Experiments on a woodfired bakery oven

Faculty of Physics

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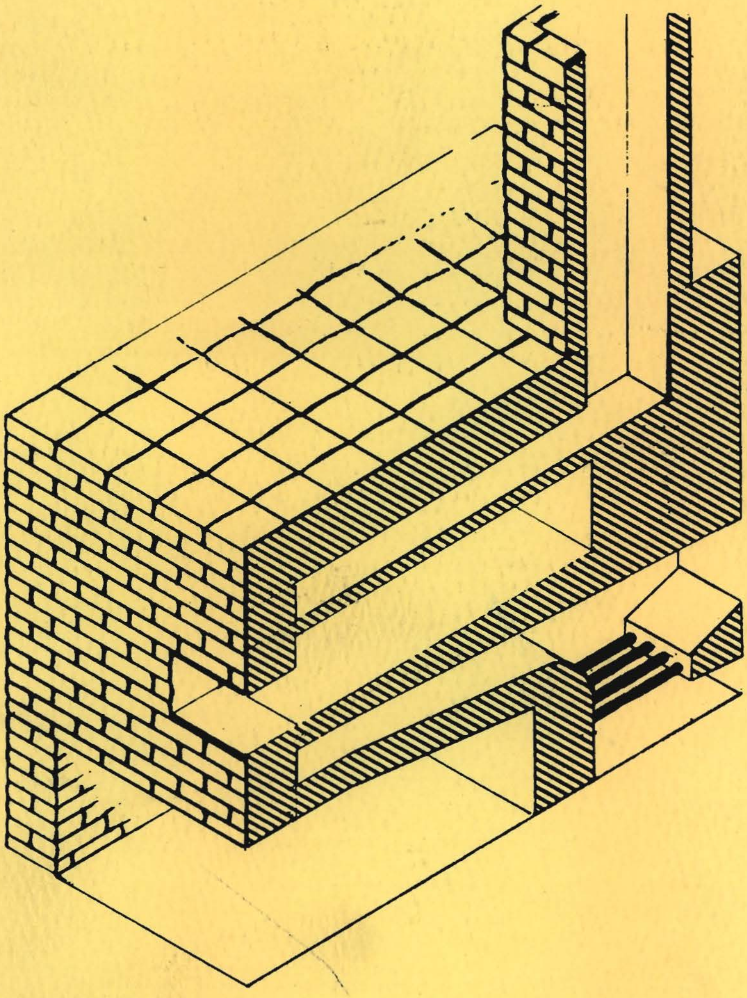

A report from
The Woodburning Stove Group
Eindhoven University of Technology, Eindhoven
July 1990

10 years of research on woodburning technologies

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Houtkachelgroep

Eindhoven
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The Netherlands

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Contents

Preface

" What is the efficiency of an open fire? "

This apparently simple question triggered the establishment of the Woodburning Stove Group (WSG) at the Eindhoven University of Technology a little over ten years back. However the answer to the question has turned out to be much more complex and as was anticipated in the introduction to the first report from the group, " ..... it could not be provided in terms of a single number. " (Krishna Prasad, 1980). Yes ! the group's research effort has resulted in an answer that is made up of a great deal of numbers. Some of these numbers have culminated in engineering designs, a few of which are being produced and used for cooking in many developing countries of the world.

WSG officially came into existence with a project funded by the Dutch Minister for Development Cooperation in 1980. The proposal of the project was prepared in 1979, the year of the now famous (or should one say infamous) second oil shock. Thus the emphasis was on energy and its conservation. Since the household cooking was the principal consumer of woodenergy, the woodburning cookstove was the primary focus of the group's work in the early years. With changing times, the group has enlarged its repertoire of activities, to include other users of wood (as exemplified by the present report), kerosene stoves (to take into account the so–called fuel substitution in the policy maker's jargon) and clean combustion (as a means to promote indoor air quality). Over the years students from this university and other higher educational institutes in Holland, as well as research workers from developing countries have contributed to the research output of the group.

From its inception, a concern of central importance for the sustenance of the group's work has been the translation of research results into hardware in users' kitchens. This was sought to be achieved in diverse ways. Members of the group have provided technical assistance to stove projects in many developing countries through direct participation in design, production and testing in laboratories as well as in users' kitchens. Much of this work has been written up as contract reports and in principle can be obtained only from the sponsors. A second category of communication of research results is through more conventional channels – technical journals and conference presentations – which are usually very abbreviated reports of the work.

To overcome the limitations of the above two categories of publications, the group puts out its own reports from time to time, which are much more elaborate, covering as they do, detailed descriptions of measurement techniques and analysis procedures. The present report belongs to this last category. Other reports in this series are listed at the end of this report and can be made available to anybody interested, for a small fee to cover the copying costs. A complete list of publications of the group can be obtained by writing to the group.

Finally the group would be happy to answer any queries about woodburning technologies in general (as far as that is feasible) and our current concerns in particular.

Nomenclature

SYMBOL	QUANTITY	SI UNIT
a	Ash content	–
A	Area	m^2
B_c	Calorific value of charcoal	kJ/kg
B_e	Baking efficiency	–
B_f	As–fired calorific value	J/kg
B_0	Gross calorific value of the fuel determined by a conventional bomb calorimeter in a laboratory	J/kg
C	Carbon content of the fuel	–
Cp	Specific heat capacity at constant pressure	J/kgK
Δ	Difference	–
H	Hydrogen content of the dry fuel	–
H_s	Sensible heat losses through the chimney	–
λ	Excess air factor	–
m	Moisture content on dry basis	–
M	Mass	kg
M_f	Fuel consumption	kg/s
N	Number	–
ν	Fraction of wood converted into volatiles	–
O	Oxygen content of the fuel	–
O_e	Oven load efficiency	kg flour/m^2.h
O_f	Oven load factor	kg flour/m^2
p	% carbon in the wood	–
P	Power output of the fire	kW
P_n	Nominal power output	kW
$\bar{P}$	Average power output	kW
q_{co}	Heat losses due to incomplete combustion	J/s
q_r	Radiative heat flux from the flames	J/s

ρ	Density	kg/m^3
S_c	Specific energy consumption	MJ/kg flour
t	Time	s
T	Temperature	K
V	Volume	m^3
V_{st}	Stoichiometric amount of air	m^3/kg wood
y	Difference between oxygen and 8 times the hydrogen content in the wood	–

Subscripts

a	ambient
b	baking including loading and unloading/ one baked loaf
c	ceiling/charcoal
C	center position
dw	dry wood
f	fuel/flame
fl	flour per loaf/floor
g_{ch}	flue gas temperature at the chimney entrance
h	heating up
l_b	loaves per batch
LB	left back position
LF	left front position
m	laboratory model
p	prototype
RB	right back position
RF	right front position
st	stoichiometric
T	total
T_g	flue gas temperature
T_l	total number of baked loaves
v	volatiles
w	wood
ww	wet wood

1 Introduction

Previous work

The previous report on woodfired bread ovens (Schutte et al., 1988) covered a number of mathematical models. These models were developed to contribute to the design of improved indirectly woodfired bakery ovens. "Improved" in this context means ovens that produce better quality of bread and are more fuel efficient than the presently used woodburning bakery ovens in developing countries. The intention of the mathematical models is to use them for making reasonably accurate predictions on heat transfer characteristics and design parameters for an arbitrary sized oven, without having to do a large number of experiments. There is however an inevitable uncertainty about using these mathematical models, since they only approximate the real situation. To improve the accuracy (reliability) of these models, they need to be compared with experimental results and be adapted where necessary.

Because the report on mathematical modelling of a woodburning bakery oven was published in 1988, it seems useful to start the present report with a short summary of the previous report, without going into too much detail. The oven design discussed here is to be used by small scale village bakeries in several developing countries. It has a baking capacity of 55 breads of 500 g every baking cycle. For better understanding of the mathematical models, the process of the bread manufacture is described. Because in most developing countries, modern machines are either not available or too expensive, the bread is to be manufactured through the so–called traditional process of bread manufacture. The general steps involved are described below. Depending on the type of dough manufactured, the production process can be partially different.

i) Mixing and kneading of the raw materials. Initial formation of the characteristic gluten skeleton of dough.
ii) First period of rising.
iii) Dividing and puffing up of the dough pieces.
iv) Second period of rising.
v) Preparation of the dough pieces for the final period of rising.
vi) Third and final period of rising.
vii) Baking of the bread.

Steps ii till vi are required to obtain the proper shape, dough structure, specific bread flavours and volume. The final step, baking of the bread, is the most complex one, because many chemical and physical processes are involved. During baking, the bread pieces obtain there final volume, texture, color and taste. For dough pieces of 550 g, the baking compartment gas temperature should be about 240^{0}C and the minimum baking time required is about 10 minutes.

The first mathematical model describes a method to compute the required amount of heat that has to be transferred from the bottom and top flues to the baking compartment, in order to bake the bread and balance the heat losses. This method assumes that the baking compartment floor and ceiling temperature remain constant during the baking process. The heat transfer mechanism between the floor and the bottom of the baking pans as well as between the ceiling and the top of the breads is represented by heat transfer between horizontal parallel plates. In both cases the heat transfer appears to take place by a combination of conduction and radiation. Heat transfer by natural convection is excluded because (a) in the horizontal orientation between the ceiling and the top of the breads, the gravity is in the direction opposite to the temperature gradient, and (b) between the floor of the baking compartment and the bottom of the baking pans, the Rayleigh number, which gives the ratio of free energy liberated by buoyancy to the energy dissipated by heat conduction and viscous drag (Schinkel, 1980) is smaller than the critical value of 1708. However the calculations showed that most of the heat between the floor and the bottom of the pans is transferred by conduction. This appears to depend much on the distance assumed between the floor and the bottom of the baking pans. Most of the heat transferred between the ceiling and the top of the breads takes place by radiation. Due to the assumptions made for this model, approximately 85% of the heat required is supplied through the baking compartment floor. The heat losses through the walls and door of the baking compartment are less than 2% of the total amount of heat transferred to the baking compartment.

In practice, there will be a certain drop in floor and ceiling temperature during the first few minutes of the baking process due to the large absorption of heat by the relatively cold dough pieces. It is impossible to supply enough heat to maintain constant floor and ceiling temperatures. This means that part of the total heat supplied, necessary to bake the bread, derives from the heat that has been accumulated in the floor and ceiling during the heating up phase of the oven. The second model takes into account this drop in floor and ceiling temperature, and it is assumed that these temperature drops are uniform. To obtain a uniform and equal temperature drop of the baking compartment floor and ceiling, the amounts of heat supplied by the floor and ceiling and thus the masses of the floor and ceiling, are to be related to the amounts of heat transferred through the floor and ceiling (which were different as computed with the first model). Due to this initial drop in floor and ceiling temperature, the baking times increase by about 25%.

The accumulated heat in the outside walls of the oven is lost to the surroundings when the baking process is stopped and the oven starts cooling down. The amount of heat that is lost to the surroundings depends among other things on the thickness of the walls, the composition of the walls, the cooling periods and the presence of a cold air flow going through the flues. The amount of retained heat after a cooling down period, directly affects the required amount of fuel necessary to bring back the oven to its baking temperature. Since the use of a thicker wall not necessarily means a reduction in energy consumption (after all a thicker wall requires more energy to heat up), it seems useful to determine the balance between the wall thickness (energy saved by insulation), the time of use (loss of accumulated heat to the surroundings during the time the oven is not used), and the costs for the wall. An attempt to do this was made with the third mathematical model. The calculations showed that for a given cooling period, the retained amount of heat in the side walls increases with an increase of the wall thickness, thus confirming the impression that a larger wall thickness reduces the energy consumption. However the absolute amounts of heat that are lost to the surroundings for the different wall thicknesses show a completely different picture. It turns out that, independent of the cooling period, there appears to be a wall thickness at which the maximum amount of heat is lost to the surroundings. This means for example that there is hardly any difference in heat loss, after the same cooling period, between a wall thickness of 10 cm and a wall thickness of 25 cm. Yet the amount of energy, necessary to bring back the walls to their required baking temperature, will be smaller for a 10 cm thick wall than for a 25 cm thick wall.

In case of the present design, the combustion air required is sucked into the combustion chamber, through the air inlets in the combustion chamber door, completely by means of the net draught of the oven. The net draught of the oven is the specific oven draught (including the chimney) minus the pressure losses that occur in the flues. The net draught should not become too large, because too much excess air cools down the temperature of the combustion gases, which results in a reduction of the heat transfer as well as in a deterioration of the combustion quality. The air supply to the combustion chamber is controlled by means of dampers, placed in the door of the combustion chamber and in the chimney. The use of dampers is only meaningful in case a small change in damper position results in a corresponding change in air supply. Thus it is required that there is not too much difference between the specific oven draft and the total pressure loss. The fourth mathematical model examines the influence of flue shapes and dimensions on the specific oven draught and the pressure losses in the present design. The calculations showed that the effect of a bend or T–connection on the pressure loss is much larger than the pressure loss caused by a straight flue part. It was also shown that there is not especially one point in the system where a large pressure drop occurs, but that it is more or less equally divided over the total flue system.

The fifth mathematical model discusses a method to compute the temperature distribution over the baking compartment floor and ceiling, the temperature distribution of the combustion gases going through the flues, and the temperature time history of the baking compartment gas, during heating up of the oven. Thus predictions can be made of the heating time of the oven and temperature distribution in the oven at the time the oven has reached its required baking temperature for different power outputs, excess air factors, flame temperatures and flue dimensions. The heat transfer problem was simplified to a transient one dimensional problem (the temperature changes with respect to time and distance). From the gases flowing through the flues, the heat is transferred in three directions.

(i) Heat transfer to the baking compartment via the baking compartment floor and ceiling (y–direction).
(ii) Heat transfer from one flue part to the next flue part (x–direction).
(iii) Heat transfer to the surroundings via the side walls of the flues (z–direction).

For these calculations only the heat transfer in the y and x direction were examined. The heat transfer in the y–direction was subdivided into three individual heat transfer mechanisms.

i) Heat transfer from the hot gas flow to the baking compartment floor and ceiling.
ii) Heat transfer through the baking compartment floor and ceiling.
iii) Heat transfer from the baking compartment floor and ceiling to the baking compartment atmosphere.

Present work

This is the second report on woodfired bakery ovens published by the Woodburning Stove Group. In this report a detailed description of the construction of the laboratory model (scaled down version of the prototype bread oven) is presented in chapter 2. With this model, a large number of experiments was carried out, which are described and discussed in chapter 3. These experiments concentrated primarily on the thermal qualities of the oven i.e. heating–up time and temperature distribution under various circumstances (different power outputs, air supply settings and applied principle of combustion). However to get some provisional data on the temperature behaviour in the oven during the baking of bread, two baking experiments were carried out. Extensive baking tests are planned to be carried out with the full size oven. Besides the thermal qualities of the oven, much attention was given to the quality of combustion, with regard to the fouling of the flues and consequently deterioration of the heat transfer efficiency in the oven.

Originally it was planned to heat the oven during all experiments by means of burning wood in the so–called conventional mode of operation. However, during the course of the experiments, a new principle of combustion named "downdraft" was being tested in a cookstove by Dr. Hasan Khan (1989). The results of his tests regarding the emission of pollutants and the temperatures that could be obtained, were very promising. Since the deposit of soot during the conventional mode of burning turned out to be a big problem in this oven, it was decided to apply the downdraft principle of combustion to this oven. Thus the experiments were extended for a period similar to that spent on the conventional principle of combustion. The differences in construction of the combustion chambers for both principles of combustion are described in chapter 2.

Both the development of mathematical models of the oven, as well as the experimental work done with the laboratory sized model of the oven form parts of the process in designing a full sized improved woodfired bakery oven. Since the experiments were carried out with a laboratory sized model (scale 1:2), the experimental results obtained need to be scaled up to the dimensions of the prototype. This will be discussed in chapter 4, using a mathematical model earlier developed (Schutte et al., 1988). However, before applying the mathematical model for the dimensions of the full sized prototype, it first needs to be compared with the experimental results in order to examine its accuracy and if necessary it is to be modified. The nature of the modifications and the comparison study are also described in chapter 4. Finally, an attempt is made to predict the influence of several parameters (power output, excess air factor and flue dimensions) on the heating time and corresponding fuel consumption in the full sized prototype bread oven.

2 Construction of the laboratory model

2.1 Introduction

For the experimental work a laboratory model was constructed. This model is a scaled down version of the prototype bread oven design. The reasons for using a scaled down model are:

(i) we have yet no guarantee that the present construction and design principle of heating will work; during testing, it could appear that several changes in the design may be required; in a full scale model it would be much more difficult to carry out these adjustments;

(ii) it is easier to operate and test it; and

(iii) it reduces the building costs.

For proper testing and scaling up the results to the prototype, the laboratory model should meet the following requirements.

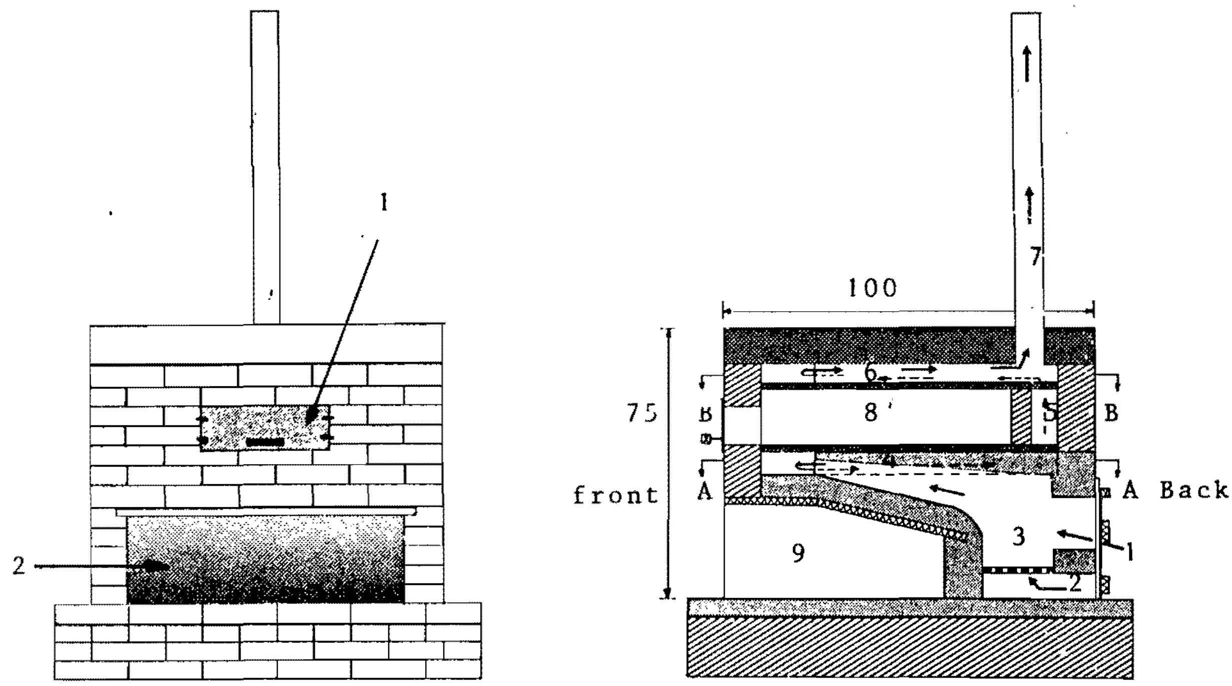
(i) The laboratory model should simulate the complete heat transfer mechanism of the prototype bread oven design.

(ii) To change different parameters and examine the influence of these changes, or carry out repair activities, the model should be designed in such a way that it can be dismantled and reassembled in about a day's work.

(iii) The building materials of the model should as much as possible be the same as for the prototype. Otherwise the values for the thermal properties of the building materials will be different, and therefore make it very difficult to use the experimental results as criteria for the final prototype design.

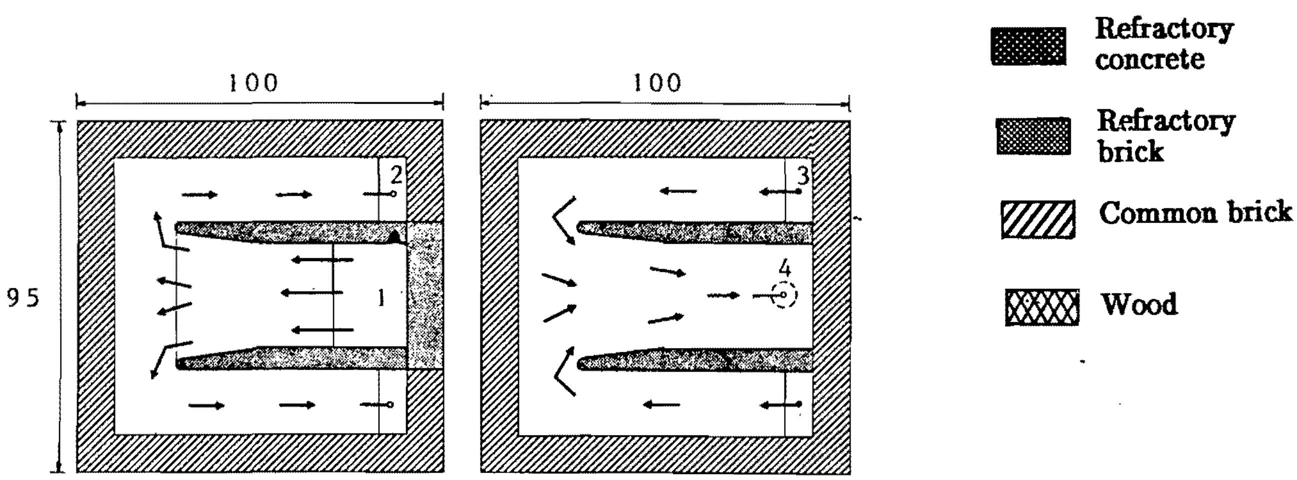
2.2 The Construction

The Size of the oven
Being a scaled down version of the prototype, the linear dimensions for the



(a)
Face of the oven
1:Baking compartment door
2:Rising or drying chamber

(b)
Longitudinal section
1:Primary air inlet; 2:Secondary air inlet; 3:Combustion chamber; 4:Bottom flues; 5:Back flues; 6:Top flues; 7:Chimney; 8:Baking compartment; 9:Rising or drying chamber.



(c)
Section A–A:Bottom flues
1:Combustion chamber exit
2:Back flue entrances

Section B–B:Top flues
3:Back flue outlets
4:Chimney entrance

Figure 2.1: Shape and dimensions (in cm) of the laboratory model as it is used for the experiments with conventional burning of the fuel.

laboratory model are proportionally reduced by a factor of 2 with respect to the dimensions of the prototype. Sometimes it was not entirely possible to carry through this rule, because of the characteristic dimensions of the available building materials. We have however minimized these differences as much as possible. Figure 2.1 shows (a) the face of the laboratory model, (b) a longitudinal section of the model, and (c) cross sections of the bottom and top flues.

The base

A platform (height of about 0.15m) made of common bricks is constructed, covered with a thick metal plate to obtain a smooth base on which the laboratory model rests. The advantage is that the entrance door to the combustion chamber is at a more convenient height for charging wood at regular time intervals, thus simplifying the work during the experiments. The construction of the oven itself used common and refractory bricks, and refractory cement as a mortar.

The combustion chamber

In the course of the experiments, two principles of combustion were studied: (a) conventional burning and (b) downdraft burning (Prasad and Verhaart, 1987). Each needed its own type of combustion chamber. The combustion chamber used for the conventional burning of the fuel was constructed first. It is shown in figures 2.1b and 2.2. For this principle of combustion, the combustion chamber with walls of refractory bricks, is divided into two parts separated by a grate on which the fuel burns. This makes it possible to supply both primary (from under the grate) and secondary air (from above the grate) into the combustion chamber. The flames and flue gases rise from the grate and enter the central bottom flue. The grate has a surface area of 0.05 m^2, and is made of hollow iron tubes with an outside diameter of 0.006m.

The door of the combustion chamber is made of sheet steel of thickness, 0.003m. In the door, holes are drilled with a diameter of 0.01m each. At the most 15 of these holes are used for the primary air inlet. For the secondary air a maximum of 23 holes can be used. The sliding dampers to control the primary and secondary air inlet are also made of sheet steel of 0.001m thickness. They can be pushed sidewards to change the air inlet area, which changes the resistance for the air flow into the combustion chamber. We assume that use of the dampers will improve the combustion efficiency and with that improve the total heat efficiency of the oven. Wooden handles are used on the door of the combustion chamber and on the dampers to prevent the burning of users' hands.

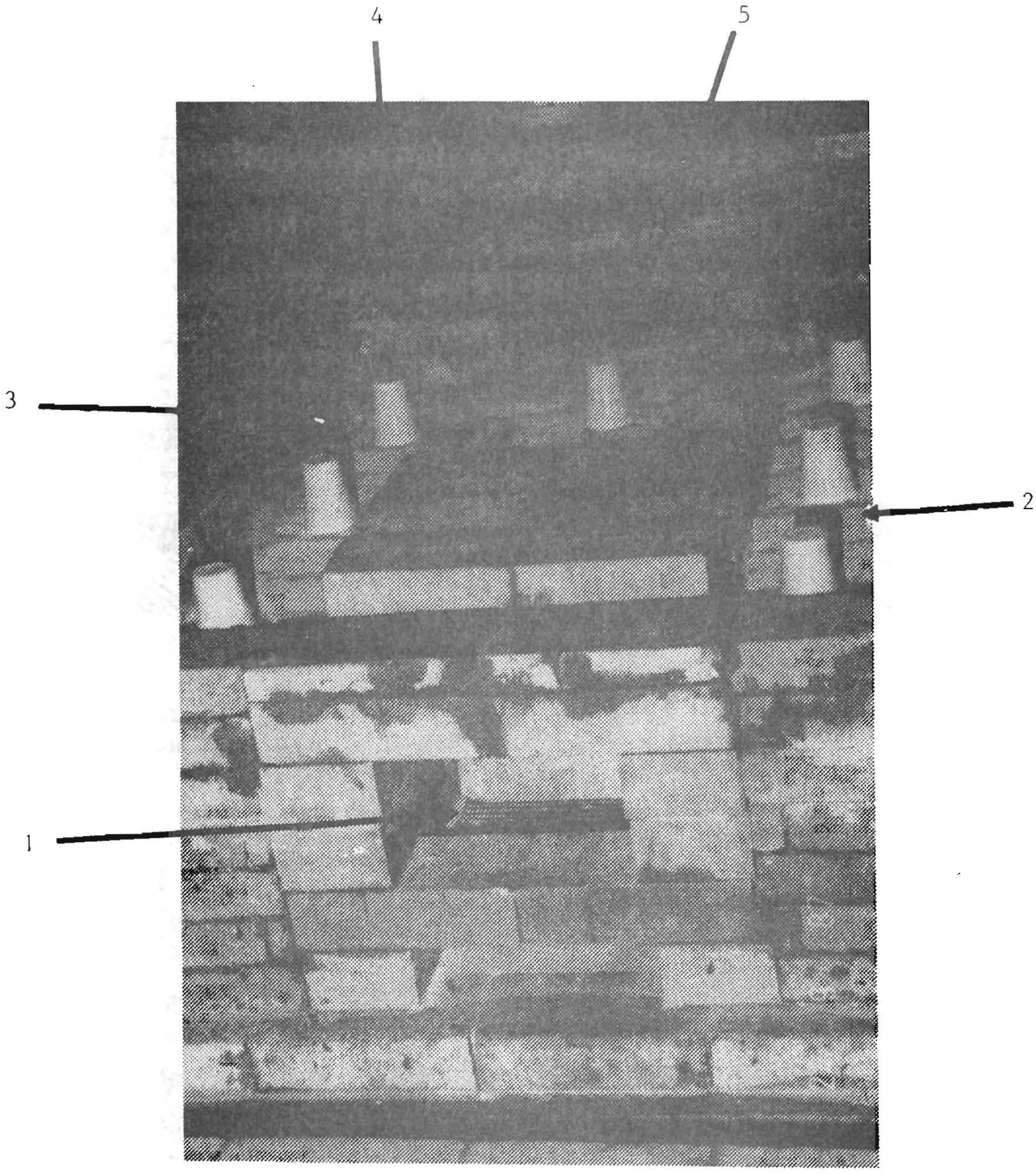


Figure 2.2: **View of the back of the laboratory model. At this stage of building, the baking compartment and top flues have not yet been installed. Only the combustion chamber for conventional burning (1), bottom flues (2), side (3) and front wall (4), and the baking compartment door (5) are visible. During building plastic cups were used to protect the thermocouples from getting damaged.**

The construction of the combustion chamber used for downdraft combustion is shown in figure 2.3. A steel sheet of thickness 0.001m is placed in the combustion chamber, leaving only a small gap at the back. The area of the grate on which the fuel burns is kept the same as for the conventional burning principle. It is preferable to construct the separating sheet with refractory bricks. However for the purpose of the present work, a steel sheet proved adequate.

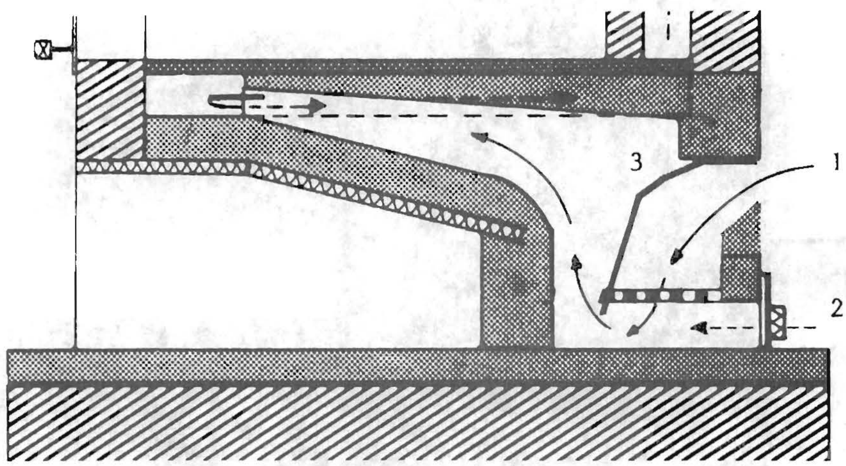
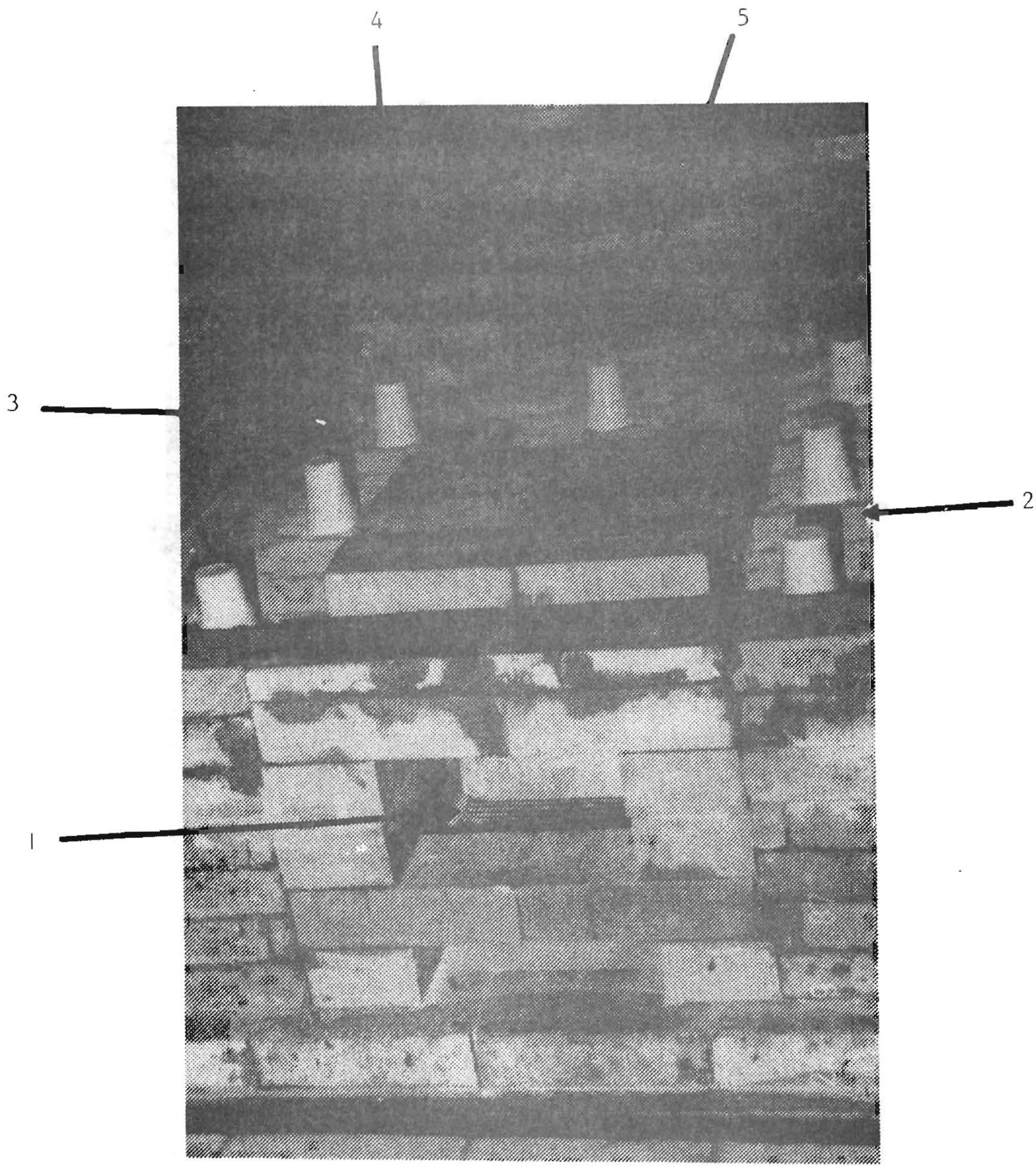


Figure 2.3: **Longitudinal section of the bottom part of the oven, showing the shape of the combustion chamber for downdraft combustion.**
1: Fuel feeding and air supply hole; 2: Secondary air by–pass; 3: Sheet steel.

The principle of downdraft combustion was first tested in 1985 by the Woodburning Stove Group (Prasad and Verhaart, 1987). Since then, several experimental stoves have been built and recently extensively tested (Khan & Verhaart, 1989). When the oven is in downdraft operation, the chimney draught induces air to flow downward through the fuelbed supporting combustion of the fuel on the grate. The flames and flue gases pass from under the grate through the small gap behind the steel sheet and enter the central bottom flue. Thus in contrast to conventional burning, the flue gas flow, air flow and the fuel feeding are all in the same direction (co–current mode or parallel flow). The most remarkable feature of this burning principle, is that it produces very clean combustion. Detailed information on downdraft combustion is given by Khan & Verhaart (1989), Khan (1990), Verhaart et al. (1989), and Khan et al (1989).

Because all the air enters through the fuel feeding hole, there is no distinction between primary and secondary air inlet. However, this design is fitted with a secondary air by–pass (see figure 2.3, no. 2). By lifting this sheet, air is directly supplied under the grate. The doors to close the fuel feeding hole and the secondary air by–pass are made of sheet steel of thickness 0.001m.



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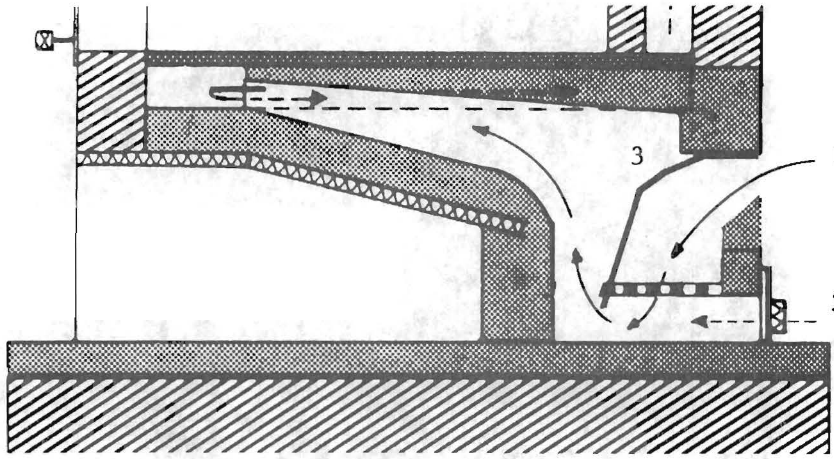

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The flues

The baking compartment is heated by means of a hot gas flow, going through flues that surround the baking compartment at the bottom, the back and the top (See figure 2.1). The central bottom flue and the left and right front bottom flues are supported by wooden plates (See figures 2.1b and 2.4).

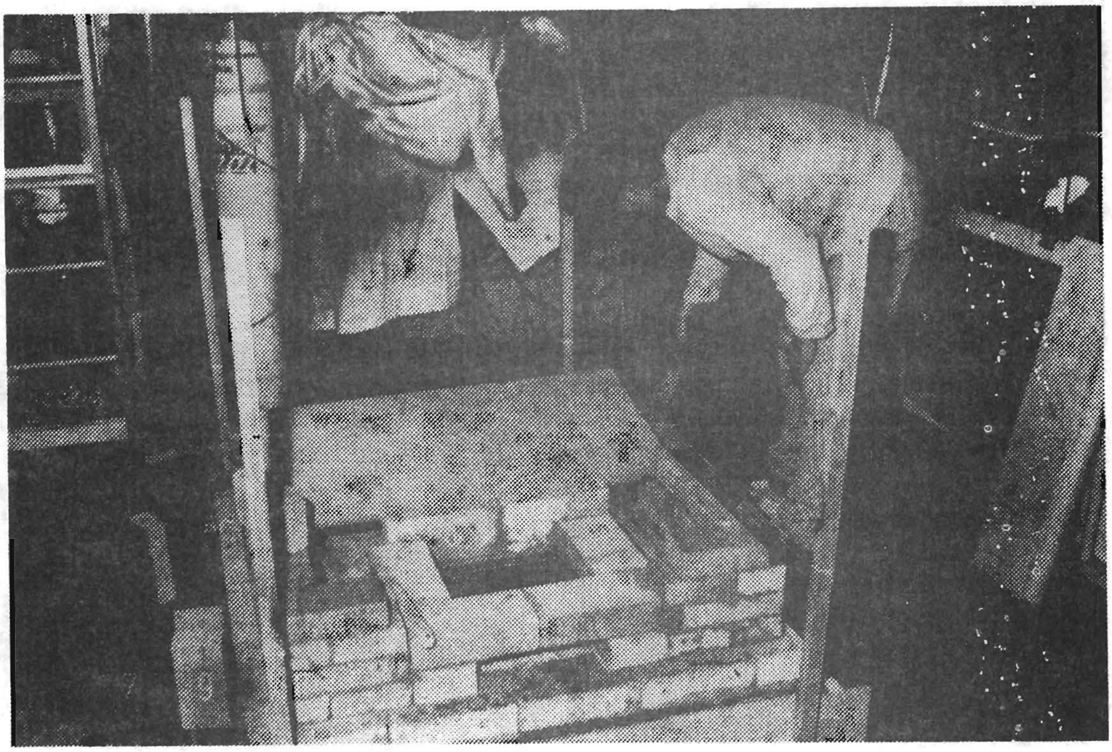

Figure 2.4: **Wooden plate of thickness 0.02m, used to support the central bottom flue.**

The wooden plates are covered with a layer of refractory bricks. It turned out that by the end of the downdraft burning experiments, the plates started to char at the points where thermocouple wires were inserted through the plates. The floors of the bottom side flues are also constructed with refractory bricks (see figure 2.2) to reduce the heat losses and to withstand the flue gas temperatures, still being relatively high in these flue parts.

Between the baking compartment floor and the central bottom flue an insulation layer is fixed, made of refractory bricks, which is slanting towards the end of the central bottom flue (See figure 2.1b). This insulation should prevent too high temperatures on the center part of the baking compartment floor. Thus providing a more uniform temperature over the floor.

The baking compartment

For the prototype design we estimated a required baking compartment floor thickness of 0.05m (Schutte et al., 1988). This does not include the extra insulation layer just above the central bottom flue. In case of the laboratory model, this would imply a floor thickness of about 0.025m. It is however very difficult, if not impossible, to construct a floor of this thickness with common or refractory bricks. We therefore made use of thin concrete plates (thickness is 0.02 m) that could withstand high temperatures and had adequate firmness (See figure 2.5).

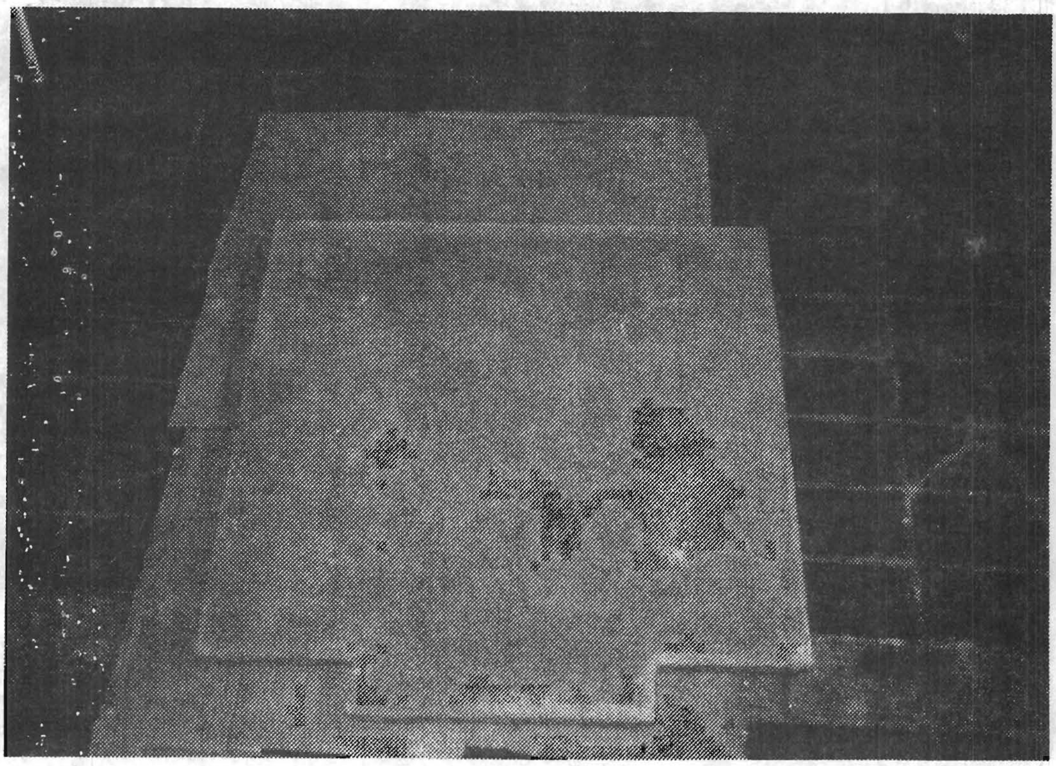

Figure 2.5: **Concrete plate used for the floor and ceiling of the baking compartment. At one side of the plate, rectangular spaces are left open for the back flues.**

Furthermore, these plates have the advantage that they can be removed quite easily when necessary, and that the thickness can be increased by just placing another plate on top of it. The same plates are used for the baking compartment ceiling. Because it must remain possible to remove the plates, they can not be fixed to the walls. Besides, due to the difference in expansion coefficient between the bricks and the plates, a gap was to be provided between the walls and the plates to prevent cracking of the plates during heating up. The gaps are closed with ceramic cord to make sure that the flue gases do not enter the baking compartment. The cord has to be replaced regularly to ensure this. In the eventual prototype, the baking compartment floor and ceiling will be made of bricks and set permanently. Furthermore to make the baking compartment leakproof, which is of great importance to the bread quality, it is being considered to let the actual baking compartment consist of a steel box. This will also improve the homogeneity of the temperature. The box will rest on the insulation layer just above the central bottom flue and/or be hung up on the side walls of the oven. To ensure sufficient heat accumulation for the floor and ceiling, bricks can be placed on the floor and top of the box.

To be able to remove these concrete plates in the laboratory model relatively easily, the top of the oven is made of a removable thick (0.10m) concrete plate. Also a part of the back wall can be removed. This is shown in figure 2.6.

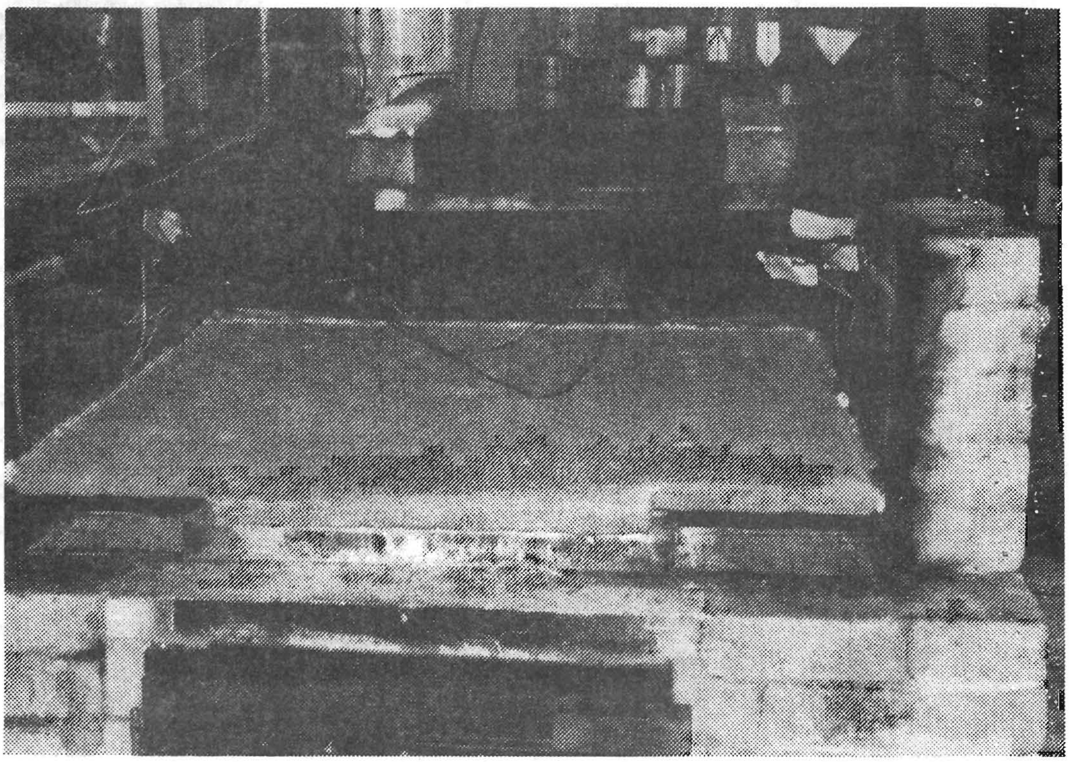

Figure 2.6: The top plate is removed just like part of the back wall.

The side and front walls of the laboratory model simultaneously account for the side and front walls of the baking compartment.

The chimney

The two most important reasons for using a chimney are:

(i) to provide the required draught, which is a necessity to obtain proper combustion; and

(ii) to lead the flue gases, once they have passed the top flues, out of the bakery.

For the experiments with conventional burning, an aluminum chimney was used with a diameter of 0.07m and a length of 1m. In the chimney a damper is installed. A construction sketch of the damper is shown in figure 2.7.

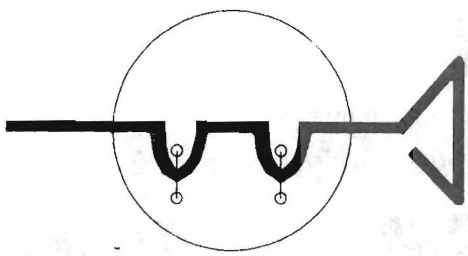

Figure 2.7: Construction of the chimney damper.

The chimney damper can be used to control the draught and by that the velocity of the flue gases. The damper opening is expected to control the level of power output. The use of a chimney damper is only meaningful in case a small change in damper position results in a corresponding change in air supply. This requires that there is not too much difference between the specific oven draught and total pressure loss. Schutte et al. (1988) calculated that the total flow resistance in the system is quite small. Therefore to prevent a large change in specific oven draught, the total air inlet area of the holes in the door of the combustion chamber and the cross sectional area of the chimney should not differ too much. Otherwise the chimney damper has to be closed almost completely to produce any effect.

For downdraft combustion all the air is sucked through the fuelbed. This requires a much larger chimney draught. Therefore both the length and the diameter of the chimney were increased for downdraft operation. The length became 1.80 m and the diameter 0.10 m. Apart from a flue gas sampling point for gas analyses, an air injector is installed in the chimney. This is shown in figure 2.8. To start downdraft combustion, some pressurized air is blown into the chimney through this injector tube. This brings about a forced chimney draught, large enough to suck the combustion air through the fuelbed. As soon as the flue gases in the chimney reach a temperature of 70 – 80°C (this takes about 30 seconds), the air injection can be stopped and natural draught takes over.

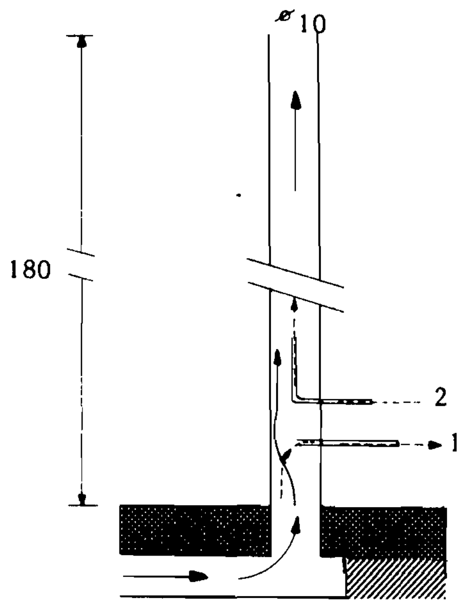

Figure 2.8: **Chimney construction used in the experiments with downdraft burning. 1: Flue gas sampling point; 2: air injection point. (Dimensions are in cm)**

3. Experiments with the laboratory model

3.1 Introduction

The experiments carried out with the laboratory model are primarily concentrated on the thermal properties of the oven. They can be divided into 4 separate groups.

(i) Temperature rise and distribution in the laboratory model in case of (a) conventional burning and (b) downdraft burning, with different power outputs and varying air supply. In the presentation of the results, a distinction will be drawn between the baking compartment gas temperatures, the baking compartment floor and ceiling temperatures and the flue gas temperatures.

(ii) Analysis of the flue gases for both principles of combustion. In case of conventional burning, the power outputs as well as the damper positions in the door of the combustion chamber and in the chimney were varied. The main interests are the flue gas CO/CO_2 ratios and the soot deposit in the flues.

(iii) Baking compartment floor, ceiling and gas temperatures as a function of time during the baking of bread, while heating the oven with conventional or downdraft burning.

(iv) Cooling down characteristics of the laboratory model.

3.2 Combustion Parameters

Before presenting the experimental procedures, results and discussion, it seems useful to discuss some relevant combustion quantities already at this point.

Wood species

As fuel for these experiments we have chosen pieces of White Fir. For the experiments with conventional burning, the wood dimensions were 20*20*110 mm. The dimensions of the woodpieces are adjusted to the dimensions of the grate, because the fuelbed should not become too thick, to guarantee that enough air can be sucked through it. Besides, enough combustion space should remain available for the proper combustion of the volatiles (Prasad and Verhaart, 1987). In case of downdraft burning, woodpieces with a dimension of 20*20*70mm were used, because Khan & Verhaart (1989) showed that during downdraft operation in a cookstove, burning of smaller woodblocks resulted in cleaner combustion. The properties of White Fir are listed in table 3.1 (Sielcken, 1983; Bussmann, 1988).

	Percentage
Ultimate analysis	
C	50.7
H	5.3
O	43.1
Proximate analysis	
Volatiles	80.0
Fixed carbon	20.0
Ash	-
Density (kg/m^3)	410
Gross calorific value (MJ/kg)	19.9

Table 3.1: Properties of White fir

The moisture content (m) of the woodpieces can of course vary. Therefore we have determined the average moisture content on the basis of the dry wood weight for every parcel of wood (see equation 4.6 in section 4.2).

Power output

To examine the heating time and the corresponding temperature distribution in the laboratory model at various constant power outputs, a steadily burning fire is required. One way of obtaining such a fire is by supplying the fuel continuously. However, for a woodfired oven, wood can only be supplied batch–wise. For such a batch process a so–called nominal power can be defined according to the following expression (Bussmann, 1988).

$$P_n = \frac{\Delta M_f \; B_f}{\Delta t} \qquad (3.1)$$

This nominal power can be varied within limits by changing ΔM_f, Δt or both. In case of substantial build-up of charcoal in the fuelbed, heat released by the charcoal is supplied to the flues even after the last flames have disappeared. This has not been accounted for in equation 3.1. In such cases it might be better to use a power output definition based on the total duration of the experiment, leading to the concept of an average power output.

$$\bar{P} = \frac{\Sigma \Delta M_f \; B_f}{t_T} \tag{3.2}$$

During the experiments with conventional firing, the wood is supplied every 10 minutes, unless otherwise stated. In the case of the downdraft burning experiments, charging intervals of 6 minutes were chosen during heating up of the oven. As soon as the oven reached baking temperatures (± 240°C), the "simmering" period started and the charging intervals were increased to 10 minutes.

Air quantities

Direct measurement of air drawn into the flues of the oven is not an easy experimental task. Therefore we have relied on the carbon balance to infer the air flow through the flues using measurements of gas composition, fuel quantities and the ultimate analysis of the fuel. A confusing factor in this process of measuring could be the oxygen present in the wood, which also participates in the combustion process along with the oxygen in the air. However, Sielcken (1983) derived equations which enable us to compute the stoichiometric amount of air and the excess air factor, using only the ultimate analysis of the fuel (In our experiments we used White Fir, see table 3.1) and the CO and CO_2 content of the flue gases. The expression for the stoichiometric amount of air in m^3 to burn completely 1 kg of wood may be written as follows (Bussmann, 1988):

$$V_{st} = \left[\frac{1}{[1 + m][1 + a]} \right] \left[\frac{\frac{p}{12} - \frac{y}{32}}{0.21} \right] 0.224 \tag{3.3}$$

where: m is the moisture content on dry basis.

a is the ash content.

p is the percentage of carbon in the wood.

y represents the difference between the oxygen and 8 times the hydrogen content in the wood ($y = \%O_2 - 8\ \%H_2$).

The stoichiometric amount of air in the case of White Fir with 0% moisture will be 4.5 m^3/kg of wood. However, in addition to the theoretical (stoichiometric) amount of air necessary to burn the wood, a sufficient amount of air must be supplied to the wood in excess, in order to achieve complete combustion. In general, model calculations and experimental data provide values for the required excess air factor which are in between 1.6 and 2 (Verhaart, 1981; Bussmann, 1988). The actual excess air factor obtained during an experiment can be computed with the following general equation (The complete derivation of the expression is given by Sielcken, 1983).

$$\lambda = \frac{1 - [CO] - [CO_2]}{\frac{1}{0.21}\left[\frac{1}{2} - \frac{3}{8}\frac{y}{p}\right][CO] + \frac{1}{0.21}\left[1 - \frac{3}{8}\frac{y}{p}\right][CO_2]} + 0.21 \qquad (3.4a)$$

Using the properties of White Fir (table 3.1), equation 3.4a can be simplified to:

$$\lambda = \frac{1 - [CO] - [CO_2]}{2.36[CO] + 4.74[CO_2]} + 0.21 \qquad (3.4b)$$

The excess air factor is also defined as the ratio of the total amount of air drawn into the combustion chamber and the stoichiometric amount of air required for the combustion process. Thus the total amount of air that is drawn into the combustion chamber can now be defined as:

$$V_T = \lambda\, V_{st} \qquad (3.5)$$

Sensible heat losses

The gas analysis results and the temperature measurements also enable us to get some indication about the unused energy going up the chimney. This can be illustrated using a simple relation between the energy supplied by the fuel and the energy entering the chimney (Prasad and Verhaart, 1987).

$$H_s = \frac{\left[V_{st}\ \lambda\ \rho_{T_g} + 1\right]\left[T_{g_{ch}} - T_a\right] Cp_{T_g}}{B_f} 100\% \tag{3.6}$$

The energy entering the chimney is considered to be a heat loss because as soon as the flue gases leave the central top flue and enter the chimney, the remaining heat capacity of the flue gases can no longer contribute to the heating up of the baking compartment. This is also the reason that during the experiments the thermocouple installed in the chimney was positioned at the entrance to the chimney. Expression 3.6 shows that an increase of the air supply and/or an increase of the flue gas temperatures at the chimney entrance, immediately results in an increase of the sensible heat losses. The density and the specific heat capacity depend on the composition and the temperature of the flue gases. We can approximate the values for them using the following expressions (Schutte et al., 1988), taking the physical properties of air as those of the flue gases.

$$\rho_{T_g} = \frac{353.2}{T_{g_{ch}}} \tag{3.7}$$

$$Cp_{T_g} = 0.2 T_{g_{ch}} + 933.3 \tag{3.8}$$

Where:
$T_{g_{ch}}$ = Flue gas temperature at the chimney entrance (K).

3.3 Instrumentation

Temperature measurements

To measure the temperatures at different places in the model, a total of 18 thermocouples have been installed. 11 thermocouples are distributed through the baking compartment according to figure 3.1. Figure 3.2, which represents the shape of the flues in the model, indicates the positions of the thermocouples installed in the flues. Furthermore, one thermocouple each has been installed in the center of the left and right outer wall of the oven.

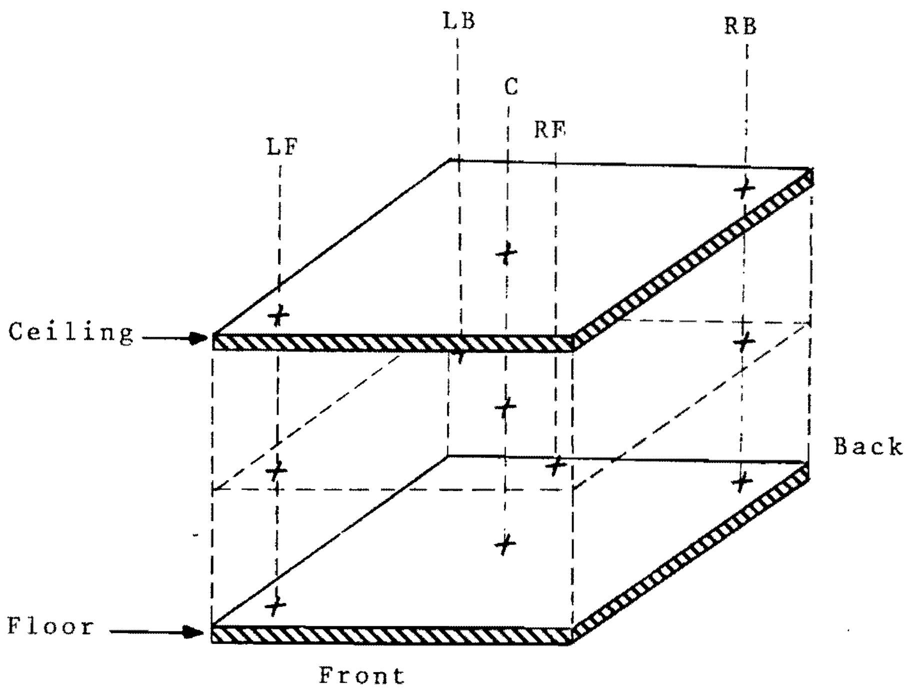

Figure 3.1: **Location of the thermocouples on the baking compartment floor, the baking compartment ceiling and in the baking compartment gas halfway between floor and ceiling.**
C = position center; RF = position right front; LF = position left front; LB = position left back; RB = position right back.

Flue gas analyses

The CO and CO_2 contents of the flue gases leaving the chimney are measured by means of a CO&CO_2 BINOS–IR–Gasanalyser. The principle of the meter is described in Measuring and Analytical Techniques, Reference manual Binos–IR–Gasanalyser. The CO/CO_2 meter is calibrated for each experiment with a standard gas. During the experiments with conventional burning, the composition of the standard gas was: 0.506% CO; 5.88 % CO_2; 8.20% O_2 and the remaining part is nitrogen gas. During the downdraft burning experiments a new calibration gas was used which consisted of 0.47% CO; 6.04% CO_2; 7.34% O_2 and the remaining part nitrogen gas.

The temperature and flue gas data is recorded by means of Hewlett–Packard data–acquisition equipment, which makes it possible to use a PC for further processing of this data.

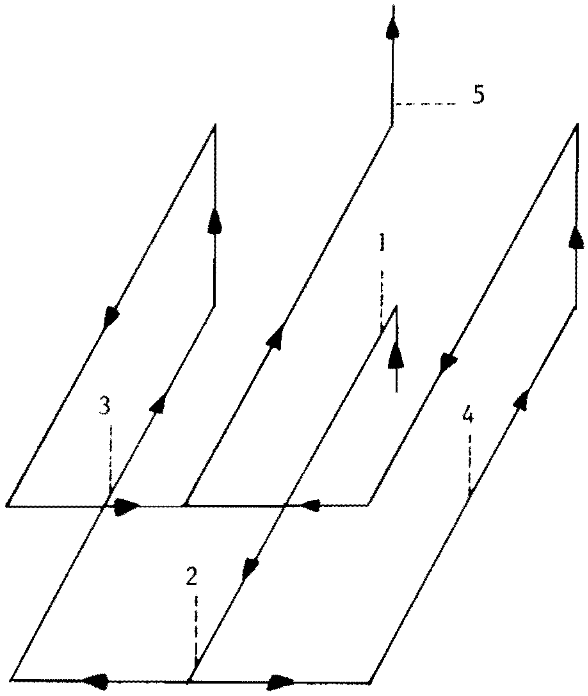

Figure 3.2: **Place of the thermocouples in the flues of the laboratory model. 1: combustion chamber exit; 2: end of the central bottom flue; 3: halfway the left side bottom flue; 4: halfway the right side bottom flue; 5: chimney entrance.**

3.4 Preparatory Firing

Whenever a new oven is built, it is recommended to heat the oven several times for several hours before starting to bake bread. The oven needs to get used to high temperatures by gradually increasing the power output and the periods of heating. This is important since the construction materials contain moisture. Part of the heat supplied is used for the evaporation of this moisture. This will result in a reduced temperature rise in the baking compartment.

The oven is heated 3 times with a nominal power output of 7.5 kW, applying the conventional principle of combustion (see for definition of the nominal power output section 3.2). Primary air (p) was supplied through 15 holes and secondary air (s) through 23 holes in the door of the combustion chamber. During these heating periods, the chimney damper was fully opened. Figure 3.3 shows the effect of heat extraction for the evaporation of moisture on the temperature time history of the baking compartment atmosphere. Each curve represents the average value of the 5 measuring points in the baking compartment gas (see figure 3.1; the 5 locations showed only little difference in temperature). It clearly shows a difference in temperature rise between curve 1 and 2. The very small differences in temperature rise between the second and third time of heating indicate that most of the moisture has been evaporated and that the oven can be expected to be ready for the experiments.

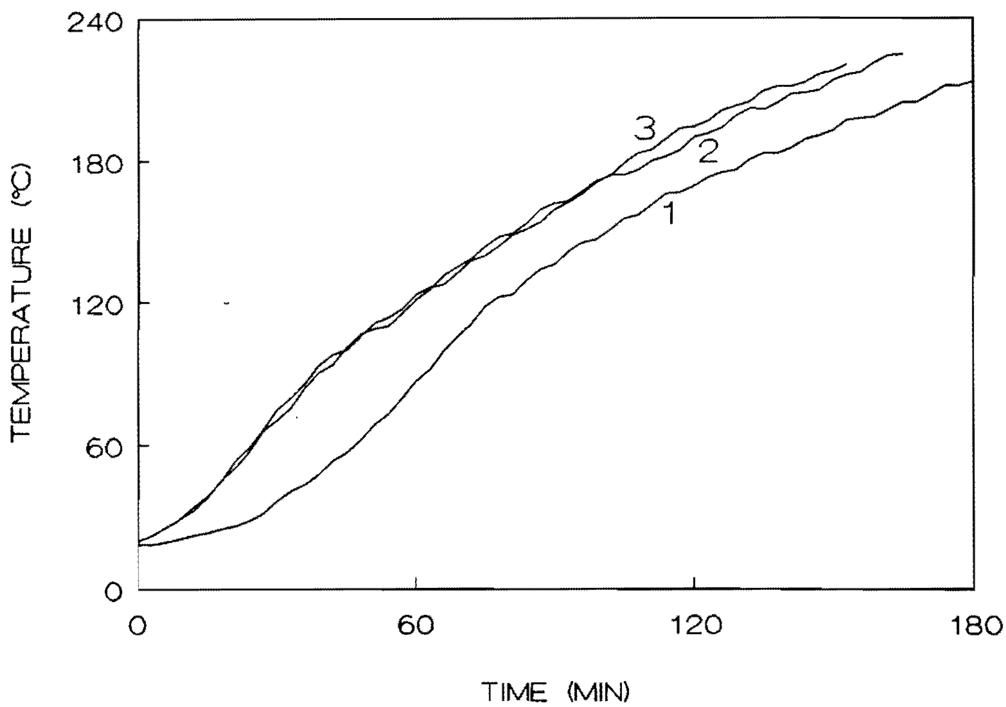

Figure 3.3: **Average temperature time histories of the baking compartment gas during pre-heating of the model.**
1 = first time heating; 2 = second time heating; 3 = third time heating.

3.5 Experimental Programme

We have examined the temperature rise and distribution in the laboratory model, as well as analyzed the flue gases leaving the chimney, under different circumstances. The main variables studied are summarized in table 3.2. The damper in the chimney during the experiments with conventional burning was kept fully open.

Sl. nr.	Power output (kW)	Conventional burning				Downdraft burning
		Primary air inlet		Secondary air inlet		Air inlet area (cm^2)
		No. holes opened	Area $(cm)^2$	No. holes opened	Area (cm^2)	
1	7.5	15	11.8	23	18.1	-
2	7.5	5	3.9	10	7.9	-
3	7.5	2	1.6	3	2.4	-
4	10	15	11.8	23	18.1	-
5	10	2	1.6	3	2.4	-
6	12.5	15	11.8	23	18.1	-
7	12.5	5	3.9	10	7.9	-
8	15 *	-	-	-	-	165.8

* Average power output of the downdraft experiments carried out.

Table 3.2: Experimental circumstances examined

3.6 Results and Discussion

This section provides a survey of the experimental results as well as a discussion of the results. The main interest are the temperatures recorded in the baking compartment. After all, that is where the baking takes place. The quality of the baked bread is strongly influenced by the temperatures in the baking compartment. However to explain the experienced temperature behaviour in the baking compartment, it is useful to present the temperatures recorded in other parts of the oven, the analysis of the flue gases and the fuel consumption (power output).

Temperature rise of the baking compartment gas

The rise in average baking compartment gas temperatures for the experimental conditions listed in table 3.2, are shown in figure 3.4. The plotted temperatures for each experiment are in reality the average values of the 5 measuring points in the baking compartment gas (see for the location of these points figure 3.1,). The figure shows that a baking temperature of 240°C (required to bake small white loaves), is only reached in case of downdraft burning (curve 8) and in case of conventional fires of 10 and 12.5 kW (curves 4, 6 and 7). However, the differences in heating times between downdraft and conventional burning are considerable. In table 3.3,

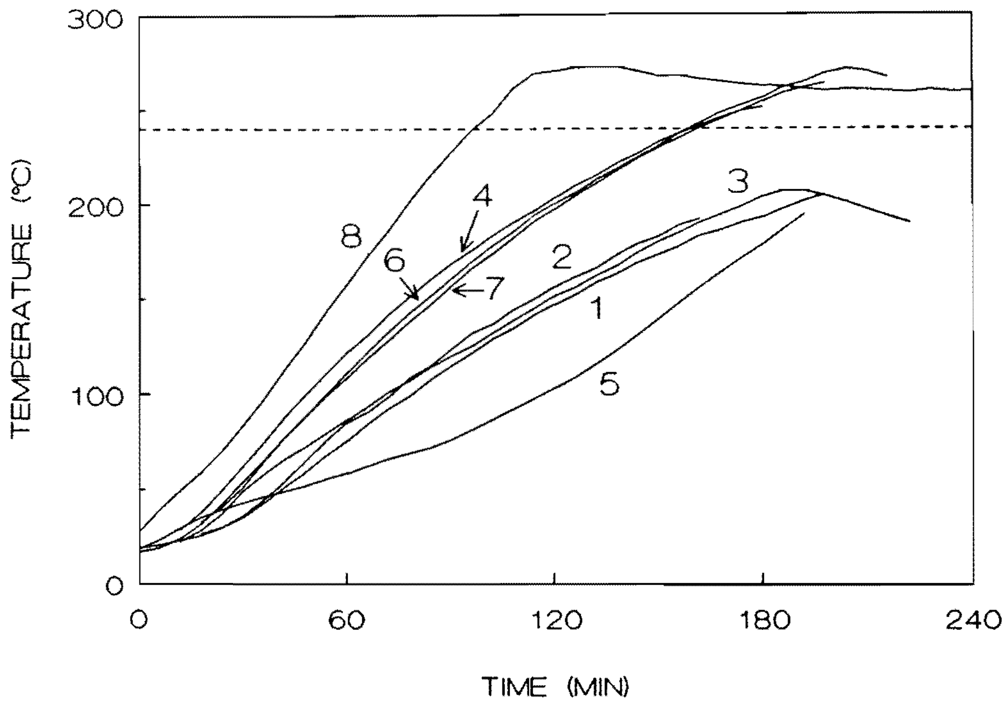

Figure 3.4: **Average rise in baking compartment gas temperature for different power outputs and air supply configurations.**
1: 7.5 kW (p=15, s=23); 5: 10 kW (p= 2, s= 3);
2: 7.5 kW (p= 5, s=10); 6: 12.5 kW (p=15, s=23);
3: 7.5 kW (p= 2, s= 3); 7: 12.5 kW (p= 5, s=10);
4: 10 kW (p=15, s=23); 8: 15 kW downdraft burning.

the heating times are listed as well as the corresponding amount of wood consumed.

Sl. nr.	*Conventional principle of burning*	*Wood consumed (kg)*	*Heating time (min)*	*Temperature reached (°C)*
1	7.5 kW (p=15; s=23)	5.1	186	206
2	7.5 kW (p= 5; s=10)	5.1	177	196
3	7.5 kW (p= 2; s= 3)	4.8	186	208
4	10 kW (p=15; s=23)	5.7	160	242
5	10 kW (p= 2; s= 3)	6.9	200	195
6	12.5 kW (p=15; s=23)	6.7	162	241
7	12.5 kW (p= 5; s=10)	7.4	162	240
	Downdraft principle of burning			
8	15 kW (average)	6.0	97	239

Table 3.3: **Heating times and the amount of wood consumed during the heating up period**

It is seen from the table, that the time savings in heating the oven up to 240°C, using downdraft combustion, is about 60 minutes. Moreover, the fuel saving is approximately 17%. The difference in heating rate is probably due to the possibility of applying higher power outputs during downdraft combustion. When discussing the results of the flue gas temperatures, it is shown in figure 3.6 that during downdraft combustion higher flue gas temperatures are recorded at the chimney entrance. In combination with the larger chimney diameter and length used during downdraft combustion, this results in a stronger draught, which leads to higher flue gas velocities and consequently an increased heat transfer.

After the baking temperature was reached during downdraft burning, the power output was reduced in such a way that the temperatures in the baking compartment remained constant ("simmering" period). Maybe that a 7.5 kW conventional fire would eventually also result in an acceptable baking temperature, but the heating period would become unacceptably long.

It is striking to see that for a 10 kW and 12.5 kW fire (curves 4, 6 and 7 of figure 3.4), the differences in temperature are negligible. For the 7.5 kW fire (curve 1,2 and 3 of figure 3.4), the temperature rise, compared to a 10 kW conventional fire, is smaller by about 55°C. Curve 5 (10 kW, p = 2 and s = 3) shows a strong divergent behaviour. After about 2 hours of firing, the temperature difference between curve 4 (10 kW, p = 15 and s = 23) and curve 5 has increased to almost 100°C. This is not only the result of a reduced air supply, otherwise a similar behaviour should most likely have been noticed for the 12.5 kW fire with a reduced air supply. During the experiment of curve 5, there were severe problems to keep the wood burning during the first half hour of firing. Whenever the door of the combustion chamber was closed after charging of fresh wood, there was a large decay of the fire. It was only after 70 minutes that we were able to reach the required power output of 10 kW. Before that time it fluctuated between 6 and 9 kW. At t = 80 to 90 minutes the temperature in the baking compartment starts to react to this increase in power output. The delay of about 10 to 20 minutes is caused by the oven mass, which absorbs part of the supplied heat before releasing it to the baking compartment atmosphere.

If we only consider the temperatures recorded in the baking compartment gas, figure 3.4 gives the impression that the influence of different air supply settings on the baking compartment gas temperature is altogether very small, if not, negligible. However, without the support of the gas analysis results (CO/CO_2 ratios and excess air factors) it is difficult to discuss this question properly at this stage. The results on gas analysis will be discussed later on in this section. The maximum temperature difference reached in case of the 7.5 kW conventional fires (curves 1, 2 and 3) is 10^{0}C. For the 12.5 kW conventional fires, the maximum temperature difference is even smaller.

Temperature distribution on the baking compartment floor and ceiling.

The temperature distributions on the baking compartment floor and ceiling have also been studied for the conditions listed in table 3.2. We have selected only three points for temperature measurements on both the floor and ceiling in order to keep the data processing within acceptable limits. By positioning the thermocouples in a diagonal over the floor and ceiling, we expected to obtain the best possible indication about the floor and ceiling temperatures. Of course this implied that we assumed the center, front and back sections of the floor and ceiling to have each uniform temperatures. Table 3.4 summarizes the temperatures reached at different locations on the baking compartment floor and ceiling (positions are indicated in

position on floor	Conventional burning							Downdraft burning
	7.5 kW			10 kW		12.5 kW		15 kW
	1	2	3	4	5	6	7	8
	p=15 s=23	p=5 s=10	p=2 s=3	p=15 s=23	p=2 s=3	p=15 s=23	p=5 s=10	
C	351	304	312	375	297	364	370	312
LF	241	300	256	335	269	319	328	382
RB	243	220	209	253	160	245	254	263
position on ceiling								
RB	198	172	177	224	124	211	221	221
LF	150	193	180	248	160	234	242	220
C	200	200	207	262	168	253	262	219

Table 3.4: Temperatures (^{0}C) reached on the baking compartment floor and ceiling after 180 minutes of conventional firing, and after 100 minutes of downdraft burning.
p : number of holes through which primary air is supplied;
s : number of holes through which secondary air is supplied;
C : position center; LF : position left front; RB : position right back.

figure 3.1) after 180 minutes of conventional firing with 7.5 kW, 10 kW and 12.5 kW and various air supply configurations, and after 100 minutes of downdraft burning (average time period in which the oven reached a baking temperature of 240°C). The following can be read from table 3.4.

(i) The order of the floor and ceiling positions in table 3.4 follow the flue gas path through the oven. Because the flue gases release heat to the baking compartment floor and ceiling, they show a continuous decrease in temperature from the combustion chamber exit to the chimney entrance. Due to the drop in flue gas temperature from combustion chamber exit to chimney entrance, the heat transfer from the flue gases to the flue walls reduces accordingly. Thus a similar order in baking compartment floor and ceiling temperatures can be expected. This is true for the floor temperatures except in the case of downdraft burning (no. 8) and the 7.5 kW conventional fire (no. 1). In case of downdraft burning, the highest floor temperatures are recorded at the left front position. This can be explained as follows. Although the flame temperature is independent of the power output, the flame length is a strong function of the power output (Bussmann, 1988). Since the power outputs applied during downdraft combustion were higher than during conventional burning, it was not a surprise to notice that during downdraft combustion, the flames even entered the bottom side flues, while they remained in the central bottom flue during conventional burning. In contrast to the central bottom flue, there is no extra layer of insulation between the bottom side flues and the floor. Due to the presence of flames, the heating rate of this part of the floor increases, while in the central bottom flue, the insulation layer is to be heated first. To make the temperatures on the floor more uniform during downdraft combustion, it might be advisable to extend the insulation layer into the side flues. The authors are not able to provide a solid explanation for the divergent floor temperatures recorded during the first experiment with a 7.5 kW conventional fire.

The ceiling temperatures measured are almost exactly opposite to the flow path of the flue gases, especially in the case of the 10 and 12.5 kW conventional fires, where the center ceiling positions reach even higher temperatures than the position RB on the baking compartment floor. The reason for this is probably that the front, and especially the center part of the ceiling receive a lot of radiative heat from the front and center parts of the floor, which have much higher temperatures than the rest of the baking compartment.

(ii) Because the floor temperatures are higher than the ceiling temperatures, the direction of the gravity is the same as the temperature gradient. This creates the possibility of natural convection. From a bread quality point of view, there is no objection against free convection, provided that the floor and ceiling temperatures are reasonably uniform to guarantee a uniform heat transfer by means of free convection to the dough pieces. Unfortunately the temperature differences are considerable, except for the ceiling temperatures in the case of downdraft combustion. The actual effect of these unequal temperatures on the bread quality is uncertain and can only be examined by means of baking tests. However it was noticed, especially during the downdraft experiments, that the temperature differences between the three locations on the baking compartment floor reduced during the period the oven was fired with a reduced power output (not shown in table 3.4). We might therefore conclude that it is advisable to let the oven rest for some time before baking is started. Thus the floor and ceiling temperatures will become more uniform, which definitely improves the quality of baking. Furthermore, the uniformity will also be improved by using metal plates or baking tins instead of baking the bread directly on the floor (see also the baking experiments in section 3.7). Firstly because the small layer of air between the floor and the metal plates/baking pans introduces an extra heat resistance which reduces the temperatures at the dough surfaces, preventing them from being burnt. Secondly, the high conductivity of the metal will result in more uniform temperatures at those places where the dough surfaces make contact with the metal sheets (baking pans).

(iii) Between the 7.5 kW conventional fires and the 10 kW conventional fire (p = 15; s = 23), the differences in measured floor and ceiling temperatures are considerable, while there are only small differences between the 10 and 12.5 kW conventional fires. A similar temperature behaviour was noticed in the results of the baking compartment gas temperatures.

Power output

The equations for the nominal and average power output were already given in section 3.2. Figure 3.5 shows an example of the computed nominal power outputs in the case of conventional burning (curves 1, 2 and 3; primary air supplied through 15 holes and secondary air supplied through 23 holes in the combustion chamber door). Curve 4 shows the computed nominal power output for the downdraft burning experiments. The weights of the wood charges were measured just before they were added to the fire at each time interval. The figure clearly shows the

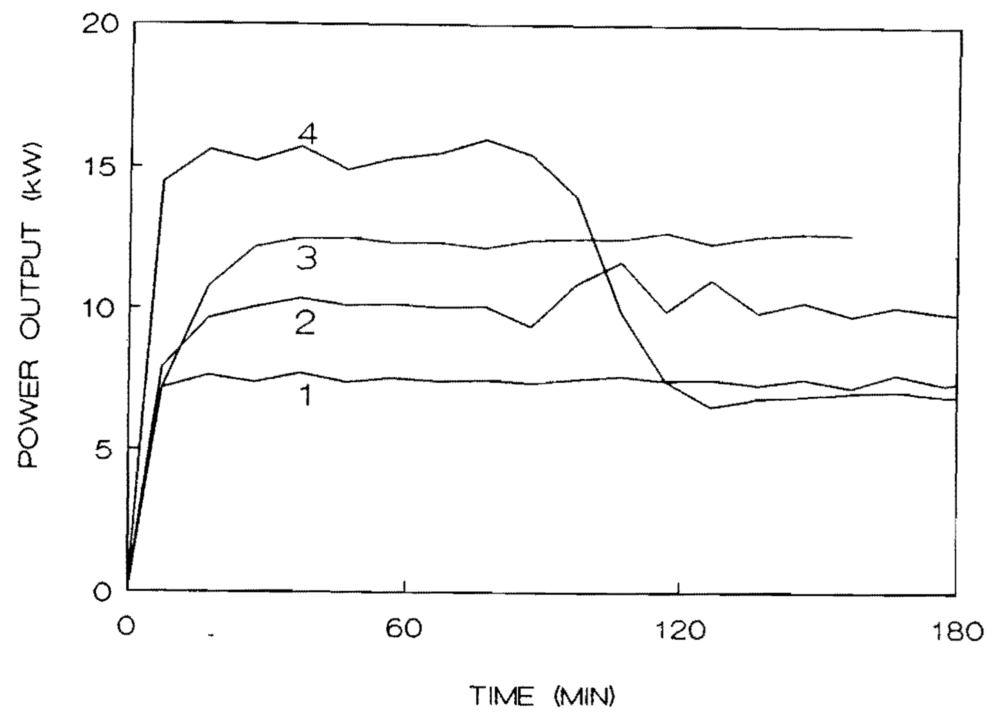

Figure 3.5: **Power outputs as a function of time. Conventional burning (primary air holes opened=15; secondary air holes opened=23) : 1: 7.5 kW; 2 : 10 kW; 3 : 12.5 kW; 4 : Downdraft burning.**

reduction in power output during downdraft burning after the baking temperature is reached. Although the baking compartment gas temperature is still above 240°C (see figure 3.4) and therefore allows further reduction of the power output, it has not been possible to further reduce it because in that case not enough charcoal is left on the grate for the ignition of the fresh wood charges. This charge size problem can possibly be solved by placing a damper in the chimney to reduce the draught and with that reduce the burning rate. However, as will be discussed later on, a damper should be used with some caution since it can largely affect the combustion quality (if the draught is reduced too much, not enough oxygen is supplied to the fuel which results in a strong deterioration of the combustion).

Studying the nominal power outputs of the different conventional burning experiments does not provide an explanation for the almost identical baking compartment gas, floor and ceiling temperatures in case of 10 and 12.5 kW fires and the rather large temperature differences between these two power outputs and the 7.5 kW fires (see figure 3.4 and table 3.4). Due to the batch–wise supply of wood, small differences in wood weight per charge and time interval between two charges occur. The nominal power output curves therefore usually show some fluctuation. However in the case of the 10 kW conventional fire, some mistakes in charge size must have been made which have caused this relatively large fluctuation.

Flue gas temperature distribution

The locations at which flue gas temperatures are measured, were already indicated in figure 3.2 (section 3.3). Due to the batch–wise supply of wood, the flue gas temperatures strongly fluctuate over time. In that case for ease of survey it is recommended to make use of the so–called moving average. This method filters the very high frequency fluctuation. In these experiments, the charging interval was 10 minutes and the sampling frequency once every minute. Therefore we used a moving average over 10 measured points. This means that the first plotted value of a curve becomes the average of the measured values 1 to 10. The second plotted value of a curve becomes the average of the measured values 2 to 11. The third plotted value the average of the measured values 3 to 12 etc.. Figure 3.6 shows the flue gas temperatures as a function of time at the combustion chamber exit (closed

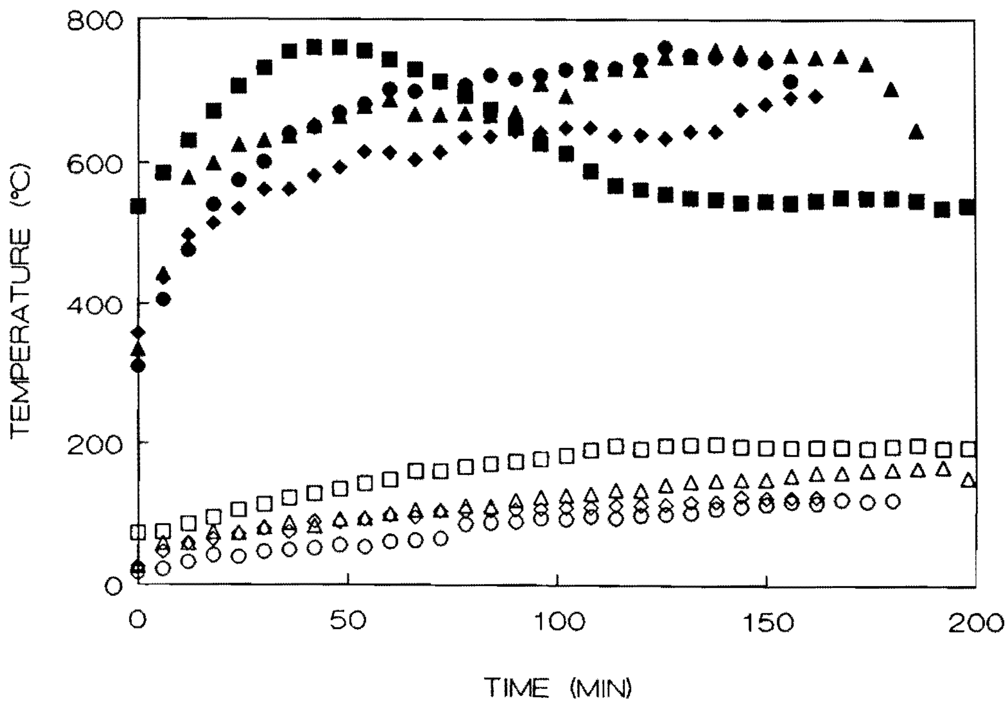

Figure 3.6: **Flue gas temperatures as a function of time at the combustion chamber exit (closed symbols) and at the chimney entrance (open symbols).**
◆ and ◇ = 7.5 kW; ▲ and Δ = 10 kW); ● and O = 12.5 kW; ■ and □ = 15 kW downdraft.

symbols) and at the chimney entrance (open symbols) in case of 7.5, 10 and 12.5 kW conventional fires with primary air supply through 15 holes and secondary air supply through 23 holes, and in case of downdraft burning. The average power output during heating up, using downdraft combustion, is about 15 kW (0 – 100 min.). During the "simmering" period, the average nominal power output is about 7 kW. Although it is not shown in figure 3.6, a change in number of opened primary and secondary air holes in the case of conventional burning, had only marginal influence on the flue gas temperatures. Figure 3.6 shows that in case of a 10 kW (▲) and a 12.5 kW (●) conventional fire, the flue gas temperatures at the combustion chamber exit are more or less the same (for the duration of both experiments, the average temperature is about 665°C). In case of the 7.5 kW fire (◆), these flue gas temperatures are on the average smaller by about 65°C. During heating up with downdraft burning (0 – 100 min.), the average flue gas temperature at the combustion chamber exit is approximately 685°C. The differences in flue gas temperatures at the combustion chamber exit can be explained as follows. There are two factors which largely influence these temperatures.

(i) The excess air factor (defined in section 3.2). The higher the excess air factor, the smaller the resulting flame/flue gas temperature will be, and vice versa.

(ii) The combustion quality. A more complete combustion will result in higher flame/flue gas temperatures.

In case of the 7.5 kW conventional fire, the somewhat lower flue gas temperature at the combustion chamber exit is most likely the result of a too high excess air factor (almost 6). In case of the 10 and 12.5 kW conventional fires, the excess air factors are on the average 3.2 and 1.75 respectively (see figure 3.10). From an excess air point of view, one would expect higher temperatures in case of the 12.5 kW fire. However the gas analysis results (to be discussed hereafter) show a very poor combustion quality in the case of a 12.5 kW fire, which apparently has resulted in almost identical flue gas temperatures at the combustion chamber exit for 12.5 and 10 kW conventional fires.

The higher flue gas temperatures at the combustion chamber exit during heating up with downdraft combustion are the result of the superior quality of combustion (see already figure 3.9). The reduction in flue gas temperature at the combustion chamber exit after the heating up period is the result of a large increase in excess air factor (from 2.3 to 6.1). During the "simmering" period, large holes appeared in the fuelbed through which air passed unused. A striking thing is that it does not seem to influence the flue gas temperatures at the chimney entrance very much.

These temperatures slowly become constant (see figure 3.6, □). Since the flue gas temperatures at the combustion chamber exit have reduced and then become constant, the total heat transferred from the flue gases to the flue walls of the entire flue system, first reduces and then remains constant. As it was shown in figure 3.4 (curve 8), just enough heat arrived in the baking compartment to maintain a constant baking compartment gas temperature.

The average flue gas temperatures at different locations in the flues (see figure 3.2) for the same experiments as those in figure 3.6 are plotted in figure 3.7 (location 3 is the average of the flue gas temperatures measured halfway the left and right side bottom flue). When curves are fitted through the points, as done in the figure, they give an indication of the average flue gas temperature distribution from combustion chamber exit to the chimney entrance during heating up of the oven with different power outputs and principles of combustion.

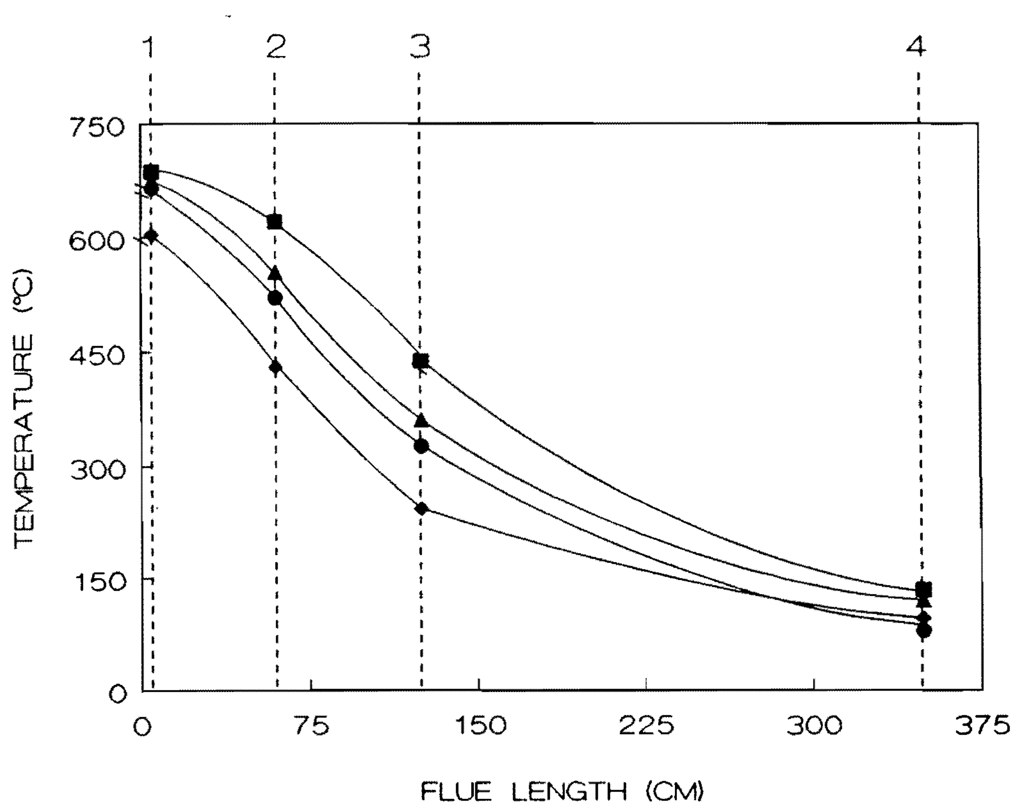

Figure 3.7: **Flue gas temperature distribution from combustion chamber exit to chimney entrance during heating up under various circumstances.**
◆ : 7.5kW (p=15; s=23); ▲ : 10kW(p=15; s=23);
● : 12.5kW (p=15; s=23); ■ : Downdraft (15kW)
1 = Combustion chamber exit; 2 = end central bottom flue; 3 = Halfway left and right side bottom flue; 4 = Chimney entrance.

The reduction in flue gas temperature from combustion chamber exit to the chimney entrance is due to the heat transfer from the flue gases to the flue walls. The amount of heat transferred depends on:

i) the temperature difference between the flue gas and the flue wall;

ii) the dimensions of the flue; and

iii) the heat transfer coefficient, which in turn depends on the flow velocity (draught), the flue gas temperatures and the dimensions of the flues.

Thus the reduction in flue gas temperature from combustion chamber exit to the chimney entrance is an indication of the total amount of heat transferred from the flue gases to the flue walls. Since the difference in recorded flue gas temperatures between a 10 and 12.5 kW conventional fire are very small through the entire flue system (this is primarily due to the poor combustion efficiency obtained with the 12.5 kW fire as mentioned before), it is obvious that one can expect more or less similar temperature rises of the baking compartment gas. The lowest flue gas temperatures are recorded during the experiments with the 7.5 kW conventional fires (this was due to the high excess air factors). If we look at the difference between the average flue gas temperature at the combustion chamber exit (600^{0}C) and at the chimney entrance (99^{0}C), it is on the average 60^{0}C smaller than in case of a 10 kW fire (see figure 3.7). Therefore less heat is transferred from the flue gases to the flue walls. As was shown in figure 3.4, this has resulted in much lower baking compartment gas temperatures.

The flue gas temperatures through the entire flue system during downdraft burning are much higher. This will, especially in the first part of the heating up period, show an increased heat transfer from the flue gases to the flue walls, since the temperature differences are larger. The higher flue gas temperatures at the chimney entrance provide a much larger draught, which increases the flow velocities and thus the heat transfer from the flue gases to the flue walls. This has resulted in a strongly reduced heating up time (see figure 3.4 and table 3.3).

In figure 3.7, the flue gas temperatures plotted at location 3 are the averages of the flue gas temperatures measured halfway the left and right side bottom flues. Although in general the differences in temperature between these two locations were small, it sometimes happened during conventional burning that differences of 30^{0}C were recorded (higher temperatures in the left side bottom flue). The reason for this is probably that when the oven was taken apart, the entrance from the right side bottom flue to the right back flue appeared to be less blocked with soot than the entrance from the left side bottom flue to the left back flue. This means that there was a larger flow resistance in the left flue, resulting in smaller flow velocities (flue gases will take the line of least resistance). Therefore less heat is transferred from the flue gases in the left side bottom flue to the flue walls, resulting in higher flue gas temperatures in this flue part.

Flue gas analyses (Combustion quality)
One of the parameters for describing the performance of a woodfired bakery oven is the quality of combustion. The combustion quality has an important impact on three variables.

(i) The heating time of the oven. For example: improper firing and insufficient air supply combined with a fuel bed build–up slow down the combustion. This will result in a reduction of the flue gas temperatures, deterioration of the heat transfer from the flue gases to the baking compartment, a reduced heating rate of the baking compartment and thus an increase in heating time.

(ii) The level of pollutants emitted from the chimney.

(iii) The amount of soot that is deposited in the flues. Not only will the deposit of soot reduce the heat transfer from the flue gases to the flue walls, but it can also block the flue passages. Thus the flow resistance will increase which reduces the amount of air drawn into the combustion chamber. This further deteriorates the combustion quality, leading to more soot deposit (vicious circle). Besides, the deposit of soot places considerable demands on the maintenance of the system.

Talking about the quality of combustion requires some understanding of the processes involved during combustion. According to Prasad and Verhaart (1987) the combustion of wood can be described as follows. As in all combustion, there is an initial heating phase, which lasts until the so–called ignition temperature is attained and the actual burning starts. For the combustion of wood this heating phase occurs in two steps. The first step involves heating to a temperature where chemical decomposition commences, liberating the volatiles. This is called the pyrolysis and takes place over a range of temperatures. After completion of this step, the combustion of wood takes two separate paths. One is the combustion of the more or less gaseous volatiles, the other is the combustion of the solid carbon. The volatiles will burn when the ignition temperature is reached. The burning depends on the extent of mixing with oxygen supplied and the temperature reached. The fixed carbon burns through a process of surface combustion. The interval of time elapsed between charging wood and the wood actually "catching" fire, depends on the size of the wood pieces and the moisture content. The larger the wood pieces and the higher the moisture content, the longer this time interval will be.

In section 3.2 we already discussed several combustion parameters (viz. excess air factor, stoichiometric air requirement, total air quantity, sensible heat losses) that can be used to characterize the combustion quality in this oven. Apart from these quantities, it will also be shown later on in this section, that the amount of soot deposit in the flues also provides complementary information on the combustion quality.

The composition of the flue gases coming out of the chimney is determined for both conventional and downdraft burning. To illustrate the combustion quality, the CO/CO_2 ratio's were computed to eliminate the dilutive influence of the excess air (CO is the result of incomplete combustion and CO_2 is the result of complete combustion).

In case of conventional burning, different power outputs and air supply settings were studied (see table 3.2) both for periods of 30 minutes as well as for periods of 3 hours. From the 3 hour experiments, of which the gas analysis results will be discussed later on in this section, the temperature curves were shown in figure 3.4. The experiments in which each air supply setting for each power output was kept constant for 30 minutes were done to get an indication about the most preferable number of opened primary and secondary air holes in the door of the combustion chamber. The first setting studied is: primary air supply (p) through 15 holes and secondary air supply (s) through 23 holes in the door of the combustion chamber. The second setting is: primary air supply through 5 holes and secondary air supply through 10 holes. The third configuration is: primary air supply through 2 holes and secondary air supply through 3 holes. During each setting, the chimney damper was kept fully open. The average CO/CO_2 ratios for the different power outputs and varying air supply settings are shown in figure 3.8.

It was found during the experiments with a 12.5 kW conventional fire, that a reduction in air supply made it very difficult to keep the fire burning. It was therefore decided to reduce the amount of wood, charged at one time. Still to keep the same power output, this also implied a reduction of the time interval between two charges. The new interval of time between two charges became 5 minutes.

Although only the average CO/CO_2 ratios are plotted in figure 3.8, the actual CO/CO_2 ratios showed quite some fluctuation, especially in the case of the 12.5

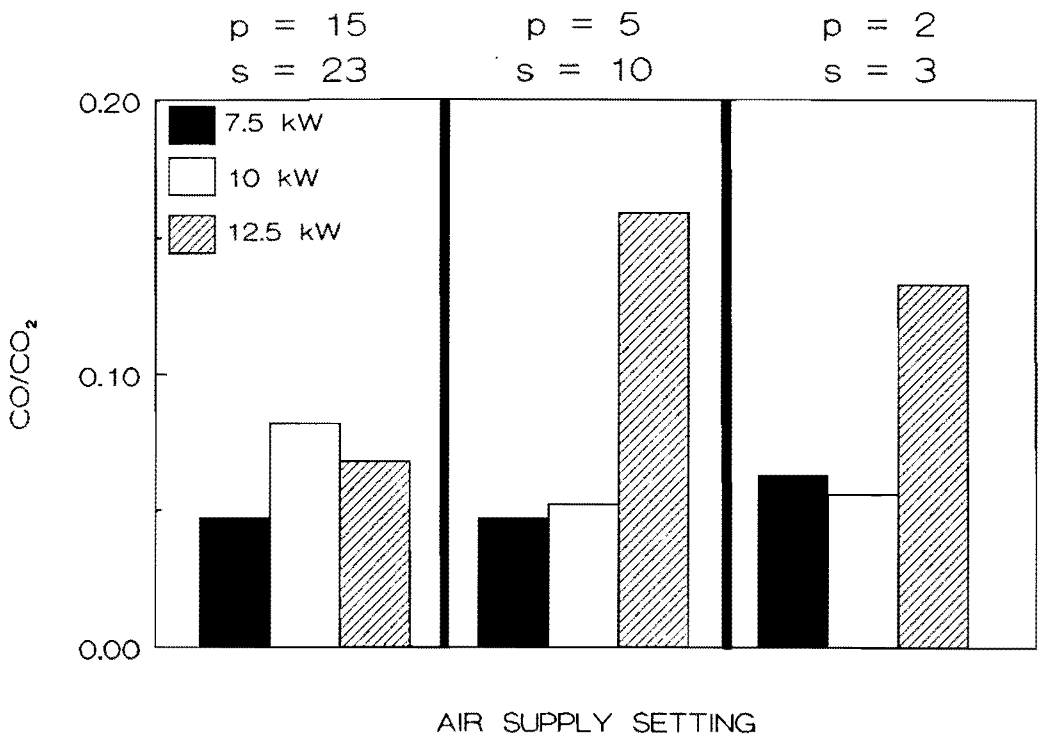

Figure 3.8: **Average CO/CO_2 ratios in case of conventional burning with varying power outputs and air supply settings.**

kW conventional fire. These fluctuations are caused by the irregular burning of wood. In the first few minutes after charging the wood, the CO/CO_2 ratio increases because an interval of time elapses before the ignition temperature is attained and the actual burning starts. During this period the combustion is poor and the CO content rapidly increases. At the time the wood actually catches fire, the combustion quality improves, causing a reduction of CO and an increase of CO_2 gas (CO/CO_2 ratio reduces). Figure 3.8 shows that for a 7.5 kW fire, there is not much difference in the average CO/CO_2 values for the various air supply settings. The average CO/CO_2 ratio for the first and second air supply setting is 0.047. For the third air supply setting the average CO/CO_2 ratio is 0.063. Except for the first setting (p = 15, s = 23), the CO/CO_2 ratios in case of a 10 kW conventional fire and various air supply settings are more or less the same and only marginally differ from the average CO/CO_2 ratios found with a 7.5 kW conventional fire and various air supply settings. The average CO/CO_2 ratio for the first air supply setting is 0.082. During the second air supply setting this average value is 0.052 and during the third setting it is 0.056. The increased average CO/CO_2 value for the first setting is the result of a sudden decay of the fire, which in this case resulted in a large reduction of the CO_2 content. In case of these two power outputs, a reduction in air supply area does not really seem to affect the combustion quality.

The results found in case of a 12.5 kW conventional fire and various air supply settings are quite different. In that case a reduced air supply clearly results in an increase of the average CO/CO_2 ratios. A total reduction in air supply inlet area of almost 61% (change from air supply setting 1 to air supply setting 2), more than doubles the average CO/CO_2 value (setting 1: average CO/CO_2 ratio = 0.068; setting 2: average CO/CO_2 ratio = 0.159; setting 3: average CO/CO_2 ratio = 0.133). The combustion quality became very poor. CO values up to 2% were measured. The CO_2 values measured, clearly remained below the values of the experiments with 7.5 and 10 kW conventional fires. Even a reduction in amount of wood charged per time interval (to reduce the fuelbed thickness and with that reduce the flow resistance) and simultaneously a reduction of this time interval between two charges to maintain the same power output, did not improve the combustion quality.

How should we judge the results presented in figure 3.8? When is the combustion quality acceptable from a health point of view and when not? Among other things, this depends on the way the flue gases are removed. In case the gases stay in the bakery atmosphere, none of the examined configurations is really acceptable, although it is difficult to give a limit. For gas–fired appliances a maximum CO/CO_2 ratio of 0.01 for normal operation is given (Sulilatu and Krist–Spit, 1985). But because a chimney is required anyway to achieve sufficient draught, it is more likely that the flue gases are removed through this chimney that ends outside the bakery. In that case, a maximum allowable CO/CO_2 ratio is more difficult to give, because the emission will not directly affect the users' health. Still the higher the CO/CO_2 ratio, the smaller the combustion efficiency. From a gas analysis point of view we therefore can conclude that for a 7.5 kW conventional fire the most preferable air supply setting is: primary air supply through 5 holes and secondary air supply through 10 holes. In case of 10 and 12.5 kW conventional fires, the optimum configurations are: primary air supply through 15 holes and secondary air supply through 23 holes. Curve 5 in figure 3.4 showed that the temperature rise in the baking compartment can be affected by the quality of combustion. However the same figure also showed that under normal conditions, different air supply settings only have marginal effect on the temperature rise in the baking compartment.

Apart from changing the air inlet area in the door of the combustion chamber, we also burnt some batches of wood with the chimney damper at an angle of 45^0 and a primary air inlet through 15 holes and a secondary air inlet through 23 holes. The nominal power output during this experiment was 7.5 kW. The results were frightening. CO values of more than 6% were measured and an almost immediate reduced temperature rise in the baking compartment was recorded. Especially in the first 5 minutes after the wood is charged, the combustion is very poor. Apparently the net draught that remains after positioning the chimney damper at a 45^0 angle, is much too small. Because even under these conditions the fuelbed thickness still reduces, the net draught increases after some time and more primary air can be sucked through the fuelbed. This improved the combustion quality somewhat. However closing the chimney damper during heating up of the oven is not advisable. Still it might be a good way to control the fire/power output during simmering or the baking of bread (see also section 3.7).

In the gas analysis experiments with conventional burning, described above, each setting examined has only been kept constant for about 30 minutes. But it is also interesting to study the behaviour for longer time intervals (3 hours). These 3 hour experiments can then be compared with the results of the downdraft burning experiments of which only 4 hour sessions were carried out. Since it was shown in figure 3.4 that the required baking temperature of 240^{0}C could not be obtained with a 7.5 kW conventional fire within 3 hours, only the flue gas analysis for the 3 hour experiments with the 10 kW and 12.5 kW conventional fires will be presented in comparison with the gasanalysis results obtained during downdraft combustion. During heating up, using downdraft burning, the power output is on the average 15 kW, while the air entrance is fully opened. During "simmering" (starts after about 90 – 100 minutes), the average power output is about 7 kW and the air entrance is kept closed for almost the entire period (except for charging fresh wood, the entrance door to the combustion chamber was opened).

Figure 3.9 shows the CO/CO_2 ratios in case of conventional burning with 10 kW (1, p = 15; s = 23) and 12.5 kW (2, p = 15; s = 23) and for 15 kW downdraft burning (3). In case of different air supply settings during conventional burning, an identical behaviour was noticed, only the absolute values found were different (the differences were smaller than 10%). The difference in combustion quality between downdraft burning and conventional burning is obvious. Compared to heating with downdraft burning (0 – 100 min.), the average CO/CO_2 ratio of a 10 kW conventional fire is about 3.6 times larger. During simmering, this difference increases to a factor of 5.8. The difference in average CO/CO_2 ratio between a 12.5

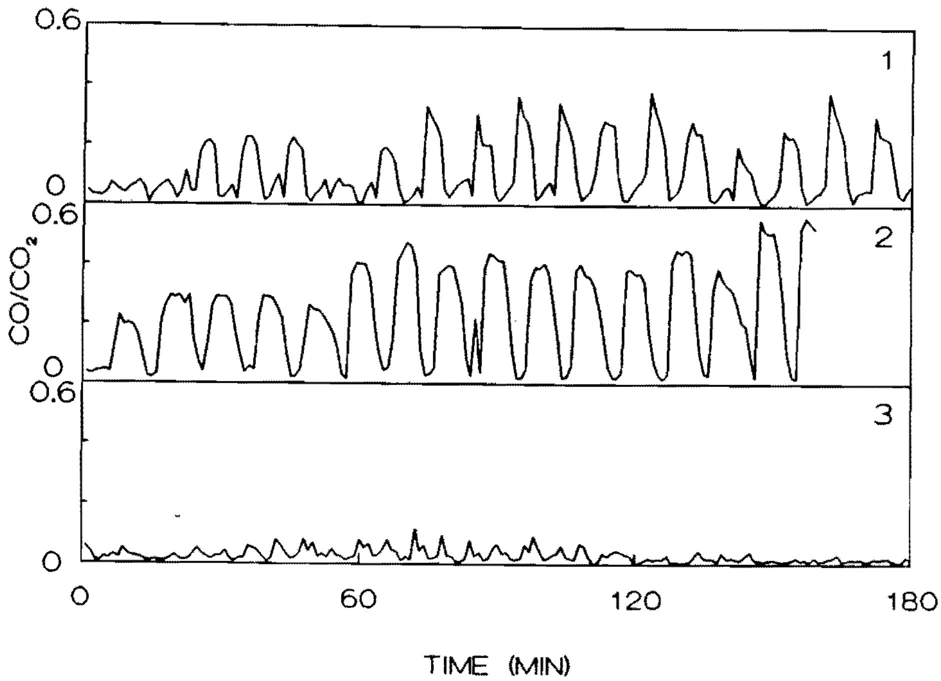

Figure 3.9: **CO/CO_2 ratios of the flue gases in case of: 1: 10 kW conventional burning ($p = 15$; $s = 23$); 2: 12.5 kW conventional burning ($p = 15$; $s = 23$) and 3: 15 kW downdraft burning.**

kW conventional fire and downdraft burning is even larger – a factor of 7.1 during the heating up period with downdraft burning and a factor of 11.3 during simmering. Three important factors are assumed to be responsible for the superior combustion quality during downdraft operation (Verhaart, 1989).

(i) Due to the higher flow velocities of air and volatiles, the mixing is much better than in conventional burning mode.

(ii) The combustion temperatures obtained are much higher (Khan, 1990).

(iii) The time interval between liberation of the volatiles and their reaction with oxygen is very short, compared to the conventional system. Thus there is little time for chemical reactions like condensation or polymerization to products which are much more difficult to convert into CO_2 and H_2O.

A fourth reason could be that the combustion during downdraft burning is essentially confined to a thin layer of fuel (Khan et al., 1989). Heat transfer to higher layers of fuel is restricted to radiation from burning char while the downdraft flow of air cools the fresh fuel. In the thin layer of burning fuel, the air gives up only part of its oxygen. Below the grate it mixes with volatiles at a high temperature, resulting in a much cleaner combustion.

The CO/CO_2 curves in figure 3.9 for a 10 and 12.5 kW conventional fire show that the minimum values reached remain reasonably constant. The peak values, especially for the 12.5 kW fire, show a slight increase during the heating process. In case of the 12.5 kW conventional fire, the width of the peaks is larger than the width of the troughs. This indicates that most burning of one charge takes place under poor conditions. This becomes even more clear if we study the excess air factors for these experiments. The excess air factors were computed with the application of equation 3.4b and are shown in figure 3.10.

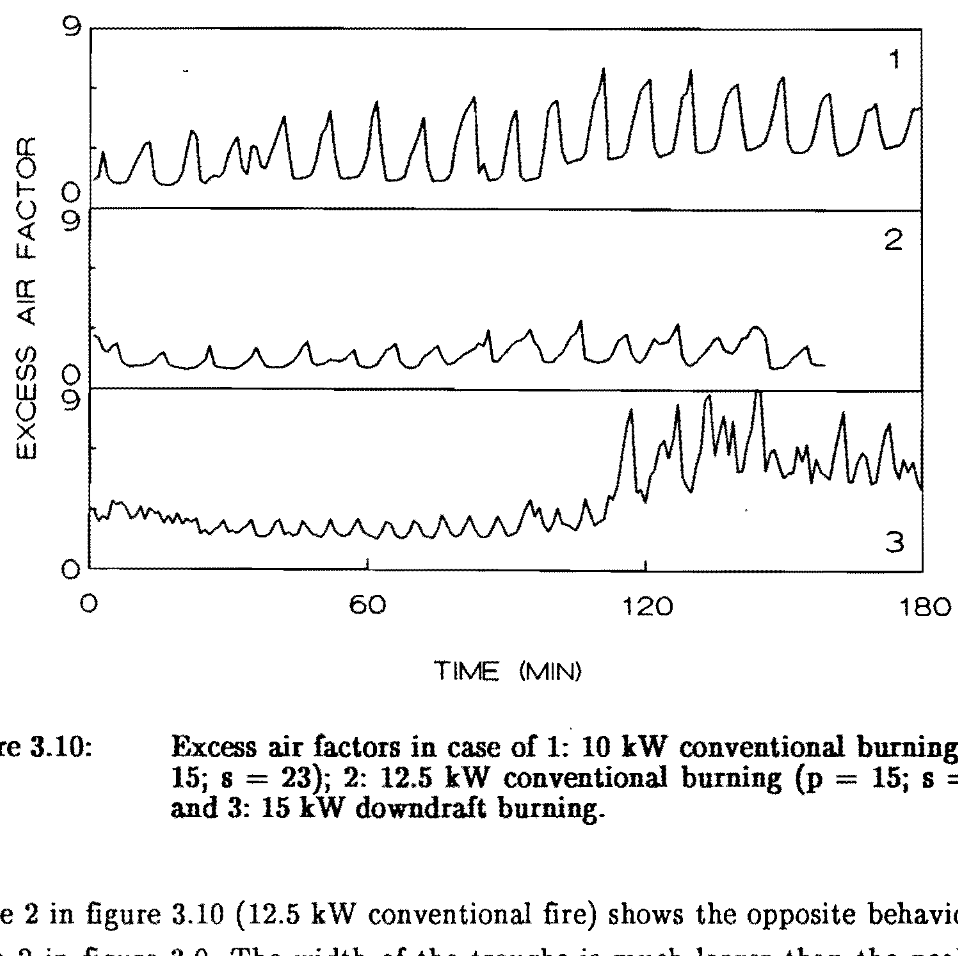

Figure 3.10: **Excess air factors in case of 1: 10 kW conventional burning (p = 15; s = 23); 2: 12.5 kW conventional burning (p = 15; s = 23); and 3: 15 kW downdraft burning.**

Curve 2 in figure 3.10 (12.5 kW conventional fire) shows the opposite behaviour of curve 2 in figure 3.9. The width of the troughs is much larger than the peaks. In the early part of the heating period, the excess air factor remains below 1.2. Later on this slightly increases (The average excess air factor for the total duration of the experiment is 1.6). It becomes clear from figure 3.10 and the CO/CO_2 ratios found in case of a 12.5 kW fire, that the air supply conditions are insufficient, especially in the first few minutes after a fresh amount of wood is charged. The fuelbed thickness becomes too large, which blocks the passing of primary air through the fuelbed. The primary air is completely depleted of oxygen at the time it emerges from the fuelbed. In the thick fuelbed, CO_2 will be partly reduced to CO, which lowers the gas temperatures since this reaction is endothermic. Supply of more secondary air is not likely to improve the situation, because it will only further reduce the temperatures in the combustion chamber leading to further deterioration of the combustion quality.

During conventional burning with a 10 kW fire, the average excess air factor is 3.2. The character is however much more peaky. Especially at the end of a time interval, the excess air factors largely increase. This is probably due to the exhaustion of the fuelbed. During downdraft combustion, the heating period (0 – 100 min.) shows a very constant excess air factor (average is 2.3). These results, combined with the CO/CO_2 ratios for downdraft burning, give reason to believe that the combustion is very regular. During the "simmering" period, the excess air factor increases sharply (average is 6.1). Although the air entrance area was kept completely closed after wood was charged, the chimney draught still forces a lot of unused air being sucked through the system. In this case, a chimney damper might have been useful to reduce the amount of excessive air.

With equation 3.6 (section 3.2), the sensible heat losses from the chimney entrance during these three experiments are computed and shown in figure 3.11. The figure clearly illustrates the general behaviour. During heating up with downdraft burning, the sensible heat losses remain almost constant. On the average, they are higher than with conventional fires of 10 and 12.5 kW. This is primarily due to higher flue gas temperatures measured at the chimney entrance. During the simmering period, the sensible heat losses largely increase, which is primarily the result of the increased excess air factor. The use of a chimney damper would probably also reduce the sensible heat losses during the "simmering" period.

In case of conventional fires, the sensible heat losses increase during the experiment, especially in case of a 10 kW fire. This is the result of both an increase in the flue gas temperatures at the chimney entrance and an increase in excess air factors, as the experiments went along. The increase in flue gas temperature can be explained on the basis of the fact that the heat extraction from the flue gases reduces with time. An explanation for the increase of the excess air factor is less clear. The increase in fluctuation during the second half of the conventional

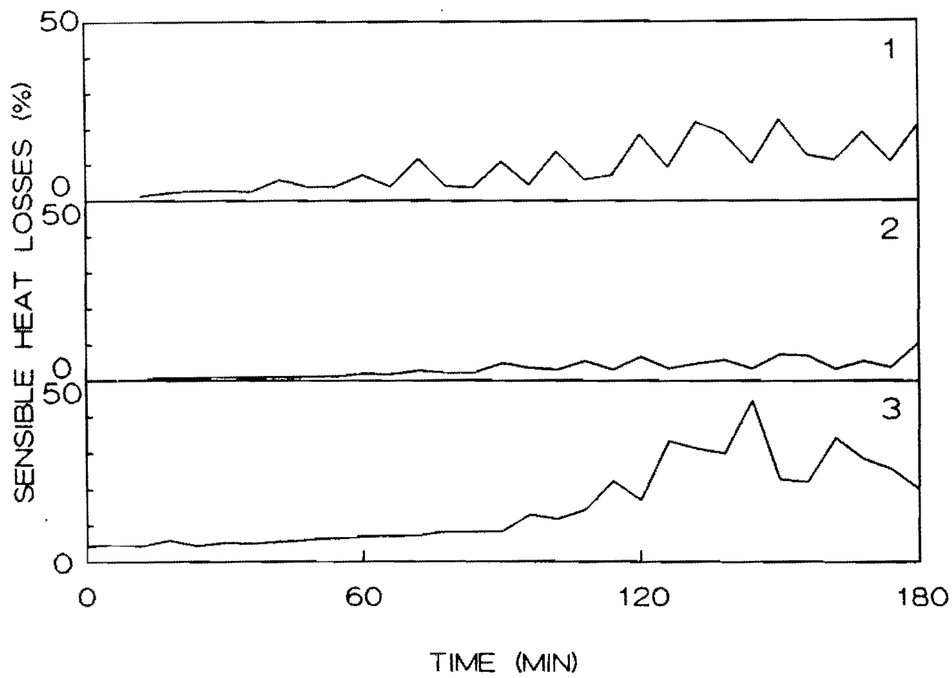

Figure 3.11: **Sensible heat losses (%) at the chimney entrance in case of 1: 10 kW conventional burning (p 15; s = 23); 2: 12.5 kW conventional burning (p = 15; s = 23); and 3: 15 kW downdraft burning.**

burning experiments, is the result of a continuously increasing difference between the minimum and maximum values of the excess air factors in these experiments.

Soot Deposit

The combustion of hydro–carbons can result in intermediate compounds that combine in such a way that carbon rich solid particles develop. These particles are named soot (Günther, 1974). The continuation of soot development is essentially explained by means of the agglomeration of combustion products. For that many theories exist, which all state that C, C_2, C_3, aromatic compounds, polycyclical hydro–carbons and acetylene are the most important intermediate compounds that originate. In case the starting molecule has a cyclical shape, it is assumed to remain unchanged during soot development. In case the starting molecules are chain–shaped, it is assumed that C_2H_2, C_4H_2 and C_6H_2 develop, being intermediate compounds in the formation of larger molecules. These intermediate compounds can combine with radicals like CH, CH_2 and CH_3. These develop new radicals which in the end, via several chemical steps (poly–acetylene, cyclic compounds), become soot particles of different sizes but all with a globular shape. The globular soot particles, present in the flame, stick together in the shape of a string of pearls. The diameters of the globular soot particles are in the range of $0.01\mu m - 0.05\mu m$. In richly sooting flames, particles with a diameter upto $0.2\mu m$ can be found. The sizes of the developed particles depend little on the kind of fuel used. Each globular particle consists of about 10^4 crystals. Each crystal consists of about 5 to 10 carbon strings. Each string has about 100 C atoms.

The amount of soot that develops during the combustion of a fuel depends on several factors.

(i) The temperature during combustion. As a result of high temperatures, the hydro–carbons lose more and more hydrogen atoms. In the end only carbon remains.

(ii) The molecular structure of the fuel. A fuel with small hydro–carbon molecules produces less soot than one with large hydro–carbon molecules.

(iii) The availability of oxygen. In case of combustion with enough excess air and proper volatile/air mixing, there will hardly be any soot. Slow mixing, high temperatures and lack of oxygen show an increased development of soot.

(iv) Finally, the pressure in the system.

Richly sooting flames have a yellow color, while non–sooting flames have a soft blue color because of the combustion of CH, C_2 radicals and CO.
In large combustion chambers, the development of soot as an intermediate compound can be positive, because it improves the heat release by radiation from the flames. Therefore soot formation and combustion should be arranged in a way that at the end of the combustion period, no soot has remained unburnt. Otherwise this not only means a loss of unused energy, but also pollution of the equipment and the surrounding.

The soot which will deposit in the flues can have either a positive or a negative effect on the heat transfer from the flue gases to the baking compartment floor and ceiling. This strongly depends on how the soot is deposited.

<u>Positive effect</u>
A smooth thin layer of soot can improve the heat transfer because it approaches the situation of a black surface which shows an increased absorption of heat.

<u>Negative effect</u>
In case the layer of soot is very irregular and/or thick, the heat transfer by conduction will reduce because the soot now acts as an insulator.

Between some of the experiments it was necessary to carry out some repair work to the oven. The repair activities that took place during the experiments with conventional burning, revealed a substantial amount of soot in the flues. For the laboratory model, this soot was removed simultaneously with the repair work. In the prototype some provisions will be available to clean the flues from soot, without having to break down the oven. It is however desirable that the amount of soot deposited requires no more cleaning than once every two weeks, preferably once a month.

The amount and way of soot deposition has been studied after 30 hours of conventional burning experiments and 30 hours of downdraft burning experiments. Figure 3.12 shows the laboratory model seen from the back before it was taken

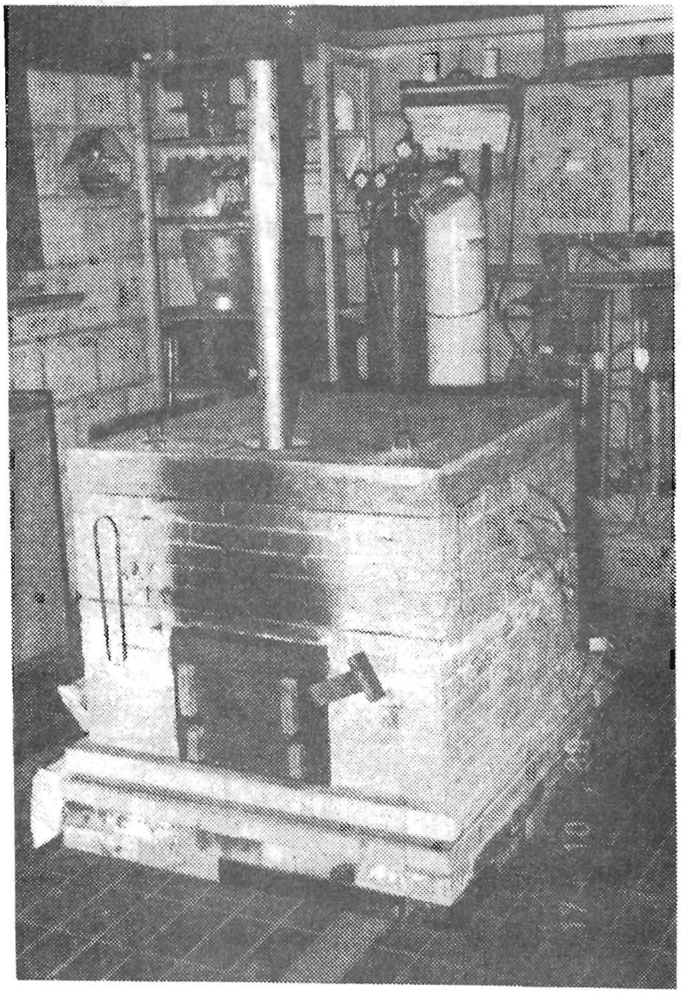

Figure 3.12: The laboratory model seen from the back.

apart. When 30 hours of firing were finished, the thick concrete top plate and part of the back wall were removed. Thus the top and back flues could be examined for

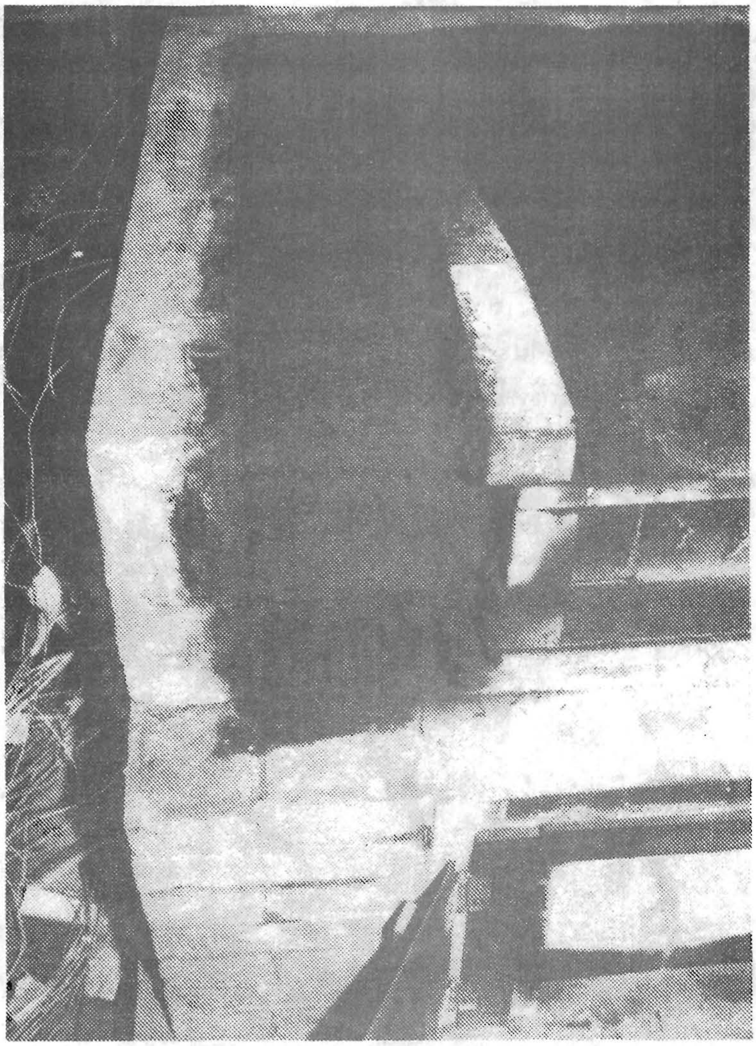

Figure 3.13: Right side top flue, part of the central top flue and the right back flue after 30 hours of conventional burning.

the presence of soot. Figure 3.13 shows the right side top flue, part of the central top flue and the right back flue after 30 hours of conventional burning. Figure 3.14 provides a closer look at the right side top flue. There is a very irregular and sometimes also a thick layer of soot. The heavily covered wires, crossing the right top flue (figure 3.14), are thermocouple wires which are brought in to measure baking compartment gas temperatures. The wires have acted like nuclei for piling up of soot.

A close up view of the left back flue in figure 3.15 shows that, after 30 hours of

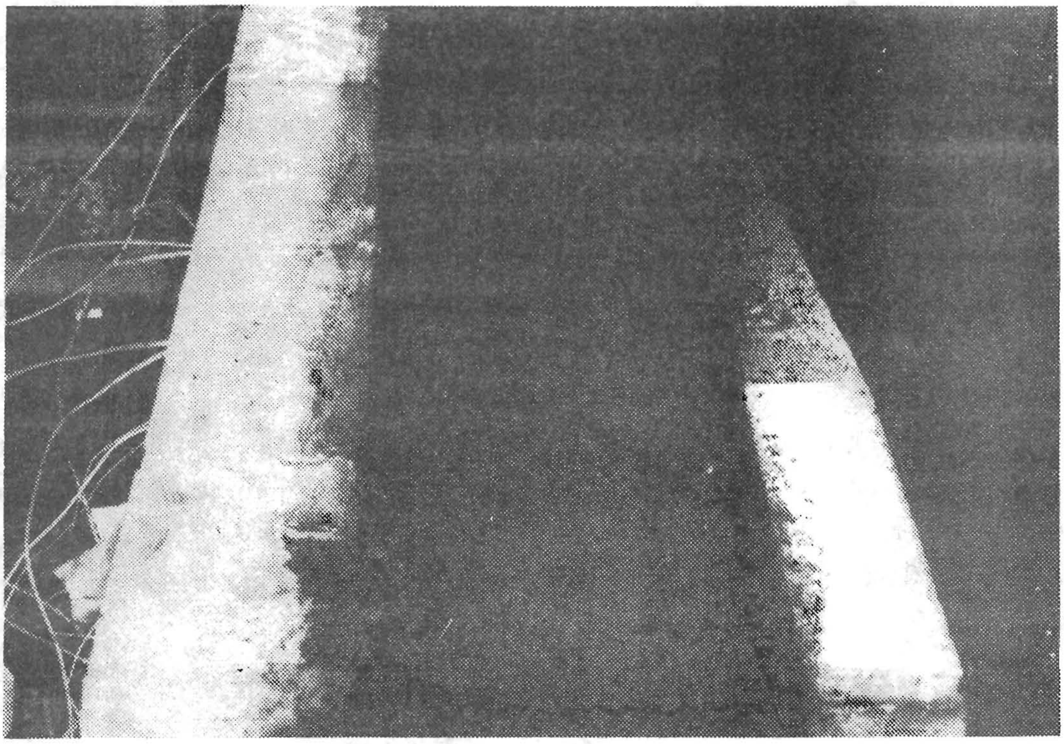

Figure 3.14: Right side top flue after 30 hours of conventional burning.

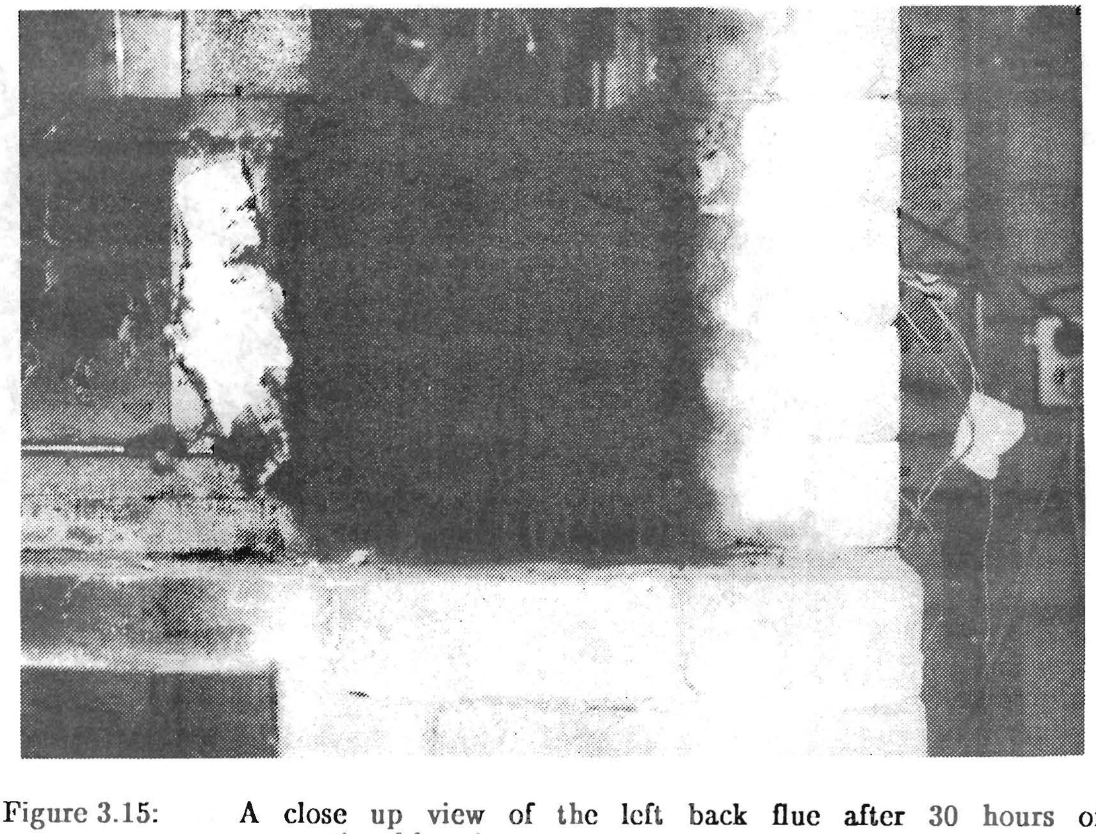

Figure 3.15: A close up view of the left back flue after 30 hours of conventional burning.

conventional burning, the entrance from the left side bottom flue to the left back flue is partially blocked.

When the top plate and part of the back wall were removed after 30 hours of downdraft burning experiments, the sight was totally different. Only the central top flue and partially the left and right side top flues were covered with a very thin and smooth layer of soot. In the back flues, no soot had deposited at all. Unfortunately, the photographs taken of these flue parts, were unfit for presentation.

When the baking compartment ceiling (together with the top flues), the back wall of the baking compartment and the bottom plate (baking compartment floor) were removed after 30 hours of experiments with conventional burning, the situation in the bottom flues showed even worse than in the top flues. Figure 3.16 shows a close up of the left side bottom flue seen from the back.

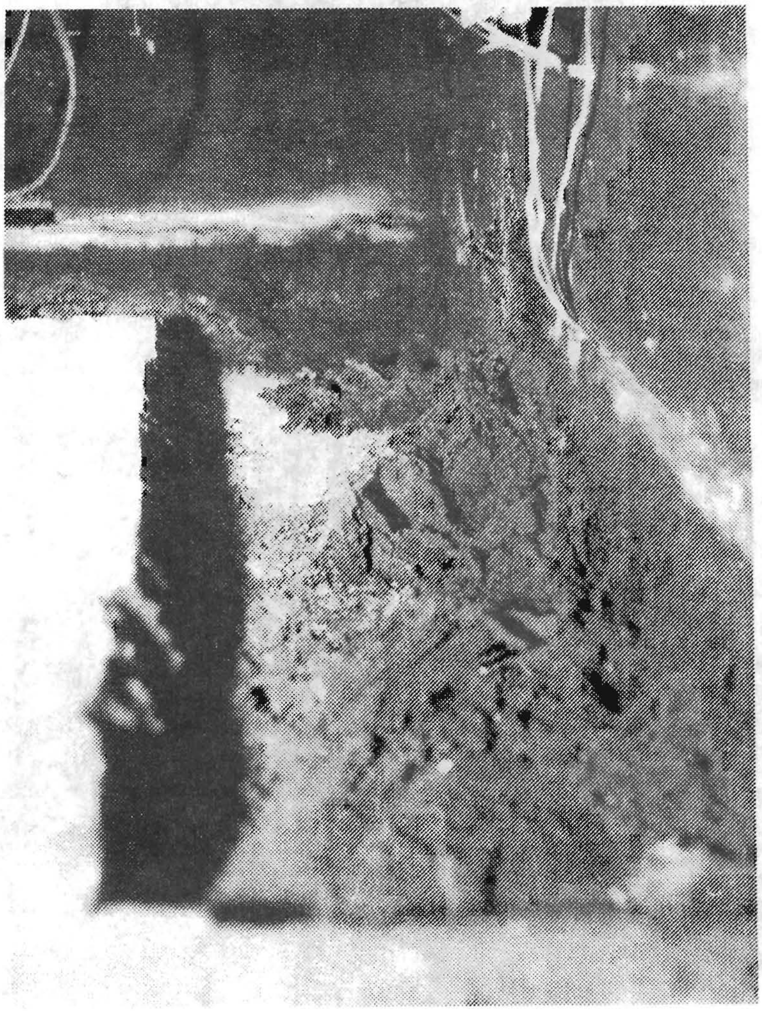

Figure 3.16: Close up view of the left side bottom flue after 30 hours of experiments with conventional burning

The photograph shows that an enormous amount of soot is deposited in the right half of the flue (against the left outer wall). It is the same in the right side bottom flue. This soot deposit has reduced the passage area of the flue gases tremendously. Because the flue gases will take the line of least resistance, the soot once deposited will be virtually left untouched and not be burnt away by the passing flue gases It turned out that after 30 hours of experiments with downdraft burning, the left and right side bottom flues remained clean. Only some ash particles were found. Figure 3.17 shows the right side bottom flue after 30 hours of downdraft burning.

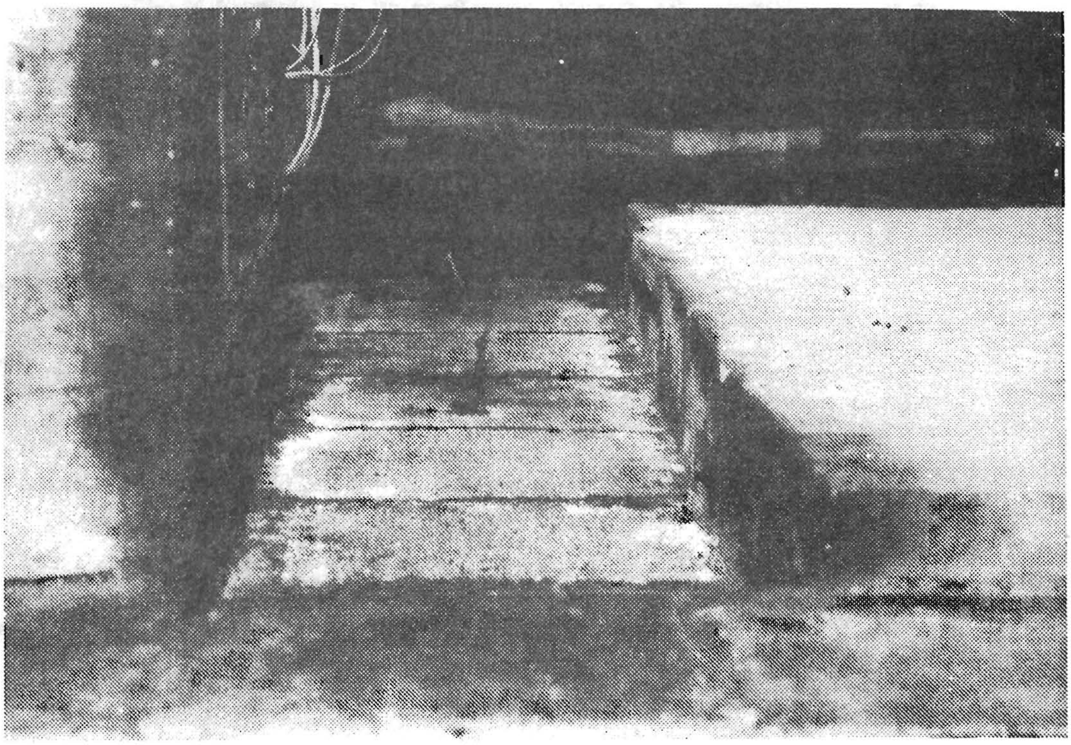

Figure 3.17: Right side bottom flue after 30 hours of downdraft burning.

The only channel that remained free from any soot deposit after the conventional burning period, was the central bottom flue. Apparently the flue gas temperatures reached in this part of the flues were high enough to prevent soot formation. Figure 3.18 shows a picture of the central bottom flue after this conventional burning period, taken through the inlet opening for wood, looking upwards into the flue channel. There is only some ash deposit. It is hardly a surprise that the central bottom flue also remained clean after the experiments with downdraft burning.

In figure 3.19 the bottom surface of the concrete plate used as the floor of the baking compartment during conventional burning is shown. The place where the

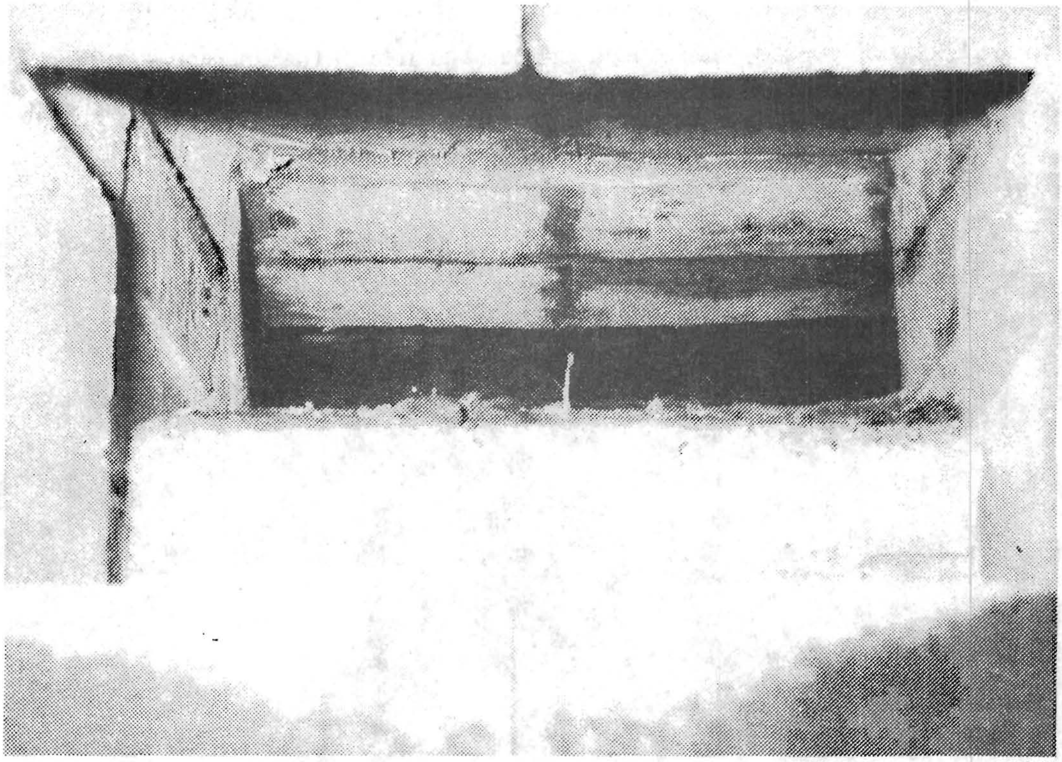

Figure 3.18: **View into the central bottom flue after 30 hours of conventional burning.**

concrete bottom plate rested upon the insulation layer between the central bottom flue and the baking compartment floor, corresponds with the light colored rectangular at the bottom surface of the plate. The photograph shows that as soon as the flue gases leave the central bottom flue and enter the bottom side flue, the soot no longer gets burnt away. The soot hangs down in thick pickings from the roof of the flue. The whole sight is comparable with stalactites and stalagmites in caves. The bottom surface of the concrete plate used as the baking compartment floor during downdraft burning, remained clean.

A root cause of the large soot formation during conventional burning is probably an insufficient oxygen (air) supply through the holes in the door of the combustion chamber combined with too thick a fuelbed. Especially with 12.5 kW conventional fires, the excess air factors were much too small for proper combustion. However, it is also very likely that the incoming air, especially the secondary air, gets insufficiently mixed with the volatiles and will only further cool the gases emerging from the fuelbed. The main reason for this is probably that the combustion

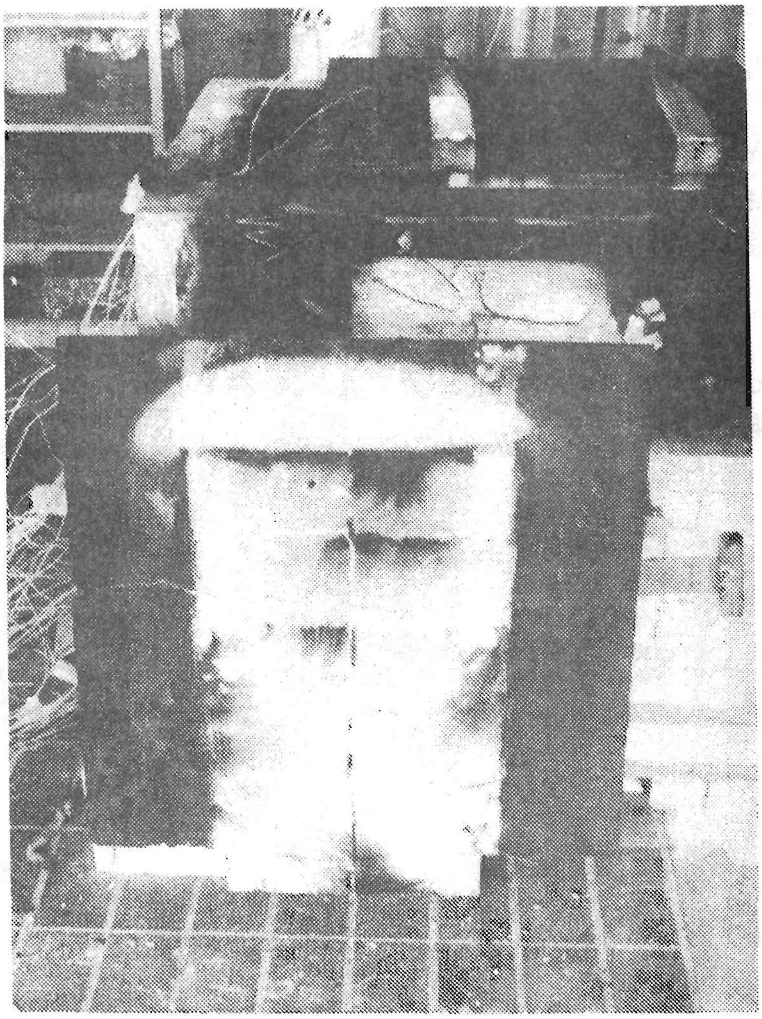

Figure 3.19: Bottom surface of the concrete plate used as the baking compartment floor during conventional burning.

chamber exit (start of the central bottom flue) is only about 15cm above the grate. This means that the liberated volatiles will only stay in the combustion chamber for a very short time and almost immediately disappear into the central bottom flue. The time for mixing of the volatiles with air is therefore very short.

When downdraft burning was used, hardly any soot was found in the flues. Only a very smooth thin layer over part of the top flues was found (this might even have contributed to the very uniform temperature distribution found on the ceiling). The volatiles burn away before intermediate compounds that play a part in the formation of soot, can originate. Another reason might be that due to the much higher flue gas temperatures, soot that is formed anyway, immediately gets burned away. The flue gas temperatures in the top flues are probably not high enough for burning of the soot.

3.7 Baking Character of the Oven. Temperature Rise and Distribution in the Baking Compartment during the Baking of Bread

Introduction

Although the experiments carried out with the laboratory model are primarily concentrated on the thermal qualities of the oven (heating rate, temperature distribution, combustion quality), we also took the opportunity to obtain some data about the baking character of the oven. Therefore two baking experiments were carried out. The first one, using conventional heating, was done directly after the pre–heating periods described in section 3.4. The baking experiment with downdraft combustion was done after the downdraft experiments, previously described, were finished.

The baking character of an oven is best described as the thermal contribution to the overall quality of the finished product, assuming that the production process until the baking has been properly carried out and that the proper raw materials were used. The problem with this definition is the word quality. An acceptable quality, in this case the quality of bread, can be interpreted differently from one person to another. Of course extreme baking results like for example completely burnt bread or half baked bread, will not be acceptable to anyone. However some people prefer bread well baked with a very brown crust and a substantial structure, others like light colored bread with a soft structure. We therefore say that quality is what the consumer expects. We only will give some general guide–lines to judge the character of an oven.

(i) The temperatures in the baking compartment should be high enough (between 230^{0}C and 275^{0}C depending on the kind of bread that is baked) to prevent the baking to be extended too long, causing the bread to dry out. This will affect the texture, taste and color of the bread unfavorably.

(ii) Too high baking temperatures (>300^{0}C) are undesirable. The crust will burn before reactions like gelatinization of the starch and denaturation of the gluten could take place. A minimum duration of about 10 to 15 minutes for a 500g loaf is required for these reactions to take place properly.

(iii) The temperature distribution in the baking compartment should be as uniform as possible, although some deviation in temperature, resulting in different gradations of browning, does not necessarily make the bread unsellable (Think of the people who prefer a very brown colored crust).

(iv) When the bread is put into the oven to be baked, there will be a certain drop in baking compartment temperature during the first few minutes of the baking process due to the extraction of heat by the relatively cold dough pieces. Depending on the accumulation power of the baking compartment floor, ceiling and walls, a temperature drop between 20°C and 50°C is to be expected. The baking compartment temperatures should not drop below the minimum required baking temperature of 230°C and this can be achieved by using enough mass for accumulation of heat.

(v) The temperatures of floor and ceiling may differ provided that they are uniform. In that case heat transfer by means of natural convection will be uniform and not affect the bread quality unfavorably.

Apart from these guide-lines, which only will provide a general statement about the baking quality of the oven, certain figures can be used for the assessment and comparison of different types of ovens (Schmitt and Siemers, 1985). These figures are only meaningful in case the oven is used in a real production situation and not for experiments as described here. Still it seems useful to mention these figures in this context, since they can be used in testing any type of oven during a real production situation. The figures are:

(i) oven load factor

This figure describes the utilization of the given baking area if one load of bread is baked.

$$O_f = \frac{N_{l_b} \; M_{fl}}{A} \tag{3.9}$$

where:

A = baking area (m^2)
M_{fl} = mass of flour per loaf (kg)
N_{l_b} = number of loaves per batch.
O_f = oven load factor (kg flour/m^2 baking area).

(ii) oven load efficiency

In contrast to the oven load factor, this value includes the influence of the production process time and the baking parameters on the actual use of the given baking area. Normally two figures are computed. One value (higher value) includes only the actual time spent on loading, baking and unloading (t_b). The other figure (lower value) also includes the time spent on heating of the oven (t_T).

$$\text{High: } O_e = \frac{N_{T_l} \; M_{fl}}{t_b \; A} \qquad (3.10a)$$

$$\text{Low: } O_e = \frac{N_{T_l} \; M_{fl}}{t_T \; A} \qquad (3.10b)$$

where:

M_{fl} = mass of flour per loaf (kg).
N_{T_l} = total number of baked loaves.
O_e = oven load efficiency (kg flour/m² baking area.hour).
t_b = time spent on baking including loading and unloading (h).
t_T = time spent on baking, loading and unloading, and on heating of the oven (h).

(iii) specific energy consumption

This figure indicates the amount of energy used to process 1kg of flour.

$$S_c = \frac{M_w \; B_f}{N_{T_l} \; M_{fl} \; 1000} \qquad (3.11)$$

where:

B_f = as–fired calorific value of the fuel (kJ/kg).
M_{fl} = mass of flour per loaf (kg)
M_w = mass of wood used in the baking process, including the heating up period (kg).

N_{Tl} = total number of baked loaves.
S_c = specific energy consumption (MJ/kg flour).

(iv) baking efficiency

The figure describes the efficiency of the process based on the bread weight that has come out of the oven. It gives the ratio of the amount of energy needed in theory to bake the total number of loafs produced and the actual energy supplied by the fuel. The theoretical amount of energy required to bake 1 kg of bread is 570 kJ.

$$B_e = \frac{570 \ N_{Tl} \ M_b}{M_w \ B_f} 100\% \qquad (3.12)$$

where:
B_e = baking efficiency (%).
B_f = as–fired calorific value of the fuel (kJ/kg).
M_b = mass of one baked loaf (kg).
M_w = mass of wood used in the baking process, including the heating up period (kg).
N_{Tl} = total number of baked loaves.

Baking experiment
Since only two baking experiments were carried out (one with conventional heating and the other by means of downdraft heating), the results should be treated as tentative. The only purpose of presenting these results at all, is to give a provisional judgment of the baking characteristics of the oven on the basis of the general statements mentioned earlier in this section. It is expected that the information might be of some use in drawing up the final design parameters of the prototype.

The temperatures are measured at the same locations as described in section 3.3 (figures 3.1 and 3.2). One exception was made. During the baking experiment with downdraft burning, the floor temperatures could not be recorded because there was no possibility of installing the thermocouples without them being damaged by the baking plates. The flue gases were not analyzed during these baking sessions. The experimental conditions are listed in tables 3.5 and 3.6.

Time (min)	*Number of holes p. air opened*	*Number of holes s. air opened*	*Position chimney damper*	*Remarks*
0	15	23	90°	
8	5	10		
143	1		45°	Oven is at its baking temperature
162		8		
176				First load of dough is put in to bake (2.8 kg)
186				Bread is taken out of the oven
187	4			
206	2			
213				Second load of dough is put in to bake (2.0 kg)
223				Bread is taken out of the oven
241	4			
250			67.5°	
255			0°	Third load of dough is put in to bake (1.9 kg)
263			45°	
265				Bread is taken out of the oven
274			90°	
283				Fourth load of dough is put in to bake (1 kg)
293	0	0	0°	Bread is taken out of the oven

Table 3.5: Operational conditions during the baking experiment with conventional heating. An empty rectangle indicates that the situation remains unchanged. Chimney damper: position 90° is vertical, position 0° is horizontal.

In case of the baking experiment with conventional firing, the oven was heated with a 10 kW fire. Because this experiment was one of the first carried out, little was known about the temperature distribution in the baking compartment. During baking the floor temperatures appeared to be much too high. Thus the insulation layer between the central bottom flue and the baking compartment floor was not installed till after this baking experiment. A consequence of this was that the heating rate of the oven during this experiment was found to be higher than for the 10 kW conventional heating experiments done after that, and as such is not representative for the heating rate of this oven.

Time (min)	*Circumstances*
0	Start firing
100	Temperature baking compartment is 240°C
103	First load of dough pieces is put in for baking *
122	Bread is taken out of the oven
138	Second load of dough pieces is put in for baking
151	Bread is taken out of the oven
169	Third load of dough pieces is put in for baking
180	Bread is taken out of the oven
189	Fourth load of dough pieces is put in for baking
202	Bread is taken out of the oven
225	Fifth load of dough pieces is put in for baking
238	Bread is taken out of the oven
261	Sixth load of dough pieces is put in for baking
277	Bread is taken out of the oven
288	Seventh load of dough pieces is put in for baking
303	Bread is taken out of the oven
321	Three cakes are put in for baking
356	Cakes are taken out of the oven

* Each load of dough pieces weighed about 2kg.

Table 3.6: Time schedule during the baking experiment with downdraft burning

When baking temperature was reached, the power output was strongly reduced, thus maintaining a constant baking compartment temperature until the first load of dough pieces was put into the oven for baking. Apart from a power reduction, the chimney damper was positioned in a 45^0 angle to reduce the draught and by that reduce the sucking in of cold air through the holes in the combustion chamber door. During downdraft operation, the power output was gradually increased (from 8 kW at the beginning till 23 kW at the time baking temperature was reached).

Figure 3.20 shows the average baking compartment gas temperatures for both experiments. The use of an average temperature was acceptable because the temperatures measured at the 5 locations in the baking compartment gas (figure

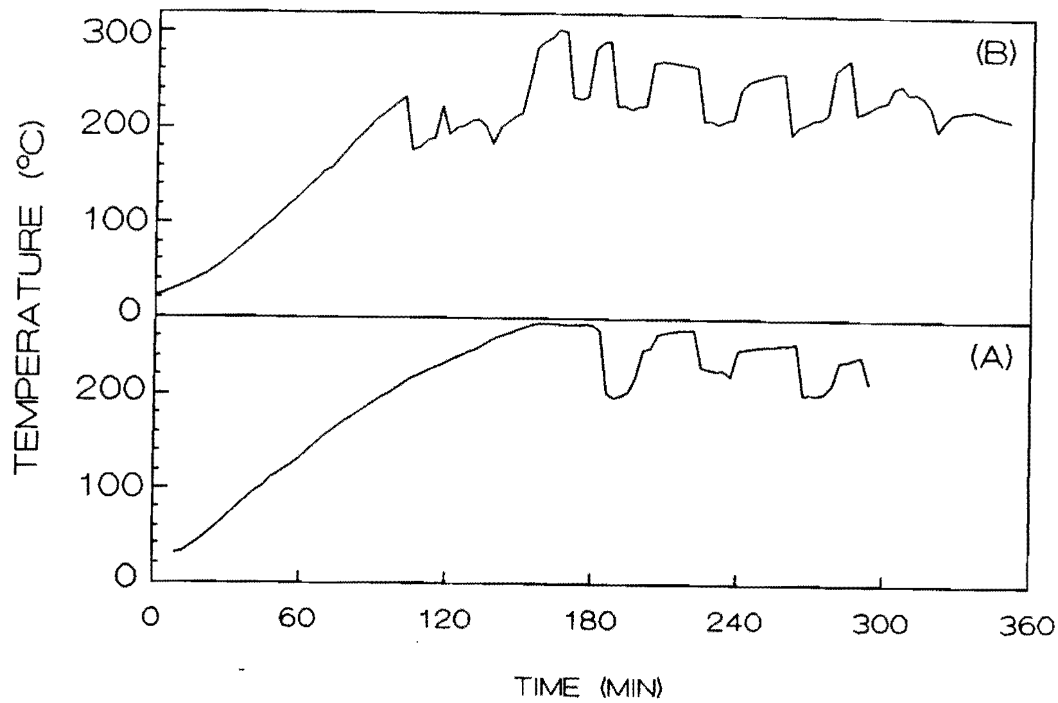

Figure 3.20: **Average baking compartment temperatures during the baking experiments with conventional heating (A) and during downdraft heating (B).**

3.1) showed little difference. The figure clearly shows that as soon as a load of dough pieces is placed into the oven for baking, the temperatures drop. The heat absorption by the relatively cold dough pieces is very large. It also happens that the temperature drops below the minimum allowable baking temperature. During baking with downdraft burning, the baking was started too early. The average baking compartment gas temperature had just reached 240°C when the first load was put into the oven. When baking was started at a higher temperature, the drops in temperature were approximately the same, but they remain acceptable. In both baking sessions, the power output was instantly increased after these temperature drops were noticed, to obtain a higher baking temperature. This appeared not to be possible during baking of one charge. The oven responded very slowly. During downdraft heating, the reaction was a little better due to a higher possible power rating. The slow reaction is due to the large oven mass on the one hand and the continuous absorption of heat by the bread on the other hand.

Figure 3.21 shows the floor and ceiling temperatures during the baking experiments. The temperature curves in (A) are recorded during conventional heating. (B) shows the average ceiling temperature during downdraft heating,

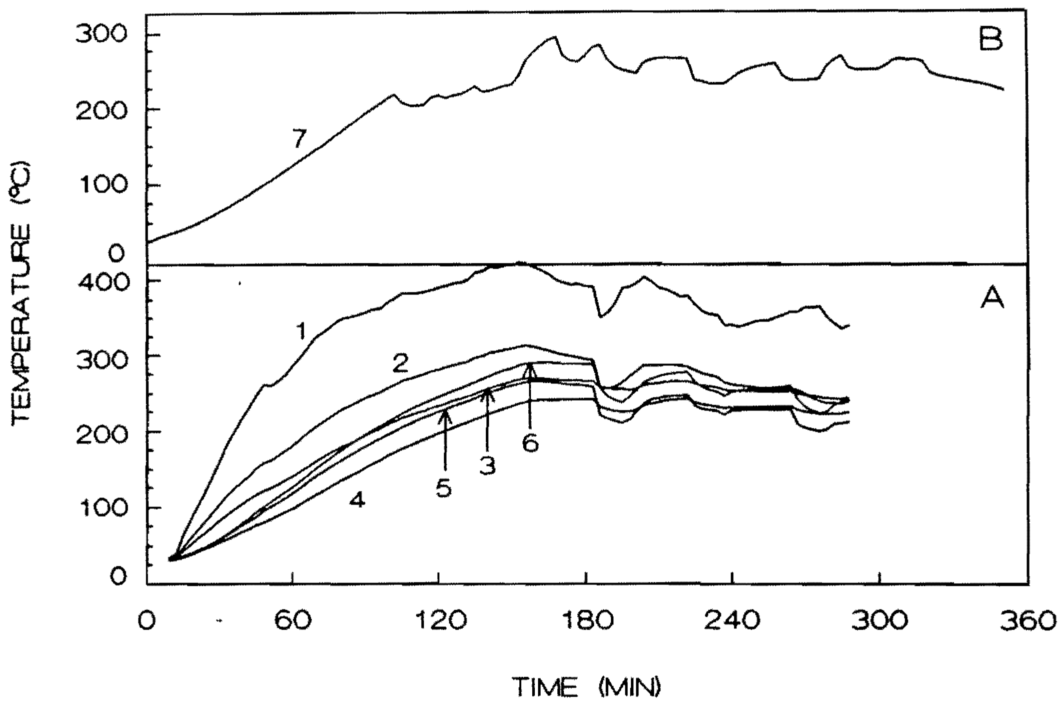

Figure 3.21: **Temperatures on the baking compartment floor and ceiling during baking with conventional heating (A) and downdraft heating (B).**
1: $T_{fl,C}$; 2: $T_{fl,LF}$; 3: $T_{fl,RB}$; 4: $T_{c,RB}$; 5: $T_{c,LF}$; 6: $T_{c,C}$; 7: $T_{c,average}$.

because the difference among the three locations were very small (floor temperatures were not recorded).

The temperature drops of the baking compartment gas noticed during the baking experiments might be considered too large (figure 3.20). This could be an indication for an insufficient accumulation power of the floor and the ceiling. Still increasing the mass of the floor and/or the ceiling (increase of accumulation mass) will also result in an increase of the required heating time. However it was also shown in figure 3.21 that the temperature drops of the floor and ceiling itself were much smaller. The temperatures remained above the minimum allowable temperature of 230°C. Therefore a better solution might be not to increase the accumulation mass, but to continue heating till a higher initial baking temperature (± 260 – 270°C) and after the oven has been left to rest for some time (to increase the uniformity of the floor and ceiling temperatures), maintain a higher power output at the moment the dough pieces are put into the oven for baking.

The bread quality during the experiment with conventional heating was reasonable except for those dough pieces which were placed in the center of the baking compartment floor. The under–surfaces of these pieces were strongly burnt. This was the result of exceptionally high temperatures at this part of the floor (± 375°C at the time the first load was placed into the oven). These high floor temperatures were the reason for increasing the insulation layer between the central bottom flue and the baking compartment floor. During baking of the other charges, the dough pieces in the center of the baking compartment floor, were placed on a metal sheet (0.001 m thickness). The quality of the bread improved very much. Burning of the under–surfaces was hardly noticed. The reason for this is that the small air layer between the metal sheet and the baking compartment floor increases the heat resistance and by that reduces the temperatures at the under–surfaces of the bread. The metal sheet increases the heat resistance marginally and the absorbed amount of heat by the sheet is also negligible compared to the heat absorption by the dough pieces. To prevent any problems in removing the bread pieces from the metal sheets, the sheets should be properly greased.

During baking with downdraft heating, all dough pieces were baked on metal sheets. The quality of the bread was better than in the first experiment. This is probably due to a more uniform temperature distribution in the baking compartment. Besides, several times during the baking period, water was injected into the baking compartment. The water immediately evaporated and partially condensed on the relatively cold dough surfaces. This prevents the dough pieces from drying out, causing a premature rupture of the crust. It will also have a positive effect on the browning of the crust, since water is an essential compound in the browning reactions named "Maillard" reactions.

3.8 Cooling Down Characteristics of The Oven.

After baking is stopped, the baking compartment walls still contain a certain amount of heat. This accumulated heat is lost to the surroundings, which results in a temperature drop of the baking compartment. The amount of accumulated heat in the walls after a cooling down period directly affects the required amount of fuel to bring back the oven to its required baking temperatures (Schutte et al., 1988). Therefore this release of retained heat to the surroundings should be prevented as much as possible. The remaining amount of accumulated heat and thus the cooling rates of the side walls depend on several factors.

(i) Composition of the baking compartment walls.
(ii) Mass of the baking compartment walls.
(iii) Construction and geometry of the baking compartment walls.
(iv) Difference between the outside wall temperature and the surrounding temperature.
(v) The presence or absence of a cold air flow through the oven.

After some of the experiments in which the oven was heated in the conventional way, the baking compartment temperatures were also examined during cooling down, with opened or closed dampers in the combustion chamber door and in the chimney (item v). The results are shown in figures 3.22 and 3.23. Figure 3.22 shows the average baking compartment gas temperature for open or closed dampers during the cooling of the oven. The dashed curve, corresponding to the right y–axis, represents the temperature difference between the two cooling down curves.

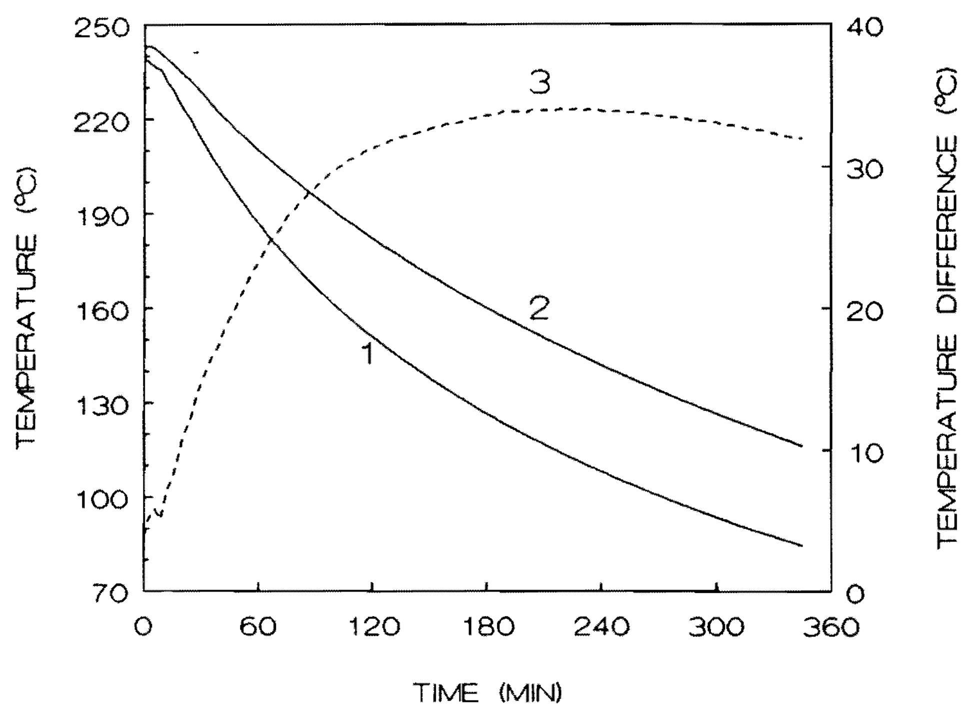

Figure 3.22: **Average baking compartment gas temperature during cooling down (left y–axis) with opened (1) or closed (2) dampers, and the temperature difference between these two (3, right y–axis).**

The figure clearly shows that cooling down with open dampers (curve 1) results in a faster temperature drop of the baking compartment gas. In the early part of the cooling down period, the baking compartment temperatures are almost the same (ΔT is about 5^0C). The temperature difference (curve 3) continues to increase until about 200 minutes of cooling down. Then the temperature difference remains constant for about one hour and starts to decrease after that. In the end the temperatures will of course both reach the surrounding temperature. This difference in temperature behaviour can be elucidated by means of figure 3.23, which shows the cooling rates corresponding to the temperature curves 1 and 2 in figure 3.22.

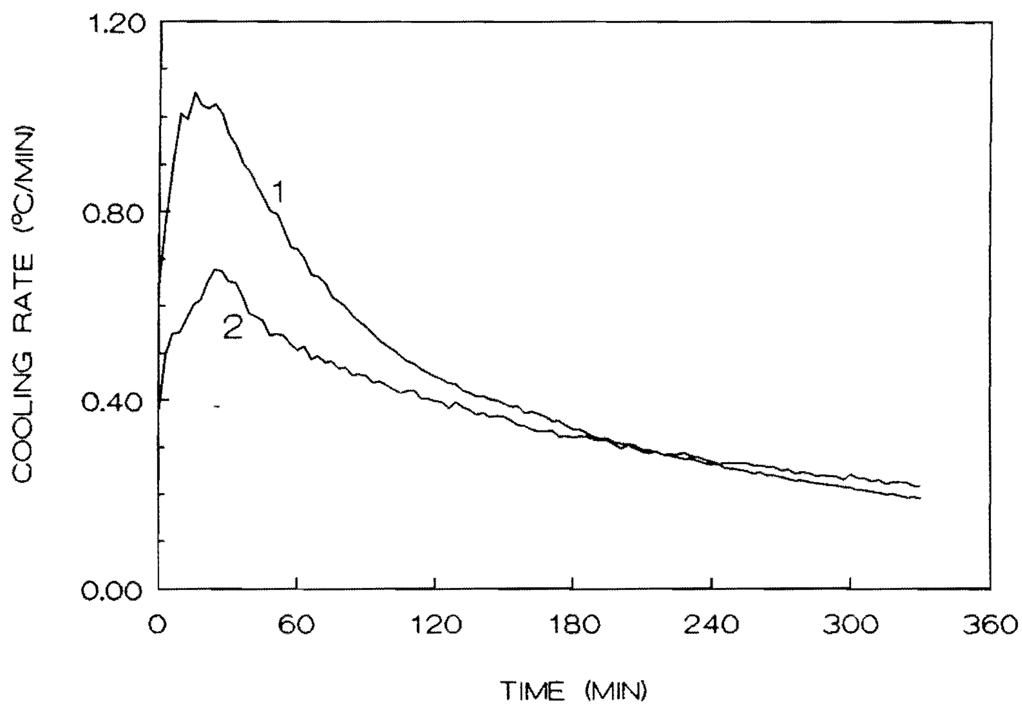

Figure 3.23: **Cooling rates of the baking compartment gas with open and closed dampers.**
1 : Dampers open; 2 : Dampers closed.

In case the dampers on the combustion chamber door and in the chimney are kept open during cooling down, the cooling rate increases to almost twice the cooling rate in case of closed dampers (resulted in an increasing temperature difference shown by curve 3 of figure 3.22). Due to the large temperature difference between the combustion chamber and the chimney, in the early part of the cooling down period, the natural draught will suck in cold air through the flues. This increases the cooling rate. Cooling down with closed dampers will prevent this sucking in of cold air despite the large temperature difference. Thus closing the dampers during the periods the oven is not used for baking, will save fuel in bringing back the oven tc its baking temperatures. How much fuel and time are saved is illustrated with the following example. Figure 3.22 showed that the temperature difference between opened or closed dampers after 5 hours of cooling down is about 33°C (curve 1: T_{bc}

= 93.5^{0}C; curve 2: T_{bc} = 126.5^{0}C). If we assume these temperatures to be the starting baking compartment temperatures for the next heating up period, figure 3.4 shows that for a 10 kW conventional fire (curve 4), the difference in heating time required to obtain a temperature of 240^{0}C is about 18 minutes. This means a difference in woodconsumption of about 640 g (15.5% of the total amount of wood required). In case of a 15 kW downdraft fire (curve 8 of figure 3.4), the difference in heating time is approximately 12 minutes. This represents a difference in woodconsumption of about 650 g (almost 20% of the total amount of wood used).

3.9 Conclusions

The comparison between the experimental results obtained during conventional burning and during downdraft burning showed large differences. In general we can state that for all circumstances examined, the performance of the oven during downdraft operation was much better than during conventional burning. Three important advantages of downdraft burning can be given.

(i) Higher possible power densities. During downdraft burning, power densities up to 50 W/cm^2 grate area were reached, while during conventional burning the maximum power density was just 26 W/cm^2. These higher power densities have resulted in largely increased heating rates and thus a strong reduction (about 1 hour) of the required heating time of the oven. Although higher power outputs were applied during downdraft burning, the reduction in heating time was sufficiently large to reduce the woodconsumption during pre–baking heating by about 17%, compared to conventional burning.

(ii) Cleaner combustion. The CO values of the flue gases measured at the chimney exit, were much lower than those found during the experiments with conventional burning mode. On the average, the CO/CO_2 ratios found during downdraft burning were smaller by about a factor of 10. Beside that, the less fluctuating composition of the flue gases indicate a much more regular combustion than during conventional burning.

(iii) Less soot deposit. The better quality of combustion during downdraft operation resulted in negligible soot deposit. Although only 30 hours of downdraft burning experiments were carried out, there is no reason to expect a very large increase in soot deposit after longer periods of downdraft burning. Therefore it becomes very likely that cleaning the flues once a month from soot and ash particles is sufficient.

4. Comparison between the experimental results and the mathematical models

4.1 Introduction

Both the experimental work done with the laboratory model of the oven (described in chapter 3) as well as the development of mathematical models of the oven (Schutte et al., 1988) form parts of the process in designing a full sized improved woodfired bakery oven. Because the laboratory model is a scaled down version of the prototype oven (the dimensions of the laboratory model were proportionally reduced by a factor of 2 with respect to those of the prototype oven), the experimental results obtained need to be scaled up to the dimensions of the full sized oven. A possibility of scaling up the experimental results to the dimensions of the prototype oven is by using a mathematical model earlier developed (Schutte et al., 1988, chapter 7). However, during the course of the experiments, some adjustments were made in this mathematical model. The nature of these modifications is discussed in section 4.2. Before scaling up the mathematical model to the dimensions of the prototype oven, it first needs to be compared with the experimental results obtained with the laboratory model, in order to examine its reliability (accuracy). This is discussed in section 4.3. Finally in section 4.4, the mathematical model is scaled up to the dimensions of the prototype oven. The influence of several parameters (power output, excess air factor and flue dimensions) on the heating time and corresponding fuel consumption of the prototype oven is predicted.

4.2 Adaptation of a Mathematical Model

In an earlier work, Schutte et al. (1988) provided a method to compute an expected temperature rise of the baking compartment gas during heating up of the oven under various circumstances (different power outputs, flame temperature, flue dimensions, etc.). Simultaneously it is possible to compute the corresponding temperature distribution over (a) the baking compartment floor and ceiling; and (b) of the combustion gases flowing through the flues. Figure 4.1 shows the temperature rise of the baking compartment gas during heating with a 7.5 kW, 10 kW and 12.5 kW conventional fire, according to the calculations with that model (the flame temperature and excess air factor were assumed to be 1000°C and 2, respectively, for all three power outputs).

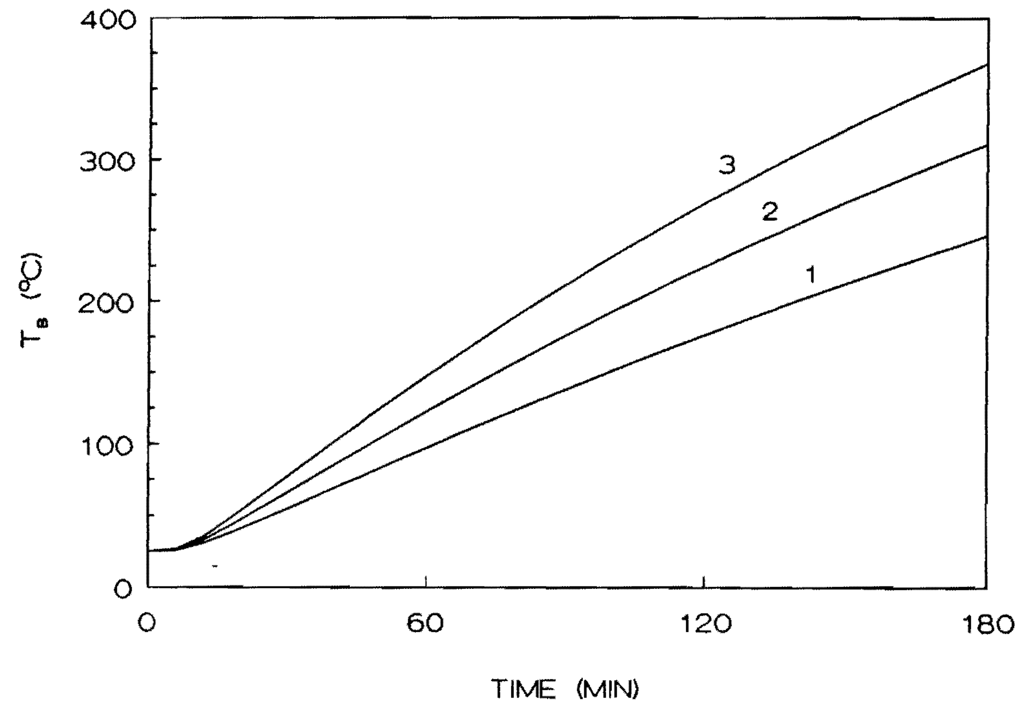

Figure 4.1: **Temperature time histories of the baking compartment gas in case of a 7.5 kW (1), 10 kW (2) and 12.5 kW (3) conventional fire. TUTSIM calculation model presented by Schutte et al., 1988 (T_f = 1000°C, λ = 2).**

At that time, the values for several parameters used in the model (i.e. flame temperature, excess air factor, the moisture content of the wood, etc.) were based on assumptions. However in practice these parameters are related to each other. For those calculations, being the first attempt in modelling a woodfired bread oven, this was acceptable. Because the results of these computations are to be compared with the experimental results, the mutual dependence of the parameters needs to be incorporated into the model.

Mutual dependence of the parameters

(i) The flame temperature depends on the amount of convective heat carried away from the fuelbed by the combustion gases, the excess air factor and the moisture content of the wood.

(ii) The required stoichiometric amount of air to burn 1 kg of wood is related to the moisture content of the wood.

The average flame temperature depending on the amount of convective heat carried away by the combustion gases from the fuelbed, the excess air factor and the moisture content of the wood can be calculated by drawing the following heat balance (Prasad et al., 1984; Bussmann, 1988).

$$M_vB_v + xM_cB_c = \frac{P}{B_f}\left[1 + \lambda\rho V_{st}\right] Cp \left[T_f - T_a\right] + q_r + q_{co} \tag{4.1}$$

Where:

$$M_v = 0.8\ M_f \tag{4.2}$$

$$B_v = \left[\frac{B_f - (1 - \nu)B_c}{\nu}\right] \tag{4.3}$$

$$M_c = 0.2\ M_f \tag{4.4}$$

$$B_f = \frac{[B_0 - (m + 9H)2575]}{(1 + m)} \tag{4.5}$$

The expressions for the excess air factor (λ), the stoichiometric amount of air (V_{st}), the density (ρ) and the specific heat capacity (Cp) were already given in section 3.2 (equations 3.3, 3.4, 3.7 and 3.8). The first term on the left hand side of equation 4.1 gives the heat generated by the combustion of volatiles, while the second term brings in the heat carried away from the fuelbed by the combustion gases. The first term on the right hand side denotes the sensible heat flux; the second term the radiative heat flux from the flames; and the last one the heat losses due to incomplete combustion (carbon monoxide production). The amount of sensible heat carried away from the fuelbed by the combustion gases, that contributes to the flame temperature depends among other things on the power output, the moisture content of the wood and the diameter of the fuelbed. The constant x is therefore a number between 0 and 1. Although Bussmann (1988) computed a value for x of 0.32, this value is only valid if the average fuelbed temperature and excess air factor for the fuelbed are taken as 1100K and 1, respectively. Recent experiments by Khan (1990) recorded flue gas temperatures of over 1100^{0}C. Thus higher fuelbed temperatures can be expected. Therefore it was decided to examine the two limiting cases.

(i) It is assumed that all the heat generated by the fuelbed is carried away by the combustion gases and contributes to the flame temperature ($x = 1$). Of course this implies that the computed T_f is overestimated.

(ii) None of the heat generated by the fuelbed is carried away by the combustion gases, but all is lost through radiation ($x = 0$). Thus T_f is underestimated.

The expectation is that the temperature rise of the baking compartment gas found during the experiments is located in between these two computed temperature time histories.

To solve equation 4.1 the following additional assumptions are made for reasons of simplicity.

(i) The combustion in the flames is complete. Thus q_{co} equals 0.

(ii) The radiative losses from the gases and soot are neglected, which implies that q_r is also 0.

(iii) No dissociation effects are taken into account. Dissociation requires a considerable amount of energy and reduces the flame temperature. Higher temperatures result in more dissociation. However the dissociation constants are rather small and will only become of some interest at temperatures much higher than those prevailing in a woodfire (Glassmann, 1977).

As stated before in section 3.6, the flame temperature is independent of the power output. Equations 4.1, 4.2 and 4.4 show the reason for this. Since q_r and q_{co} are assumed to be zero, each term on the left and right hand side of equation 4.1 can be expressed in terms of the woodconsumption M_f (equivalent to nominal power output), and therefore cancel each other.

For an arbitrary power output, 10% moisture in the wood, a Gross calorific value of 19900 kJ for 1kg of White Fir and a calorific value of 33000 kJ for 1 kg of fixed carbon, the influence of the excess air factor on the flame temperature is shown in figure 4.2.

Apart from a reduction in flame temperature with increasing excess air factor, the figure also shows that higher flame temperatures are obtained in case all the heat generated by the fuelbed is assumed to be carried away by the combustion gases. For smaller excess air factor, the differences in flame temperatures are larger.

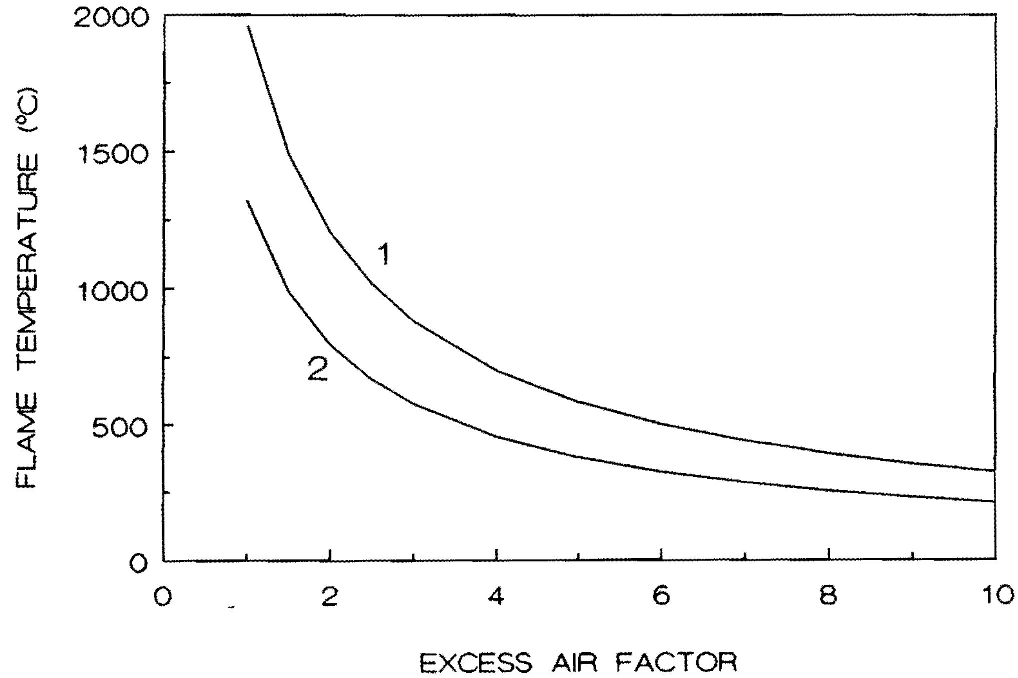

Figure 4.2: **Flame temperature versus the excess air factor. Curve 1: all heat generated by the fuelbed contributes to the flame temperature (x = 1). Curve 2: none of the heat generated by the fuelbed contributes to the flame temperature (x = 0).**

Next to be discussed is the influence of the moisture content of the wood on the required stoichiometric amount of air. The expression for the stoichiometric amount of air in m^3 to burn 1 kg of wood was already given in section 3.2 (equation 3.3). The influence of the moisture in the wood on the stoichiometric amount of air is as follows. 1 kg of wet wood contains less carbon, hydrogen and oxygen than 1 kg of dry wood. Thus less combustion air is required. Equations 4.6 and 4.7 can be used to compute the percentages carbon, hydrogen and oxygen in the wood, depending on the moisture content. The moisture content of the wood is computed on the basis of the dry wood weight.

$$m = \frac{M_{ww} - M_{dw}}{M_{dw}} 100\% \qquad (4.6)$$

Equation 4.6 can be expressed in terms of the dry wood weight depending on the moisture content.

$$M_{dw} = \frac{M_{ww}}{\frac{m}{100} + 1} \tag{4.7}$$

The individual constituents present in the dry wood must be multiplied with M_{dw}, calculated in equation 4.7, to obtain the percentages C, H and O in the wet wood. These are used to compute the required amount of stoichiometric air (see equation 3.3). Figure 4.3 shows the effect of the moisture content on the required amount of stoichiometric air to burn 1 kg of wet wood.

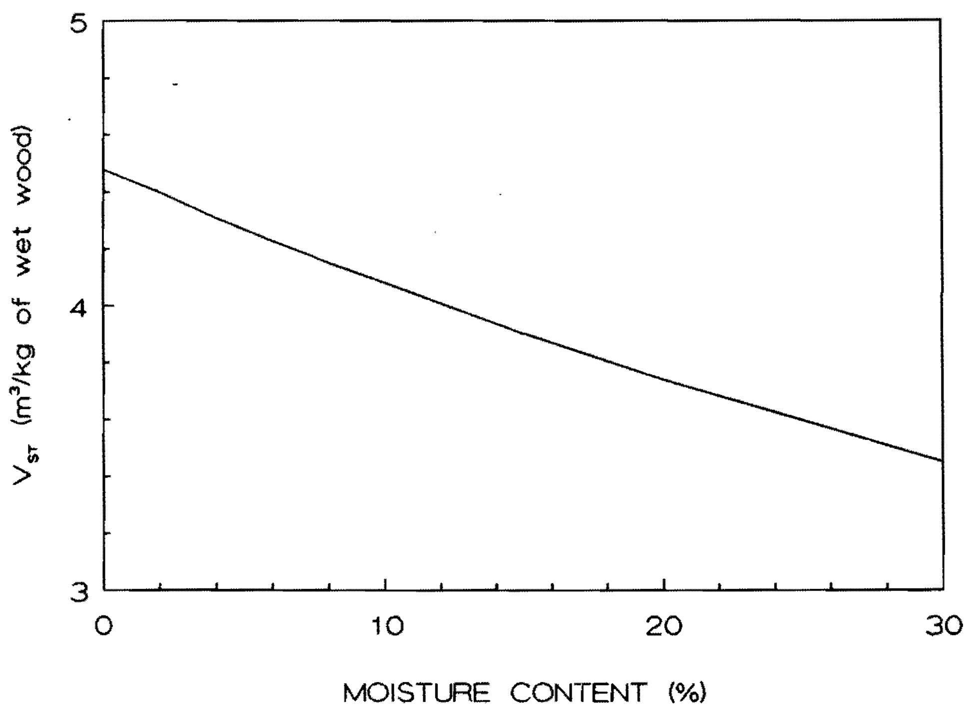

Figure 4.3: **Required amount of stoichiometric air (m^3) to burn 1 kg of wood as a function of the moisture content of the wood.**

4.3 Comparison between Computations and Test Results

Temperature rise and distribution in the laboratory model during heating up

The adjustments discussed in section 4.2 were introduced into the TUTSIM model. The results are shown in figures 4.4a, 4.4b, 4.4c and 4.4d. Curve 1 in each of the figures represents the temperature rise in case none of the heat generated by the fuelbed contributes to the flame temperature (x = 0). Curve 2 represents the temperature rise recorded during the experiments. Curve 3 represents the temperature rise in case all the heat generated by the fuelbed is assumed to be carried away by the combustion gases and contributes to the flame temperature (x = 1). The excess air factors and the initial baking compartment temperatures used for the computations are similar to those found during the experiments with corresponding power output. $T_{b,req}$ is the required baking temperature.

Except for the first 60 minutes in figure 4.4b, all temperatures recorded during the experiments with different power outputs, are in between the computed temperature curves. However, the position of the experimentally obtained temperature curves towards the computed temperature curves is different for each power output. Several factors can be responsible for this behaviour.

(i) The amount of convective heat carried away from the fuelbed by the combustion gases is different in all the experiments. Thus the actual flame temperatures were not the same. It is possible to estimate the average contribution of the convective heat to the flame temperatures in the

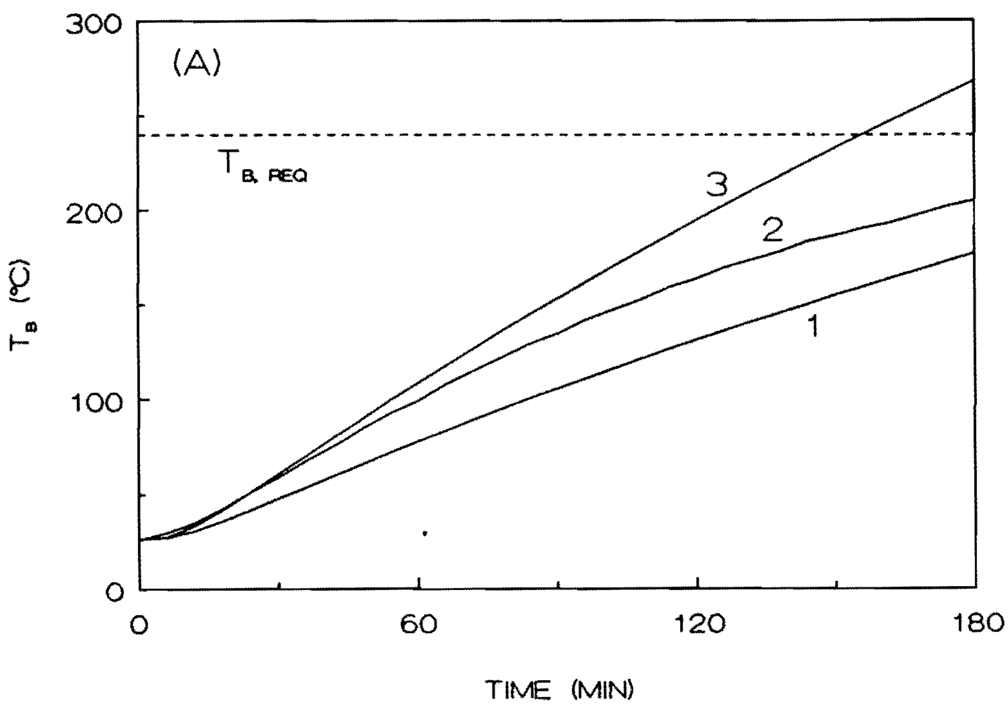

Figure 4.4a: Temperature rise of the baking compartment gas in the laboratory model for a 7.5 kW conventional fire. $\lambda = 3.54$.
1: $T_f = 503^oC$ (x = 0); 2: Experiment; 3: $T_f = 773^oC$ (x = 1).
T_b initial = 26.5^oC.

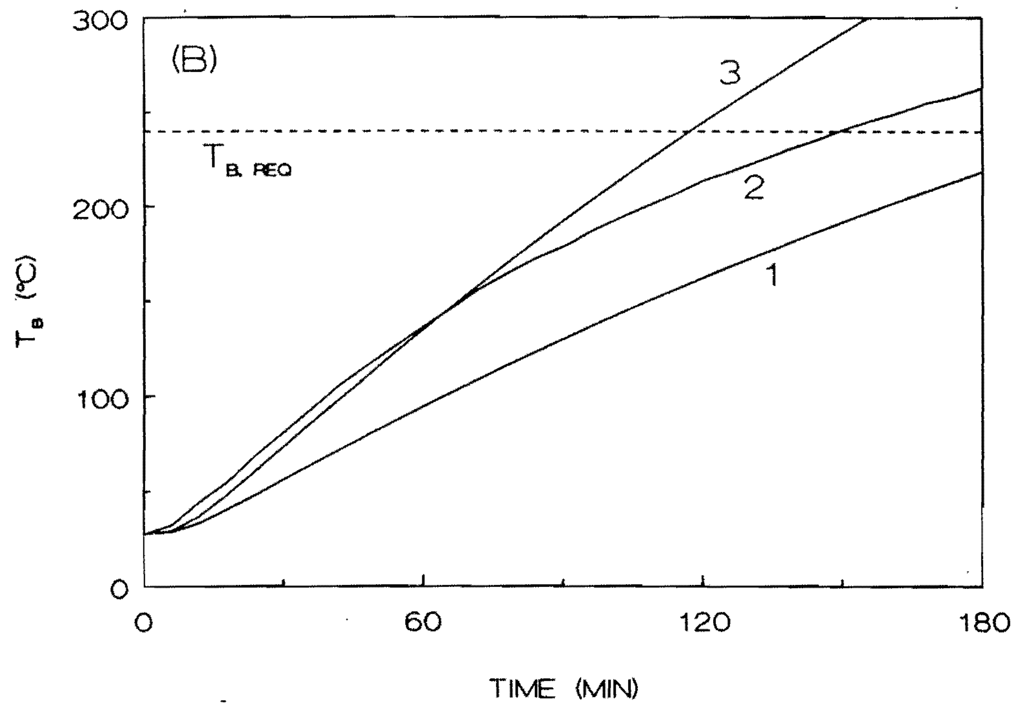

Figure 4.4b: Temperature rise of the baking compartment gas in the laboratory model for a 10 kW conventional fire. $\lambda = 3.14$.
1: $T_f = 555^0C$ ($x = 0$); 2: Experiment; 3: $T_f = 852^0C$ ($x = 1$).
T_b initial $= 27.6^0C$.

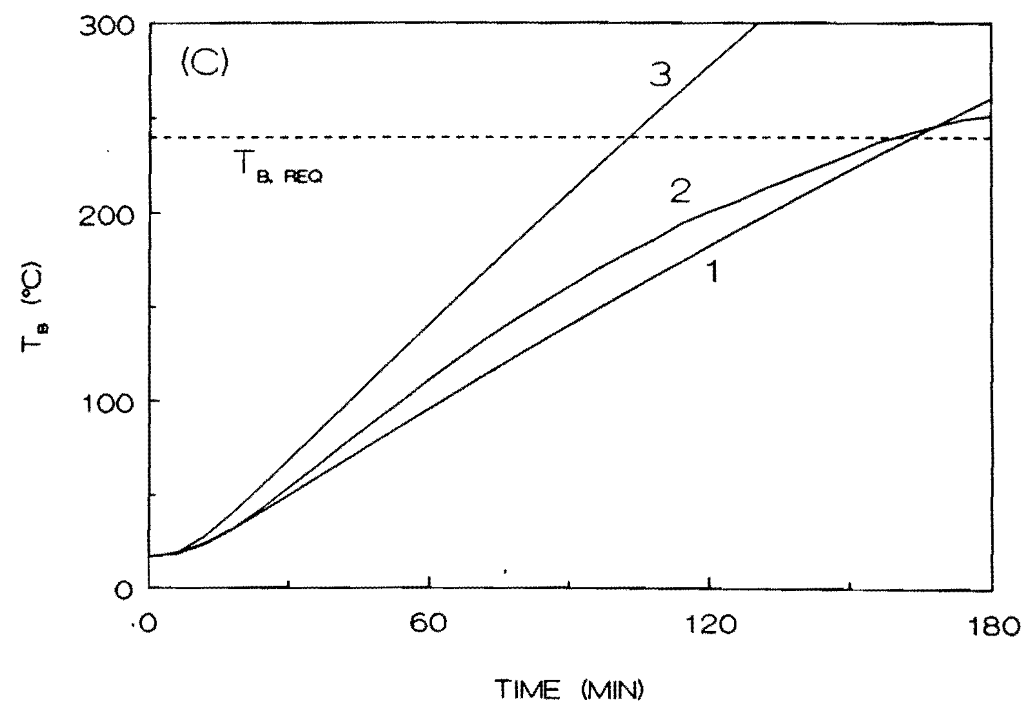

Figure 4.4c: Temperature rise of the baking compartment gas in the laboratory model for a 12.5 kW conventional fire. $\lambda = 1.57$.
1: $T_f = 959^0C$ ($x = 0$); 2: Experiment; 3: $T_f = 1443^0C$ ($x = 1$).
T_b initial $= 17.0^0C$.

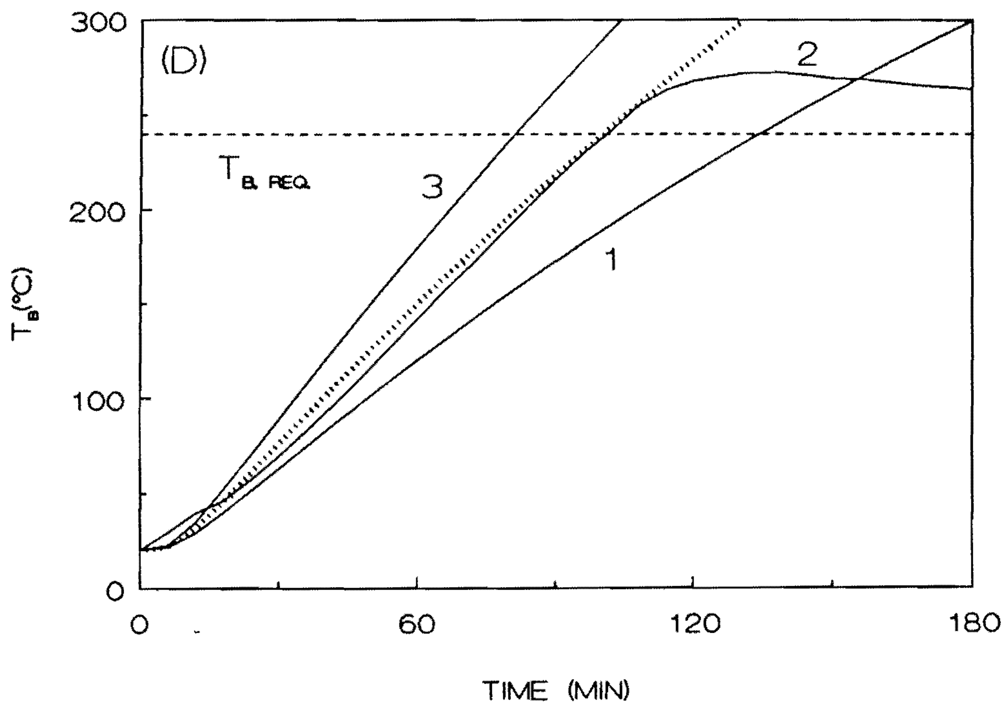

Figure 4.4d: **Temperature rise of the baking compartment gas in the laboratory model for a 15 kW downdraft fire. $\lambda = 2.28$.**
1: $T_f = 719^0C$ ($x = 0$); 2: Experiment; 3: $T_f = 1094^0C$ ($x = 1$).
T_b initial $= 20.3^0C$; ····· $T_f = 912^0C$ ($x = 0.5$).

experiments (x in equation 4.1). This is done by computing the ratio of the area in between curve 1 and 2 and the area in between curve 1 and 3. For the 7.5 kW, 10 kW and 12.5 kW conventional fire, x equals 0.52, 0.68 and 0.20, respectively. In the case of the 15 kW downdraft fire, x equals 0.48.

(ii) In the application of expression 4.1, we assumed clean combustion ($q_{co} = 0$). Especially the experiments with the conventional mode of burning showed large quantities of soot deposit and high CO/CO_2 ratios. The production of CO, according to equation 4.1, reduces the flame temperature. This is a possible reason for the experienced temperature rise in the baking compartment during firing with a 12.5 kW conventional fire. It only slightly deviates from the temperature rise in case of a 10 kW conventional fire, and it runs closely to curve 1 in figure 4.4c.

(iii) The TUTSIM programme used to compute the temperature curves 1 and 3 assumes in each case that the flow character in the flues is turbulent. In the actual experiments, the flow character might have been different (laminar or in the transition regime). Of course a different flow character means different heat transfer characteristics, which will change the positions of the computed curves with respect to the experimental curves.

In chapter 3 it was concluded that the oven should be operated in the downdraft mode of combustion to obtain the best results. Therefore it seems acceptable to state that in the case of downdraft combustion, the computer programme can be used to make a reasonable prediction of the actual temperature rise in the baking compartment and the corresponding heating up time of the oven, by computing a flame temperature that corresponds to $x = 0.5$. The result of such a computation is shown in figure 4.4d (curve $\cdots\cdots$).

Because some floor and ceiling temperatures were recorded during the experiments (see figure 3.1 for the precise locations of the thermocouples), some comparison between these temperatures and the computed average temperatures on the corresponding parts of the floor and ceiling is possible. It should however be clear that the computed temperatures represent the average temperatures for large floor or ceiling parts, while the recorded temperatures are only individual points. For comparison, the flame temperature used in the calculations is based on the assumption that 50% of the heat generated by the fuelbed is carried away by the flue gases and contributes to the flame temperature ($x = 0.5$). Under these circumstances it was found that the computed floor temperatures were about 15% lower and the computed ceiling temperatures 5% higher than the recorded floor and ceiling temperatures, respectively. This appeared to be independent of the applied power output or principle of combustion.

4.4 Temperature Rise and Heating Time of the Prototype Bread Oven

The ultimate purpose for developing the calculation model firstly presented by Schutte et al. (1988) and presently discussed, has always been to make reasonably accurate predictions of the required heating time and temperature behaviour in the prototype bread oven, under various circumstances. This would be a useful diagnostic tool to limit the number of experiments with the prototype oven, that concentrate on the thermal qualities (This was after all done in the laboratory model). Thus more attention can be given to the baking character of the prototype oven.

To apply the calculation model for the prototype oven at all, the results of the computations carried out for the laboratory model should at least reasonably correspond to the experimental results obtained with the laboratory model. This requirement is met, as was shown in the previous section. Till now, the results of the computations with the prototype can only be compared with the test and calculation results obtained from the laboratory model. Even when these results correspond nicely, this is no guarantee for the actual temperature behaviour of the prototype to react accordingly. This can only be found by testing a prototype.

For the calculations with the prototype oven, all dimensions of the laboratory model are doubled. To compare the results with those found with the laboratory model, the following assumptions are made.

(i) The heating up time for the prototype should be the same as recorded during the experiments with the laboratory model. The heating up time of the prototype oven can be represented by the following simplified equation.

$$t_{hp} = \frac{M_p \; Cp_p \; \Delta T_{hp}}{P_p} \tag{4.8}$$

where:

Cp_p = specific heat capacity of the construction material (kJ/kgK)
M_p = mass of the prototype oven (kg)
P_p = Power output (kW)
t_{hp} = heating time of the prototype oven (s)
ΔT_{hp} = difference between ambient temperature and baking temperature (K)

If we assume that $\Delta T_{hp} = \Delta T_{hm}$ (m stands for laboratory model); $M_p = 8M_m$ (an obvious consequence of linear scaling); and $P_p = 4P_m$ (assuming constant power output per unit area of the grate), the heating time of the prototype oven will be twice that of the laboratory model. Therefore to obtain the same heating time for the prototype oven, the power output in the prototype needs to be 8 times that of the laboratory model.

(ii) The excess air factors used for the calculation with the prototype oven are similar to those found during the laboratory experiments with different power outputs.

(iii) The initial baking compartment temperatures have also been chosen to be the same as for the laboratory experiments with different power outputs.

(iv) Because the flame temperature is independent of the power output (see equation 4.1), the temperatures that were used in the calculations with the laboratory model can also be used in the calculations with the prototype oven with proportionally the same power output.

Figure 4.5a shows the results of the computations (continuous lines) in case the circumstances for the 15 kW downdraft fire are scaled up to a 120 kW downdraft fire. Figure 4.5b shows the results for scaling up the circumstances of a 12.5 kW conventional fire to 100 kW. Only the results of 2 of the power outputs applied in the laboratory model and scaled up to the prototype dimensions are shown, since the behaviour recorded was almost similar for all the 4 cases. The figures show that the temperature curves for the prototype baking compartment gas (continuous lines) reasonably correspond to the curves of the laboratory model. Because of a larger delay in temperature increase during the first 12 minutes, the curves have somewhat moved to the right. This delay is probably due to the larger floor and ceiling masses, which take longer to raise in temperature. Only when there is a temperature difference between the floor/ceiling and the baking compartment atmosphere, heat can be transferred to the baking compartment.

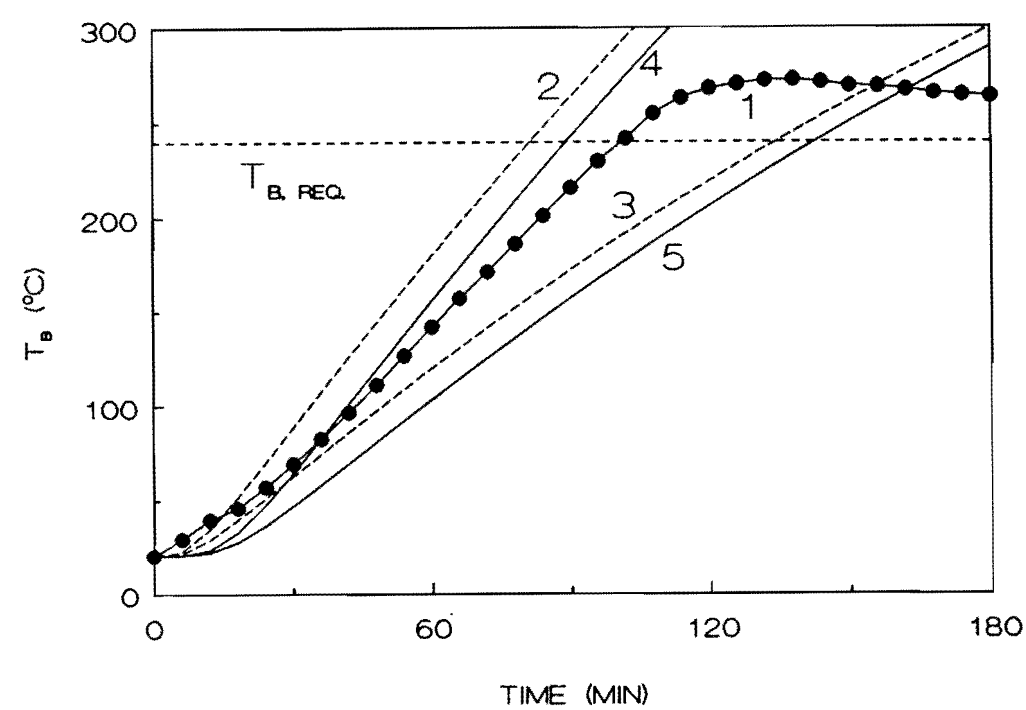

Figure 4.5a: **Temperature rise of the baking compartment gas in:**
Laboratory model: curves 1; 2 and 3 (15 kW downdraft fire)
Prototype oven: curves 4 and 5 (120 kW downdraft fire)
1: Experiment; 2 & 4: $T_f = 1094°C$ (x = 1); 3 & 5: $T_f = 719°C$ (x = 0), ($\lambda = 2.28$; T_b initial = 20.3°C)

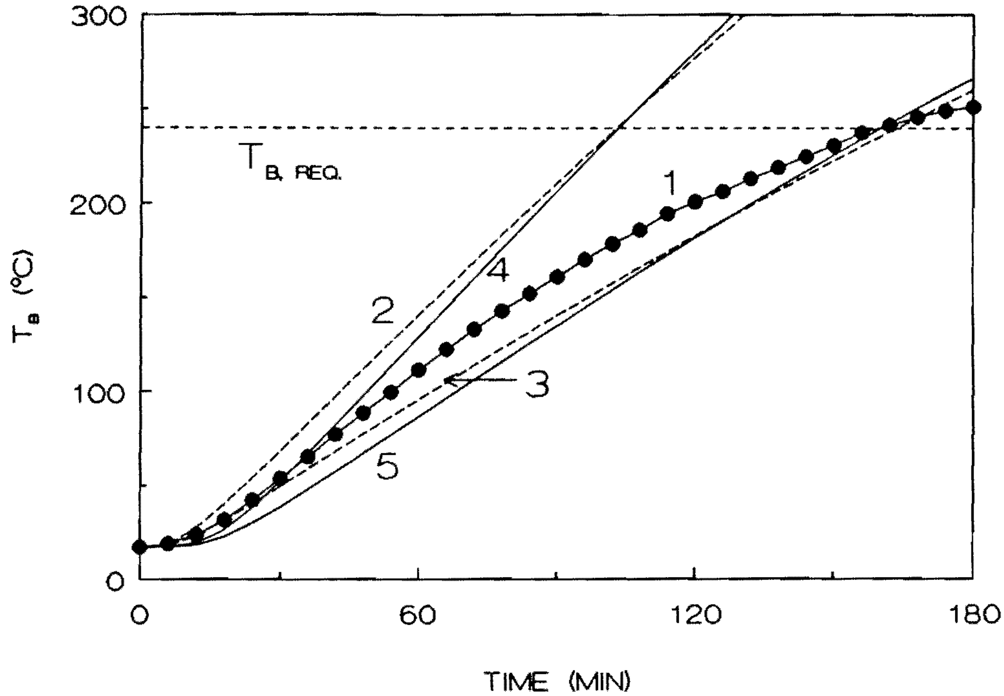

Figure 4.5b: **Temperature rise of the baking compartment gas in: Laboratory model: curves 1; 2 and 3 (12.5 kW conventional fire) Prototype oven: curves 4 and 5 (100 kW conventional fire) 1: Experiment; 2 & 4: $T_f = 1443^oC$ (x = 1); 3 & 5: $T_f = 959^oC$ (x = 0), ($\lambda = 1.57$; T_b initial = 17^oC)**

In the following tables the heating–up time of the prototype oven and the corresponding amount of fuel consumed, is predicted for various circumstances. In all cases, it is assumed that the oven is operated in downdraft mode. The flame temperatures used in the calculations are based on the assumption that 50% of the heat generated from the fuelbed is carried away by the flue gases and contributes to the flame temperatures (x = 0.5 in equation 4.1). The initial baking compartment temperature is assumed to be 20°C. The required baking temperature is set at 240°C. In table 4.1 the estimated heating times and amounts of wood

Power (kW)	*Heating time (min)*	*Fuel consumed (kg)*
80	150	42.8
100	125	44.5
120	108	46.2
140	97	48.4

Table 4.1: **Predicted heating times and fuel consumptions of the prototype oven for different power outputs. $T_f = 912^oC$; $\lambda = 2.28$.**

consumed are listed for various power outputs. Increasing the power output from 80 kW to 140 kW reduces the heating time by about 35%. Simultaneously the fuel consumption increases by 13%.

In table 4.2, the estimated heating times and fuel consumptions are listed for different excess air factors. Because the excess air factor affects the flame temperature, it was necessary to compute an individual flame temperature for each of the circumstances examined.

λ	Flame temp. (^{o}C)	Heating time (min)	Fuel consumed (kg)
1.5	1249	107	45.7
2.0	1008	107	45.7
2.5	848	110	47.0
3.0	733	113	48.3

Table 4.2: Predicted heating times and fuel consumptions of the prototype oven for different excess air factors. (P = 120 kW).

It is clear from the table that while the excess air factor has a large impact on the flame temperature, it only marginally influences the required heating time. This can be explained as follows. The decrease in flame temperature is proportionally about the same as the increase in volume flow due to an increase in excess air. The heat transfer from the flue gases to the flue walls not only depends on this temperature difference but also on the heat transfer coefficient, which in turn depends on the volume flow. Apparently these two quantities more or less balance each other. Thus the quantity of heat transferred per unit of time is about the same, which results in heating times that only marginally differ.

Table 4.3 shows the influence of a change in flue dimensions on the heating time and fuel consumption. Only the height of the flues was changed. Thus the total flue surface through which heat is transferred to the baking compartment remains the same. The reduction in heating time due to a reduction in flue height is not spectacular. Reducing the flue height to almost the dimensions of the laboratory model, only saves about 11 minutes of heating. The reason for this time saving is that due to the smaller flue dimensions, the flow velocities of the flue gases increase. This results in an increase of the heat transfer coefficient from the flue gases to the flue walls. However at the same time, due to these higher flow velocities, the residence time of the gases in the flues reduces. This has a negative impact on the total heat transfer.

Reduction in flue height(%)	Heating time (min)	Fuel consumed (kg)
0	108	46.2
10	105	44.9
20	102	43.6
30	99	42.3
40	97	41.5

Table 4.3: **Predicted heating times and fuel consumptions of the prototype oven for different flue dimensions (P = 120 kW; T_f = 912°C; λ = 2.28).**

The heating time computed for the prototype oven with 40 % reduced flue heights and a power output of 120 kW (table 4.3) is the same as for a prototype oven with 0% reduced flue heights and a power output of 140 kW (table 4.1). However, the difference in consumed amount of fuel is almost 7 kg, which is considerable. From an efficiency point of view it seems worthwhile to consider the possibilities for constructing the prototype oven with reduced flue dimensions (reduction in flue heights).

The results of the computations presented in the above tables form of course only a small part of the large number of parameters that can be examined with the calculation model. By listing the above results, the authors have tried to select and present the influence of those parameters that presently seem to be important for constructing the prototype bread oven.

5 Recommendations for the construction and operation of the prototype bread oven

5.1 Construction Aspects

The laboratory sized model of the oven, described in the present report, is based on a design provided by Mr. Beuker, a retired oven technician. The costs for the construction materials of the full sized oven, when built for example in Kenya, are estimated at about 1000 – 1100 US$. For the prototype oven, all linear dimensions of the laboratory model are in principle to be doubled. However, on the basis of the computer simulations and experimental results obtained, the following modifications are recommended for the construction of the prototype bread oven.

(i) The original design was fitted with a combustion chamber for application of the conventional principle of combustion. During the experiments with the laboratory model, the construction was changed for the downdraft burning experiments by placing a steel sheet in the combustion chamber, leaving only a small gap at the back (see figure 2.3). The grate area was kept the same. In the prototype bread oven, this separating wall should be constructed from refractory bricks to ensure a sufficient lifetime.

(ii) In the original design of mr. Beuker, the baking compartment floor above the bottom flues had a uniform thickness. During the first baking experiment (section 3.7), this resulted in too high floor temperatures above the central bottom flue. Therefore, between the baking compartment floor and the central bottom flue, an extra layer of bricks was installed that slanted towards the end of the central bottom flue. Thus providing a more uniform temperature distribution on the baking compartment floor. In scaling–up this layer of bricks and the floor part above it, they must be increased by a factor of 2, but the layer should still be slanting towards the end of the central bottom flue. However it was noticed during the experiments in which the laboratory model was operated in the downdraft mode of combustion, that the flames even entered the bottom front and side flues. Due to the presence of flames, the heating rates of these floor parts became too high. Since the prototype oven will also be operated in the downdraft combustion mode with a power output which is 8 times that of the laboratory model, the flames (which are a strong function of the power output) will certainly enter the bottom side flues, resulting in too high floor temperatures. It is therefore recommended to increase the thickness of the baking compartment floor above these flue parts by a factor of 3, instead of a factor of 2 (which would be the consequence of linear scaling). Thus the floor temperatures are expected to become more uniform.

(iii) In section 4.4, the computations for the prototype oven showed that due to a reduction in flue height, the heating time with a reduced power output remained approximately the same (the width and length of the flues were not changed in order to keep the total flue surface through which heat is transferred to the baking compartment the same). This resulted in a substantial reduction of the amount of wood required to heat the oven. However the flue heights can only be reduced to a certain extent. In principle, the cross sectional area of the flues should not become smaller than the cross sectional area of the chimney, to prevent that the flow resistances become too large, resulting in an insufficient chimney draught. In case the chimney dimensions of the laboratory model are doubled for the prototype oven, and if the condition mentioned above is met, the flue heights of the laboratory model need only to be increased by a factor of 1.4 to obtain the flue heights for the prototype oven (reduction of 30% compared to an increase of a factor 2).

(iv) It is not advisable to increase the wall thickness with a factor of 2 for the prototype (consequence of linear scaling). The reason for this is as follows. Intuitively speaking, it looks like a thicker wall will increase the amount of retained heat after a cooling period (insulation capacity), and therefore have a positive effect on the amount of fuel necessary to bring back the oven to its baking temperature. However, it is not advisable to enlarge the wall thickness beyond a certain limit, because a thicker wall (larger mass) requires more energy (fuel) to heat up and with that undo the positive effect of an enlarged insulation. Calculation by Schutte et al. (1988) showed that the amount of retained heat in the side walls after a cooling period, indeed increases with increasing wall thickness. However they also showed that the absolute amounts of heat lost to the surroundings after a cooling period of 14 hours (it was estimated that the oven would be used for 10 hours a day), is almost the same for a 10 cm and a 25 cm thick wall. Yet the amount of heat necessary to bring back the walls to their required temperatures will be smaller for a 10 cm thick wall than for a 25 cm thick wall. Still, because the heat input for the prototype oven will be 8 times that of the laboratory model, it would be unwise to maintain a wall thickness of 10 cm for the prototype oven (heat losses during heating up and baking become too large). Therefore it is recommended to construct walls for the prototype oven with a thickness of 15 cm.

(v) In the front wall of the prototype oven, lids are to be constructed at the height of the bottom and top side flues and at the height of the central top flue. Through these lids, the flues can be regularly cleaned (once or twice a month) from soot and ash particles. These lids should lock very tightly to prevent a loss of draught due to leakages at these places.

(vi) Although the chimneys of the laboratory model were made of metal, the chimney of the prototype oven can be constructed with bricks and have a square shape. The cross sectional area should be proportionally the same as for the laboratory model.

(vii) A damper should be placed in the chimney of the prototype oven. The construction principle was shown in figure 2.7.

(viii) As it was shown in figure 2.8, the chimney contains an air injection tube to bring about a forced chimney draught large enough to suck the combustion air through the fuel bed, necessary for downdraft combustion. During the experiments described in this report, pressurized air was used. This facility is unlikely to be present in small scale bakeries in developing countries. However, it is also possible to use a pair of bellows or a bicycle pump, which is connected to this air injection tube. This method will require more time before the temperatures in the chimney are high enough to cause a natural draught.

(ix) A steam provision is to be installed in the prototype oven. It can be constructed as follows. A small tube must be inserted through the back wall of the oven that ends in the baking compartment just under the baking compartment ceiling. At the other end, a small funnel, which can be filled with water is connected with a valve at the bottom. By opening of the valve, water can flow into the baking compartment where it will evaporate.

5.2 Operational Aspects

For proper operation of the prototype oven, the following aspects should be taken into account.

(i) The prototype oven must be operated in the downdraft mode of combustion. The combustion process is started by building a small conventional fire on the grate. This probably will be easier if the secondary air by–pass is opened. In this way, air can be supplied from under the grate for the combustion of charcoal. When this fire has reached the steady state after about 10 minutes, the secondary air by–pass must be closed. At the same time, the pair of bellows or the bicycle pump at the air injection tube must be used to bring about a forced chimney draught, that forces the combustion air to go downwards through the fuelbed. This draught can be supported by blowing with a piece of hardboard at the combustion chamber entrance. Shortly after that downdraft combustion will start. From that time on, the wood charges can be gradually increased until the desired power output is reached.

(ii) The power output that should be applied in the prototype oven using downdraft combustion is 120 kW. With flue heights that are 1.4 times the height of the flues in the laboratory model, the heating time to reach a baking compartment gas temperature of 240°C is approximately 100 minutes.

(iii) Although the dough pieces of 550 g each are to be baked at a temperature of about 240°C, the heating should continue until a baking compartment gas temperature of 260 – 270°C is reached. This will avoid that the temperature in the baking compartment drops below 240°C at the time the dough pieces are put into the oven for baking (a larger accumulation mass by increasing the thicknesses of the floor and ceiling is not advisable, since it will extend the heating up time). Then the oven should be left to rest for some time (15 – 20 minutes), applying a very low power output, in order to increase the uniformity of the floor and ceiling temperatures. At the time of baking, the power output must be increased strongly to prevent a too large temperature drop.

(iv) The dough pieces are to be baked on metal plates or in baking tins. This will provide more uniform temperatures at the dough surfaces that are in contact with the metal. The plates or baking tins must be greased properly to avoid any problems in removing the bread pieces from the metal sheets/baking tins.

(v) Several times during baking, water should be injected into the baking compartment. The water immediately evaporates and partially condenses on the relatively cold dough surfaces. This prevents the dough pieces from drying out, causing a premature rupture of the crust. It will also have a positive influence on the browning of the crust, since water is an essential compound in the browning reaction.

(vi) The use of the chimney damper.

(a) During the so-called "simmering" periods, the power level at which the oven is operated is strongly reduced. By partially closing the damper, the combustion rate is expected to reduce due to the reduction of the flow velocities. Simultaneously, it will also reduce the amount of excessive air that is sucked in through the combustion chamber. Thus the sensible heat losses will reduce. However in these situations, the damper should be used with some caution since partially closing it can deteriorate the combustion quality.

(b) During cooling of the oven, the chimney damper must be kept closed to prevent that cold air is sucked into the flue system, increasing the cooling rate.

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