot-furnace experiments.

5.3.2 Results from system identification experiments on MSWC plants

In chapter 3.4 results are shown from models obtained from system identification and physical models. At different preheated primary air temperatures different step responses were found. The response of the steam production on a step change in
the waste supply results in the situation of $T_{\text{prim}} = 70^\circ C$ in a response without inverse response, while in the case of $T_{\text{prim}} = 120^\circ C$ there is a clear inverse response. As already mentioned, the situation with $T_{\text{prim}} = 70^\circ C$ was only possible during a certain time when fresh, relative dry waste was burnt. This indicates that the calorific value was higher in the $T_{\text{prim}} = 70^\circ C$ than in the $T_{\text{prim}} = 120^\circ C$ situation. Unfortunately only the average calorific value ($9.5 \text{ MJ kg}^{-1}$) during the experiments with $T_{\text{prim}} = 120^\circ C$ was known. Furthermore, it was clear from the identification results, that the reaction rate is proportional to the amount of burning mass on the grate. This is contrary to the observations in the pot furnace experiments, which indicates that the reaction rate is proportional to the surface area. So, it can be concluded that the combustion mechanisms in the pilot plant experiments differ from real plants. The reason for this discrepancy can mainly be found in the dynamic movement of the waste on the actual grate ("poking"). In the packed bed, the waste-pile is a stationary pile, while in the plant waste is fed continuously into the furnace, which is strongly influencing the structure of the waste pile in the first zone.

5.3.3 Conclusions from observations

The major conclusion which can be made from the aforementioned phenomena is that the results from pot furnace experiments cannot explain the quantitative as well as qualitative observed phenomena in real plants. Hence, the conclusion that in general pot furnace experiments do not represent solid fuel combustion behaviour on grates is justified very well. The main reason for this deviation is the solid fuel movement that is induced by the grate movement. However, pot furnace experiments can be a useful tool to study some more general effects, like e.g. the effect of self-ignition, air preheating and the general combustion behaviour. With the latter the structure of the relation between ignition rate and superficial air velocity is meant especially.

5.4 Extended theory for combustion of solid fuels on a grate

In this section the present theory for the combustion of solid fuels on a grate will be extended in order to explain the observations with and without air preheating. The proposed extension is qualitative as well as quantitative; i.e. it can be used in overall dynamic process models. First, a short description of the quantitative theory will be given. In chapter 3 general mass and energy balances for the fuel layer and the gas phase have been given. An analysis has shown that for describing the main dynamics of the combustion process in the furnace, the derived partial differential equations can be transformed into some simple, ordinary differential equations. The resulting model comprises three instationary balances (mass and energy balance for the waste layer and one energy balance for the steam system) and one stationary gas phase balance. The gas phase can be considered to be stationary as the time constant of the gas phase is negligible, compared to the other time constants in the process. In the model the reaction rate is described by a classical first order surface reaction with externally limited mass
transferred:

\[ \text{Reaction rate} = RaM = \frac{1}{k_d} + \frac{1}{k_a \epsilon \rho_{\text{a}} \rho_{\text{f}}} \frac{[O_2] M_{O_2} a M}{\nu_{O_2}} \]  

(5.1)

with \( k_d \), the mass transfer coefficient \((m \, s^{-1})\), \( E_a \), the activation energy \((J \, mol^{-1} \, K^{-1})\), \( M_{O_2} \), the molecular mass of oxygen \((kg \, mol^{-1})\), \( k_0 \) a pre-exponential constant \((m \, s^{-1})\), \( R_g \) the universal gas constant \((J mol^{-1} K^{-1})\), \([O_2]\) the oxygen concentration \((mol \, m^{-3})\), \( \nu_{O_2} \), the stoichiometric factor \((kg_{O_2} \, kg_{\text{waste}}^{-1})\), \( a \) the interfacial area \((m^2 \, kg^{-1})\), \( T_p \), the temperature of the fuel bed, and \( M \) the amount of burning fuel \((kg)\).

Based upon the observations in real plants and the pot furnace experiments it was found that there are two major combustion regimes: the pipe-bowl regime and the total combustion regime. It will depend on the specific circumstances which regime will occur. In chapter 3 it has been shown that for both regimes the above mentioned reaction rate can be used. In the subsequent sections the two regimes will be described in more detail.

### 5.4.1 Pipe-bowl combustion regime

In the first section on the grate, the combustion process starts by heating the upper layer of the bed by radiation from the flames, resulting in a high release of volatiles. After this initial phase a propagation front is created which propagates slowly to the bottom of the bed. At the place where the front reaches the upper surface of the grate, the fire end is also located. During the propagation, the processes occurring in the waste layer are drying and devolatilising. Combustion of the gases will partly take place in the gas phase of the waste pile and partly in the gas phase above the waste pile. The oxygen is used for the conversion of the volatiles and if oxygen remains this will be used for gasification and combustion of the char layer.

From the laboratory experiments no relation between the reaction rate and the amount of mass could be found, in contrast to the identification experiments, which were carried out in plants were also the pipe-bowl regime was present (i.e., situations without primary air preheating). Since it was also found that the pot furnace experiments have only limited usefulness for real plants, the proposed theory is for a great extent based upon the results from the real plants. Application of formula (5.1) shows that the ignition rate is dependent on the amount of mass and has the following relation: \( R = k_d \frac{[O_2] M_{O_2} a M}{\nu_{O_2}} = U_f a M \), with \( U_f \) the ignition rate in \( kgm^{-2} \, s^{-1} \). Here \( M \) is the burning mass in \( kg \), which means only the mass in the upper part of the layer. The interfacial area \( a \) has to be determined experimentally from the experiments in the real plants. From the experiments it followed that the reaction rate is only dependent of the mass transfer coefficient \( k_d \), i.e., the reaction is mass transfer limited. For \( U_f \) empirical relations from pot furnace experiments can be used, where the superficial air velocity has to be corrected by \( U_g = X_{\text{air}} U_g - \text{part} \) and where the relation has to be multiplied by a correction factor (as discussed in section 5.3) in order to achieve a good magnitude for the combustion rate. It is also possible to fit \( k_{\text{air}}/\nu_{O_2} \) to experiments from the real plant. The achieved value can be explained from a physical point of view as already discussed in chapter 3.
When the reaction front reaches the surface of the grate the fire is extinguished. In the remaining section the present char will burn out. In fact this part of the theory is equal to the theory of Rogers [11], with the advantage that quantitative relations are given.

5.4.2 Total combustion regime

The total combustion regime always starts with the pipe-bowl combustion regime. Due to the preheated air the lower part of the waste pile will be dried. Then suddenly the process will ignite, which results in a complete combustion of the fuel layer. Two possible explanations for this sudden ignition can be given. The first explanation is that due to the grate movement, burning particles will fall through the waste pile to lower, already dried, parts where they will induce the ignition of the complete pile. It will depend on local circumstances (grate movement, temperature primary air, calorific value of the waste) if and when the ignition will occur. The second explanation concerns about the self-ignition behaviour. In the experiments with wood cubes and plastic spheres no autonomous ignition was found in practice, but with for example waste as fuel it is possible that self ignition can occur. This is because in waste the interfacial area is higher than in the performed experiments and probably in waste there are some materials present with a lower ignition temperature than the used wood-polyethylene mixture. From the moment of ignition the conversion rate is higher and the process conversion is accelerated compared to the situation before. Based upon the observations in real plants the total mass converted is dependent on the total amount of mass burning. Therefore, the same type of relation (5.1) can be used. The difference compared to the pipe-bowl regime lies in the fact that $M$ is the total burning mass on the grate, which is higher than in the situation without air preheating. The consumption of oxygen is the same as in the situation without air preheating. During both stages the oxygen is consumed in the gas phase by the volatiles and only if oxygen remains, char is converted. Further, char combustion and gasification take place after the fire has been extinguished.

Figure 5.5 summarises the above outlined different combustion regimes as a function of primary air temperatures and grate movement.

5.5 Application of the extended theory to observations

5.5.1 General observations

The occurrence in practice of the aforementioned combustion regimes depends heavily on the calorific value, the grate movement and the application of primary preheating. To explain the practical observations with the theory the combustion of solid fuels on a grate is divided into three cases: fuel with a low, moderate, and high calorific value. As the transition between the two combustion regimes (pipe-bowl and total combustion) is highly dependent on local circumstances (primary air preheating and grate movements) it is not possible to give exact values for low, moderate and high
calorific value. As a rough indication the following ranges can be given: low for values below 9 $MJkg^{-1}$, moderate for values between 9 $MJkg^{-1}$ and 12 $MJkg^{-1}$ and high for calorific values higher than 12 $MJkg^{-1}$.

**Combustion of fuel with low calorific value**

In the situation with a calorific value lower than 6 $MJkg^{-1}$ most system suppliers are using primary air preheating. The combustion normally starts with the pipe-bowl combustion. At the same time evaporation takes place in the lower part of the waste-pile. As a result the primary air flow is more or less saturated and is cooled down to the wet bulb temperature. The moisturised air flows through the upper part of the packed bed. At a certain moment, initiated by poking burning particles fall through the waste pile and can ignite the dried fuel in the lower part in the bed, causing a complete ignition. As a result the total mass on the grate is burning. In cases with no air preheating these particles will be extinguished by the cold air flow and the still wet fuel. From visual observations at real plants with higher calorific values and preheating it is known that the ignition point in that case is somewhere in the vicinity of the transition of the first and second grate section. In the case with low calorific values, this point will be further down the grate, probably somewhere in the third section.

**Combustion of fuel with moderate calorific values**

In this case (calorific values varies roughly between 9 $MJkg^{-1}$ and 12 $MJkg^{-1}$) it depends on the philosophy of the system supplier if primary air preheating is used, but mostly it is not applied. However, if primary air preheating is used, the process acts like the process with low calorific values as described above. The difference will be that the ignition point will be more close to the feeding system (end of the first
section. Without air preheating the process will combust like a pipe-bowl. At the moment the reaction front reaches the surface of the grate the fire is extinguished. In the remaining section the remaining char will burn out.

**Combustion of fuel with high calorific values**

In this case (calorific values are higher than 12 MJ kg⁻¹) no preheating is used. The process starts with the pipe-bowl regime and due to the more easy to ignite high calorific value, the fuel will ignite somewhere at the end of the first section of the grate. From that point there will be the complete combustion regime.

5.5.2 Observations from system identification

In this section the theory will be used to explain the phenomena observed in the real plants, as it is not so straightforward.

First, the situation with low primary air heat temperatures will be discussed. In this situation with relatively higher calorific values of the waste the situation is that a reaction front will propagate down through the waste pile. In this situation the front speed of evaporation is lower than in the situation with higher primary air temperatures. Therefore a total ignition situation will probably not be reached. From the model and the observations it was learnt that the reaction rate is linear to the amount of mass. This means that due to a step on the waste flow, the amount of mass converted in the top of the waste pile is increased. This increased conversion rate moves slowly down the grate, resulting in a gradual increase of the total mass burning. In the initial phase, the extra amount of waste has to be heated up, which is described by the model in chapter 3.

Secondly, there is the case with higher primary air temperatures. In this situation there are two combustion regimes: the pipe-bowl regime and the total combustion regime. Before total ignition takes place, an evaporation front propagates from upwards in the bed. At the same time the combustion front propagates downwards. Like the situation with lower primary air temperatures, the reaction rate is linearly proportional to the amount of mass. So, when a step is applied on the waste, a gradual increase of the amount of burning mass will also occur. But in this case an inverse response was found, which could only be explained by an increase of the moisture evaporation, compared to the situation before the applied step, see chapter 3. The inverse response cannot be explained by heating up the fresh new material.

However, as discussed before, the evaporation in the lower part of the bed is limited by heat transfer. The only explanation is that compared to the situation with no primary air preheating the calorific value will be lower. This implies that the fuel, compared to the case without preheating, is relative wetter and it takes more energy to heat up this more wet fuel. As a result the gas temperature in the gas phase is lowered and due to the radiation interaction the adjacent waste is influenced by this step, resulting in a shift down the grate of the breakthrough point. This results in a temporarily higher moisture evaporation compared to the combustion. After a while when the waste is dried completely and the two fronts meet each other, the total combustion regime will start.
The last point to be discussed is the application of the theory to the observations of Thunman [12], as the proposed theory is somewhat in contradiction to Thunman's theory. Thunman concluded that the observed phenomena can be explained by ignition of the fuel on the grate. This ignition is initiated by either heat transfer through the grate bars or by burning particles that are not transported along the grate by the grate movement. The first explanation cannot be the initiating mechanism as the same phenomena are also observed in incinerators with water-cooled grates. Also the second explanation is doubtful. If particles are not transported on the grate they will be lying on the grate bars. This implies that they will be cooled and ignition will become very problematic. With the theory presented in the former section it is possible to explain the by Thunman observed phenomena as will be outlined now.

Thunman used in his experiments wet wood (moisture content 30%) and the used primary air was saturated at 45 °C and further preheated up to 145 °C. Due to the very wet fuel the combustion rate front at the top is very low, resulting in bad combustion and a lot of smoke. At the same time the preheated air contains already a lot of moisture which has the result that from the wet fuel not much moisture can be evaporated. From the presented plug flow model (see Appendix B) it follows that with wood containing 50 (wt) % moisture and air preheating of 145 °C the evaporation speed is about 30% lower than the experiments presented in this chapter. But, the moisture in the air will condense at higher parts in the bed resulting in a decrease of the ignition rate with as possible consequence the generation of smoke. From the moment the total fuel layer will burn, so after ignition, it will be limited by mass transfer of oxygen which explains the relatively high amount of volatiles along the grate up to the fourth section.

Consequences for solid fuel combustion on a grate

From this research some suggestions can be given for practical use in MSWC or biomass combustion on a grate:

- From the results it follows that the total combustion time can be reduced by using primary air preheating. This implies that smaller grates can be used. A disadvantage can be the higher thermal load on the grate, which can cause an increase in damage. Therefore, a well balanced integration with water-cooled grates can result in more compact furnace designs.

- When primary air preheating is used, the application should be limited to the first sections until the breakthrough of the reaction. Energy can be saved and maybe damages to the grate in later sections, due to locally too high thermal loads, can be avoided. In practice it is difficult to determine the place where the breakthrough takes place. It can often be seen by looking to places where the grate is damaged. In general primary air preheating in the first two zones would be sufficient. Maybe primary air preheating in the first zone may help to reduce the temperature at the top of the first zone, because of the amount of water that is heated up and evaporated. Furthermore, it can be considered
to use preheating in the last zone in order to achieve a good burn-out of the bottom-ash.

5.6 Conclusions

In this chapter, the present theory for the combustion of solid fuels has been extended in order to explain the observed phenomena with preheated air. The combustion process without preheated air can be described with the classical theory proposed by Rogers. The extension of this theory with the influence of preheated air provides an explanation for all observed phenomena. The main point of the theory is that preheating of the primary air acts as a catalyst for the ignition on a grate rather than merely drying the waste. A combination of a water-cooled grate with efficient primary air preheating can result in smaller and more compact grate systems.

Furthermore, it has been shown that the reaction rate for the conversion of waste on a grate is dependent on the amount of mass. Different primary air temperatures do not influence this dependency.

An important outcome of this study is that pot furnace experiments have only a limited value in studying grate furnace combustion. The translation from the plug flow behaviour into a batch type reactor is limited by the fact that in the real plant the grate movements have a large impact on the combustion behaviour. This effect is not simulated in the pot furnace experiments. For that reason the translation of results of pot furnace experiments into practical solutions should be made with care.
Appendix A

Radiation in thermal processes

In this appendix the general radiation term in equation (3.3) will be derived. In thermal processes, radiation can play an important role. It will be shown that the equations in the former section will also hold also when the radiation is the main energy transfer mechanism. When convective heat transfer is the dominant heat transfer mechanism the aforementioned equations are simplified. And for example in fluid bed systems, because of the ideal mixing between the solid and gas phase there is no net heat exchange between these two phases, resulting in one energy balance instead of two. In fact this is a further simplification of the general system. But, consider for example a waste incineration process (see figure A.1). Waste is fed into a furnace by a ram feeder, burning during its travel along a moving grate. Remaining bottom-ash leaves the system for disposal and the flue gases are cleaned before they are emitted to the air. In this process there is a clear separation between the gas

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*Figure A.1: Diagram of a municipal solid waste incinerator.*
and solid phase. Air from under the grate will flow through the waste layer and react partially with the gases which come from the waste layer. The fuel gases enter the gas phase above the waste layer where secondary air is used to achieve a good mixing. In this highly turbulent gas phase the fuel gases will be converted to CO₂ and H₂O. As a result the temperature in this gas phase will be higher (1200°C) than the waste layer temperature (800—900°C). Because of these high temperatures radiation becomes an important heat transfer mechanism.

For many industrial applications the process can be described by a well stirred reaction chamber. For these type of reactors Hottel [37] derived some useful formulas which can easily be used in engineering applications. The main assumptions for this radiation model are:

- the gas is perfectly mixed and has a single temperature of $T_\text{g}$;
- the gas is gray;
- the walls are speckled, that means that from any point of the walls the view factor to the sink area is the same. This is only possible when the sink and the refractory surfaces are intimately mixed;
- the surface of heat sink, of area $A_\text{s}$, is gray and can be assigned a single temperature $T_\text{s}$;
- external loss through the wall is negligible.

Some assumptions seem very theoretic, but results have shown that this concept gives fair good results. In this model the flux from the gas to the heat sink is:

$$q = A\sigma(T_\text{g}^4 - T_\text{s}^4)$$  \hspace{1cm} (A.1)

with $A$:

$$A = \frac{A_\text{T}}{\frac{1}{\epsilon_\text{g}} + \frac{1}{\epsilon_\text{w}} - 1}$$

where $C_\text{s}$ is the cold or sink fraction of the total furnace envelope of area $A_\text{T}$; i.e. $A_\text{s} = C_\text{s}A_\text{T}$; $A_\text{v} = (1 - C_\text{s})A_\text{T}$.

For non gray gases this can be extended to:

$$q = \sigma A_\text{s}(\epsilon_\text{g}^*T_\text{g}^4 - \alpha_\text{g}^*T_\text{s}^4)$$  \hspace{1cm} (A.2)

where $\epsilon_\text{g}^* = \frac{1}{\frac{1}{\epsilon_\text{g}} - 1 + \frac{1}{\epsilon_\text{w}}}$ and $\alpha_\text{g}^* = \frac{1}{\frac{1}{\epsilon_\text{g}} - 1 + \frac{1}{\epsilon_\text{w}}}$ where the absorptivity $\alpha_\text{g} = (T_\text{g}/T_\text{s})^n\epsilon_\text{g}$.

The emissivity $\epsilon_\text{g}$ and the absorptivity for the most important gases (e.g. CO₂, H₂O) are given in literature. However, beside gas radiation, ash, soot and dust radiation can have an important contribution. The emissivity of gases containing water vapour and carbon dioxide is mostly below 0.2, while the values for soot and dust can be up to 0.6 depending on the dust/soot concentration [38]. The overall emissivity for gray gas and gray particle radiation can be found with the following relation [38]:

$$\epsilon_{\text{gt}} = 1 - (1 - (\epsilon_\text{g})^*)(1 - \epsilon_{\text{dust}})(1 - \epsilon_{\text{soot}})$$  \hspace{1cm} (A.3)
and for the absorptivity for real gases and gray dust and soot radiation:

\[ \alpha_{gt} = 1 - (1 - \alpha_g^*) \left( 1 - e_{dust} \right) \left( 1 - e_{soot} \right) \]  \hspace{1cm} (A.4)

As in solid fuel flames, soot and dust are the major contributions to the emissivity and \( \alpha_g \) is only about 20 percent higher (the power n is in the range of 0.45 (H\(_2\)O) -0.65 (C\(_2\)O) [38]) compared to \( \epsilon_g \) it is justified to take the absorptivity equal to the emissivity.

In conclusion, for the most general situation a general formulation is found to describe the heat exchange by radiation:

\[ q = \sigma A_s (\epsilon_g T_g^4 - \alpha_{gt} T_s^4) \]  \hspace{1cm} (A.5)

However this strongly non-linear term can be linearised in the following way:

\[ q = \sigma A \left( (\sqrt[3]{\epsilon_g} T_g^4 - (\sqrt[3]{\alpha_{gt}}) T_s^4 \right) = \sigma A (T_g^{*4} - T_s^{*4}) \]

\[ = 4\sigma A T_{av}^{*3} (T_g^{*} - T_s^{*}) = \alpha A (T_g^{*} - T_s^{*}) \]

with \( T_{av} \) the average of \( T_g \) and \( T_s \). This is similar to the relation used in equation (3.3):

\[ q = A \alpha (\epsilon_g T_g - \alpha_{gt} T_s) \]  \hspace{1cm} (A.6)

where

\[ \alpha = 4\sigma \left( \frac{\epsilon_g T_g + \alpha_{gt} T_s}{2} \right)^3 \]

\[ A = A_s \]

\[ \epsilon_g = \sqrt[3]{\epsilon_{gt}} \]

\[ \alpha_{gt} = \sqrt[3]{\alpha_{gt}} \]  \hspace{1cm} (A.7)

The error made by this linearisation is under normal conditions (\( | T_g - T_s | < 500K \) and \( T_s > 1000K \)) in thermal processes always less than 5%. Above was already mentioned that in solid fuel flames \( \epsilon_{gt} = \alpha_{gt} \). In that case the relation 3.3 simplifies to:

\[ q = A \alpha \epsilon_g (T_g - T_s) \]  \hspace{1cm} (A.8)
Appendix B

Evaporation of moisture in a packed bed

The evaporation of moisture in a packed bed due to air flowing through the packed bed can be described by the mass and energy balances. A wet particle consists of two mass parts: a dry particle \(m_{dp}\) and a water part \(m_w\). As the time constant of the gas phase is fast compared to the time constants of the solid phase the gas phase will be described stationary. The bed is discretized over the height in pieces with length \(dz\). The derived mass balances along the length \(dz\) of the packed bed are:

\[
\frac{dm_w}{dt} = -\Phi_{evap} \tag{B.1}
\]

with \(\Phi_{evap}\) in \(kg\ s^{-1}\).

\[
\frac{dm_{dp}}{dt} = 0 \tag{B.2}
\]

The latter is zero as the particles are not transported along the height. The stationary gas phase balance is:

\[
\Phi_{dG\gamma} = -\Phi_{evap} \tag{B.3}
\]

with \(\gamma\) the fraction \((kg\ kg^{-1})\) moisture in the gas phase and \(\Phi_{dG}\) the moisture free gas flow \((kg\ s^{-1})\). For the energy balance the following relations along the height hold:

For the solid phase

\[
(m_wC_{p,w} + m_{dp}C_{p,dp}) \frac{dT_p}{dt} = \alpha_p A_p(T_g - T_p) - H_{evap}\Phi_{evap} \tag{B.4}
\]

with \(H_{evap}\) the evaporation enthalpy \((J\ kg^{-1})\). And for the gas phase

\[
\Phi_{dG\gamma}C_{p,\gamma}dT_\gamma + \Phi_{dG\gamma}C_{p,\alpha}dT_\alpha = \alpha_p A_p(T_g - T_p) \tag{B.5}
\]

There are different methods known in literature to describe the evaporation rate. Here, it is chosen to follow the method described by Gnielinski [70]. In this method the evaporation rate \(\Phi_{evap}\) is described with the following relation:

\[
\Phi_{evap} = A_p\rho_{\gamma}\beta(Y^* - Y)\nu \tag{B.6}
\]
with $A_p$ the interfacial area in $m^2$

$$A_p = \frac{6}{d_p}A_d z$$

with $A$ the cross sectional area ($m^2$), $d_p$ the diameter of the particles in the packed bed $(m)$ and $dz$ the discretization along the height $(m)$ . The dimensionless saturation value is defined as follows:

$$Y^* = \frac{M_w}{M_{air}P - p_w}$$

with the saturation partial pressure

$$p_w = e^{23.6068 - \frac{2}{(23.6068 - \frac{23.6068}{T_{sat} - T_{sat,0}})}}$$

and $M_w$ the molmass of water and $M_{air}$ the molmass of air. The dimensionless factor $\nu$ is used to describe the drying process. In the drying process two phases can be distinguished. The first is the constant falling rate where free water is evaporated. 

This evaporation will be linear and the decrease in the moisture content will also be linear. After a certain moment bound water will evaporate. This moment is determined by the critical moisture content of the substance. This second stage is called the first falling rate. The water in the cell structure will more difficult evaporate than the free water. For that reason in the first falling rate the evaporation rate will be non-linear. From this moment, indicated by the hygroscopic water content $X_{hy}$ ($kg/kg^{-1}$) there will be no free water available in the substance. The factor $\nu$ in the equations above, takes these effects into account and the factor $\nu$ is described as follows:

If $X < X_{cr}$

$$\nu = \frac{X - X_{hy}}{X_{cr} - X_{hy}}$$

If $X > X_{cr}$

$$\nu = 1$$

The mass and heat transfer coefficients $\beta$ and $\alpha$ ($m s^{-1}$) follows from the classical Nusselt and Sherwood relations:

$$Sh = \frac{\beta d_p}{D} = 2 + 0.6Re^{0.5}Sc^{0.33}$$

and

$$Nu = \frac{\alpha d_p}{D} = 2 + 0.6Re^{0.5}Pr^{0.33}$$

The boundary conditions for the model are straightforward:

$$z = 0 \Rightarrow Y = Y_{air} \wedge T = T_{air} \quad (B.9)$$

$$t = 0 \Rightarrow Y(z) = Y_{air} \wedge T(z) = T_{air} \wedge T_s(z) = T_{s0} \quad (B.10)$$

This model is solved numerically. The evaporation rate is calculated by dividing a certain height $h$ by the time that the temperature of the solid material has the same temperature as the inlet temperature of the preheated air. The time is defined as the time that has elapsed from the beginning of the experiment.
Appendix C

Determination of stationarity of random data

In section 4.2.1 the method of reverse arrangements has been presented shortly. This method can be used to determine the stationarity of random data. Normally this method is used to verify a hypothesis on a α level of significance. Here, the method is extended in such a way that an exact value for the rate of stationarity can be given. Here an outline of the method will be given as well an application to different signals to get some feeling for the results.

Stationarity is investigated with the following procedure [48]:

- Divide the sample record in N equal time periods \( x_i(t) \) where the data in each interval may be considered independent.

- Compute a mean square value (or the mean value and the variance) for each period.

- Test the sequence of mean square values for the presence of underlying trends other than those due to expected sampling variations.

- Use the non-parametric reverse arrangement test to test the stationarity.

The advantage of the reverse arrangement method is that this testing procedure does not require a knowledge of either the frequency bandwidth of the data or the averaging time used to compute the mean square values. The only assumption for this method is that the fundamental period is short compared to the averaging time used to compute the sample. The reverse arrangement test can be outlined shortly as follows [48]. Count the number of times that \( x_i > x_j \) for \( i < j \) for \( i = 1, 2, 3, ..., N \). The total number of reverse arrangements is denoted by \( A \), where \( A \) is defined by:

\[
A = \sum_{i=1}^{N-1} A_i
\]  

(C.1)
where
\[ A_l = \sum_{j=i+1}^{N} h_{ij} \]  \hspace{1cm} (C.2)

and
\[ h_{ij} = \begin{cases} 
1 & \text{if } x_i > x_j \\
0 & \text{otherwise}
\end{cases} \]  \hspace{1cm} (C.3)

It is hypothesised that the sequences of mean square values (or mean values and variances) represent independent sample measurements of a stationary random variable. If this hypothesis is true, the variations in the sequence of sample values will be random and display no trends. Hence the number of reverse arrangements will be a random variable with a mean value \( \mu_A \) and variance \( \sigma_A \) [48]:
\[ \mu_A = \frac{N(N - 1)}{4} \]  \hspace{1cm} (C.4)
\[ \sigma_A = \frac{N(2N + 5)(N - 1)}{72} \]  \hspace{1cm} (C.5)

and the standardised value of \( A \), \( z_A \), is given by:
\[ z_A = \frac{A - \mu_A}{\sigma_A} \]  \hspace{1cm} (C.6)

To achieve an absolute value for the stationarity the method is extended. The results of the analysis are compared with an analysis from white noise signals, which is known to be stationary. The probability distribution for \( \mu_A \) as well \( \sigma_A \) are compared with the probability distribution of those values for white noise, which theoretically should be stationary. The overlap in area of both density functions is defined as a measure for the stationarity. Complete overlap with the theoretical distribution means a stationarity of 1 and no overlap gives 0. In order to achieve a better feeling for the concept of stationarity analysis on artificially produced data will be discussed now.

**Stationarity of white noise**

First, the stationarity of pure white noise is investigated. The probability distribution of \( z_A \) is determined by dividing the total sample record in \( N \) periods. For each period \( z_A \) is determined by splitting that period in \( M \) intervals. First, the influence of the number of intervals \( M \) is investigated. The standardised total number of reverse arrangement \( z_A \) for the mean value of 5 different white noise signals \((100000\) data points generated by Matlab) is determined as a function of the amount of time intervals \( M \). The results are given in Figure C.1. This figure shows that \( z_A \) differs a lot between the different signals. The influence of \( M \) is far less. This indicates that the chosen amount of intervals for the stationarity test does not influence the result a lot. For that reason, in the analysis the In the analysis the number of intervals, \( M \), is chosen to be 25.
Figure C.1: $z_A$ as a function of the number of subintervals $M$ for white noise.

Following that, $z_A$ has been determined for the mean value and variance of 10000 different white noise signals (each containing 10000 data points) with $M=25$ (so each interval contains 400 points). The distribution of these values and the normal distribution are given in figure C.2. This figure shows clearly that the stationarity of white noise has a normal distribution with a mean value of $\mu_A$ and a standard deviation of $\sigma_A$.

Figure C.2: Stationary distribution of white noise. $N(0,1)$ is a Gaussian distribution with zero mean and a variance of 1.
Stationarity of trended "white noise"

A white noise signal shows no trends in the variance and the mean value. However, in order to get a better understanding of the stationarity concept it is useful to investigate the influences of a trend in a further white noise like signal on the stationarity rate. First, based upon a white noise signals a trend in the mean value is introduced and secondly a trend in the variance is introduced.

First, the standardised total number of reverse arrangement ($z_A$) for the mean value of 5 different white noise signals (10000 data points) is determined as a function of the amount of time intervals ($M$). The results are given in figure C.3. This figure shows that $z_A$ differs a lot between these different signals also. The influence of $M$ is far less. So the same conclusion as for white noise can be given for white noise with a trend; the chosen amount of time intervals for the stationarity test does not influence the result a lot.

$z_A$ has been determined for the mean value and variance of 10000 different white noise signals with a trend in the mean value (10000 data points) with $M=25$. The variance of the signal is constant. The distribution of these values and the normal distribution are given in figure C.4. This figure shows clearly that the stationarity of white noise with a trend has a normal distribution, but with a different mean value of $\mu_A$ or a standard deviation of $\sigma_A$ as for white noise without a trend.

Stationarity rating

It is concluded that the stationarity of a process can be determined based on the standardised distribution of the total number of reverse arrangements. To calculate this distribution a sample record needs to be divided into an amount of equal intervals. The stationarity of these individual intervals is determined with a fixed value of $M$.
Figure C.4: Stationary distribution of white noise.

A rating of the stationarity is possible by calculation of the combined surfaces of this distribution and the $N(0,1)$ distribution of with noise, as demonstrated in figure C.5. This rate varies from 0 for non-stationary signals to 1 for stationary signals. It is possible to vary the length of the intervals in which the sample record is divided.

Figure C.5: Combined surface of $N(0,1)$ distribution and the $N(1,2)$ distribution.

This procedure reveals the periodicity of the stationarity. Figures C.6-C.9 show the periodicity of the stationarity of different signals. The high values of stationarity at low time intervals for the mean value in figure C.9 is because the interval which is used to determine the mean value is so small that no trends can be distinguished. The same holds for the high stationarity values for the variance in figure C.8.
Figure C.6: Stationarity rate of mean value and variance of white noise.

Figure C.7: Stationarity rate of mean value and variance of low pass filtered white noise.
Figure C.8: Stationarity rate of mean value and variance of "white noise" with a trend in the variance.

Figure C.9: Stationarity rate of mean value and variance of "white noise" with a trend in the mean value.
Appendix D

Sensitivity analysis

Figure D.1: Frequency spectrum and frequency dependent standard deviation of $T_\beta$ as a function of variations in the parameter $X_{\text{inert}}$. 
Figure D.2: Frequency spectrum and frequency dependent standard deviation of $T_g$ as a function of variations in the parameter $\Delta H$.

Figure D.3: Frequency spectrum and frequency dependent standard deviation of $T_g$ as a function of variations in the parameter $X_{air}$. 
Figure D.4: Frequency spectrum and frequency dependent standard deviation of $T_3$ as a function of variations in the parameter $A$.

Figure D.5: Frequency spectrum and frequency dependent standard deviation of $T_3$ as a function of variations in the parameter $a$. 
Figure D.6: Frequency spectrum and frequency dependent standard deviation of $T_g$ as a function of variations in the parameter $\Phi_{in}$.

Figure D.7: Frequency spectrum and frequency dependent standard deviation of $T_g$ as a function of variations in the parameter $\Phi_{prim}$. 
Figure D.8: Frequency spectrum and frequency dependent standard deviation of $T_g$ as a function of variations in the parameter $k_d$. 
Nomenclature

Roman symbols

- \( A \) burning area, \( \text{m}^2 \)
- \( A \) total number of reverse arrangements, \( - \)
- \( a \) interfacial surface area, \( \text{m}^2 \text{kg}^{-1} \)
- \( A_{\text{boiler}} \) lumped boiler parameter, \( \text{Js} \text{kg}^{-1} \)
- \( C_p \) specific heat capacity, \( \text{J} \text{kg}^{-1} \text{K}^{-1} \)
- \( D \) diffusion coefficient, \( \text{m}^2 \text{s}^{-1} \)
- \( d_p \) particle diameter, \( \text{m} \)
- \( e \) dimensionless parameter, \( - \)
- \( E_a \) activation energy, \( \text{J} \text{mol}^{-1} \text{K}^{-1} \)
- \( f \) frequency, \( \text{s}^{-1} \)
- \( H \) enthalpy, \( \text{J} \text{kg}^{-1} \)
- \( k \) corrosion rate, \( \text{mm} \text{h}^{-1} \text{or mm}^2 \text{h}^{-1} \)
- \( k_0 \) pre-exponential constant, \( \text{m} \text{s}^{-1} \)
- \( k_d \) mass transfer coefficient, \( \text{m} \text{s}^{-1} \)
- \( k_{\text{evap}} \) evaporation rate, \( \text{kg m}^{-2} \text{s}^{-1} \)
- \( M \) mass, \( \text{kg} \)
- \( M \) molecular mass, \( \text{kg mol}^{-1} \)
- \( m \) metal mass boiler system, \( \text{kg} \)
- \([O_2]\) oxygen concentration, \( \text{mol m}^{-3} \)
- \( p \) boiler pressure, \( \text{Pa} \)
- \( Q \) heat flux, \( \text{W} \)
- \( R \) reaction rate, \( \text{kg m}^{-2} \text{s}^{-1} \)
- \( R_g \) universal gas constant, \( \text{mol}^{-1} \text{K}^{-1} \)
- \( RE \) temperature derivative of reaction rate, \( \text{J} \text{mol}^{-1} \text{K}^{-2} \)
- \( Re \) Reynolds number, \( - \)
- \( Sc \) Schmidt number, \( - \)
- \( Sh \) Sherwood number, \( - \)
- \( T \) temperature, \( \text{K} \)
- \( t \) time, \( \text{s} \)
- \( u(t) \) input variables as a function of time, \( \text{cm} \text{min}^{-1} \)
- \( U_{\text{evap}} \) evaporation rate, \( \text{kg m}^{-2} \text{s}^{-1} \)
- \( U_I \) ignition rate, \( \text{kg m}^{-2} \text{s}^{-1} \)
NOMENCLATURE

\( U_g \) superficial air velocity \( \text{ms}^{-1} \)

\( U_v \) combustion rate \( \text{kg m}^{-2} \text{s}^{-1} \)

\( V \) acceleration factor \( \text{m}^3 \)

\( v \) velocity \( \text{ms}^{-1} \)

\( X \) mass fraction \( \text{kg kg}^{-1} \)

\( X_{\Delta H} \) energy distribution factor

\( x \) penetration depth \( 10^{-3} \text{m} \)

\( X_{\text{air}} \) air channelling factor

\( X_{\text{hyg}} \) hygroscopic water content \( \text{kg kg}^{-1} \)

\( Y \) moisture fraction in gas phase \( \text{kg kg}^{-1} \)

\( Y \) volume fraction \( \text{m}^3 \text{m}^{-3} \)

\( y(t) \) output variables as a function of time

\( z \) length \( \text{m} \)

Greek symbols

\( \alpha \) absorptivity

\( \alpha \) constant

\( \alpha \) heat transfer coefficient \( \text{ms}^{-1} \)

\( \alpha \) parameter used in radiation equation

\( \beta \) mass transfer coefficient \( \text{ms}^{-1} \)

\( \Delta H \) reaction enthalpy \( \text{Jkg}^{-1} \text{ or MJkg}^{-1} \)

\( \epsilon \) emissivity

\( \eta \) dynamic viscosity \( \text{kg m}^{-1} \text{s}^{-2} \)

\( \eta \) efficiency parameter

\( \lambda \) air factor

\( \nu \) dimensionless parameter to describe drying

\( \nu_{O2} \) stoichiometric factor \( \text{kgO₂ kgwaste}^{-1} \)

\( \omega \) frequency

\( \Phi \) mass flow

\( \phi \) molar flow

\( \rho \) density \( \text{kg m}^{-3} \)

\( \tau \) time constant \( \text{s} \)

\( \zeta \) damping factor

Subscripts

\( ac \) active

\( b \) black

\( band \) frequency bandwith

\( bt \) break through

\( c \) condensation
comb  combustible
cr   critical
da   dry air
dp   dry particle
e    emitted
evap evaporation
fd   feed
fg   flue gas
g    gas
max maximum
moist moisture
o    reference state
p    particle
pas  passive
prim primary air
recirc recirculation
s    solid
sec  secondary air
st   steam
T    total surface
th   theoretical
tot  total
trans transferred
v    water vapour
w    water
y    H/C ratio in CH\textsubscript{y}O\textsubscript{z}
z    O/C ratio in CH\textsubscript{y}O\textsubscript{z}

Stochastic variables

\begin{itemize}
\item \( E \) expectation value
\item \( M_n \) n-order moment of probability function
\item \( N \) number of samples
\item \( N(a, b) \) normal (Gaussian) distribution, with mean \( a \) and variance \( b \)
\item \( M \) time interval
\item \( N \) number of samples
\item \( P \) probability function
\item \( R \) autocorrelation function
\item \( S \) spectral density function
\item \( \mu \) mean value
\item \( \sigma \) standard deviation
\item \( \sigma^2 \) variance
\item \( M_n \) n-order moment of probability function
\item \( P \) probability function
\end{itemize}
$R$  autocorrelation function  
$S$  spectral density function  
$\mu$  mean value  
$\sigma$  standard deviation  
$\sigma^2$  variance
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List of publications

**Patents**

System for determining process parameters relating to thermal processes such as, for instance, waste incineration. EP 00937379.6 (pending)

Control system for an incineration plant, such as for instance a refuse incineration plant. EP 00971884.2 (pending)

System for continuous thermal combustion of matter, such as waste. EP 00971885.9 (pending)

System for continuous thermal combustion of matter, such as waste matter. EP 01912580.6 (pending)

**Publications**


Conference publications


Curriculum Vitae

Lambertus Bernardus Maria (Robert) van Kessel was born on February 12, 1967, in Brunssum, The Netherlands.

1978 - 1984 : He attended the Romboutscollege in Brunssum, where he received his Atheneum-B degree.

1984 - 1989 : At the Eindhoven Technical University he studied Applied Physics. His final research project was in the field of semiconductor physics.

1989 - 1991 : He followed the 2-year course process development at the Twente University. During the last year of this course, he was involved in two projects: optimisation of the polypropylene process and the optimisation of the pyrolysis process of Freon-22. In both studies mathematical modelling techniques were applied.

1992 - present: He has worked for TNO, Environment, Energy and Process Innovation. During these years his main activities were related to municipal solid waste combustion. Besides the work described in this thesis he has carried out work on the following subjects: advanced control concepts, development of a validated real-time simulator of the combustion process and general consultancy activities. Other areas in which he was active: DeNOx-technology (consultancy activities and process design), process control of thermal conversion processes, simulation technology (development of products for training and optimisation) and research activities in CO2-removal. At present he is co-ordinator of the process simulation activities in the Department of Energy Systems and deputy head of the department.