Technical aspects of woodburning cookstoves

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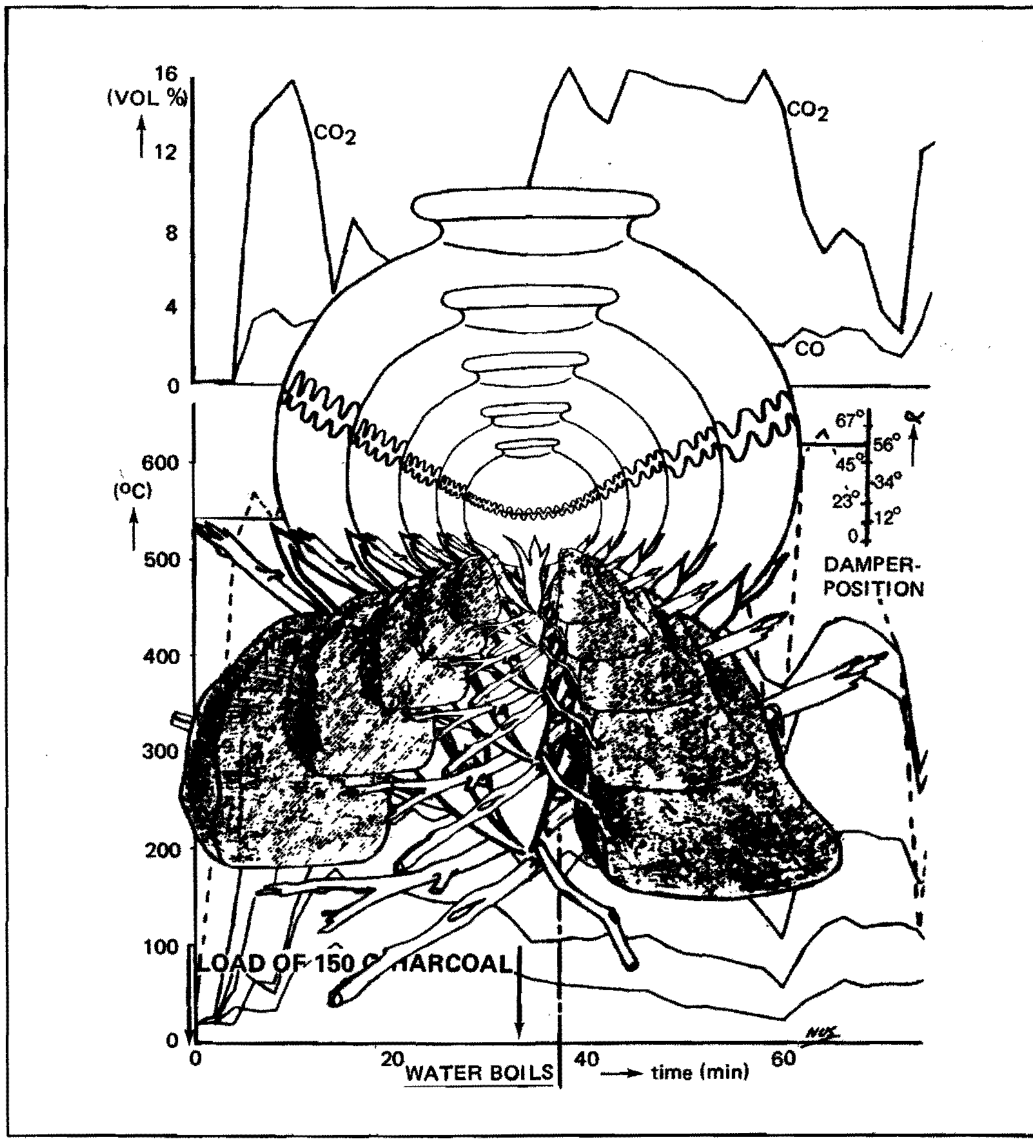

A Report from
The Woodburning Stove Group
Departments of Applied Physics and Mechanical Engineering,
Eindhoven University of Technology
and
Division of Technology for Society, TNO, Apeldoorn
The Netherlands

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PREFACE

This is a fourth report of the laboratory investigations on wood stoves at Apeldoorn, Eindhoven and Leuven. The work described in the report was carried out for most parts during the calendar year 1982. The work has been grouped under three parts.

Part A: Problem areas
Part B: Stove designs
Part C: Fuels and performance

A significant fact that emerges from the report is that wood stoves can be designed, constructed, and operated to yield efficiencies between 40 and 50%. Two crucial aspects require a closer look: that of realizing acceptable turn-down ratios (they are not quite good yet); and the carbon monoxide production containment which is a serious health hazard.

Editors
Eindhoven

Part A: Basic theoretical and experimental studies

1. OBSERVATIONS ON COMBUSTION AND HEAT TRANSFER

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1.1 Introduction

Theoretical analysis (De Lepeleire, 1981) has shown that at high firebed temperatures the direct radiation induces heat flux densities at the pan bottom which are well over what is commonly achieved by convection.
With moderate firebed temperatures however this radiation heat transfer drops dramatically due to the well known fourth power.

On the other hand, whereas the expected convection heat flux densities are limited in an open fire or with a low speed ducted gas flow, they can be fairly increased through higher speed forced flow along the pan bottom and walls. Even, with moderate firebed temperatures as expected in practice, and special emphasis on convection, it seems possible to get convection flux densities higher than those for radiation (De Lepeleire, 1982). In theory the convective heat transfer can be manipulated a lot by suitable geometry of the heat transfer zone, far more so than radiation.

These theoretical conclusions however so far have not been checked by experiments. So the intention arose to do so, and to compare convection and radiation densities.

In a first phase this will be done with a shielded fire without a chimney (type GDL1). From theory it is expected that the feasible power and flux density will be rather low due to the very limited draft available in a stove without a chimney. Afterwards similar tests are programmed on a stove with a chimney...

Preliminary results have been recorded on radiation and convection. However combustion problems interfered a lot and had to be looked into as well, as geometries for emphasis on convection induced a rather critical air supply situation in the chimneyless stove tested. Therefore the grate had to be adapted to some extent, and this point necessarily was part of the investigation.

Hereafter limited results and comments are presented on combustion and heat transfer.

1.2 Test conditions and general data

Fuelwood:

- meranti
- size : 20x20x70 mm
- dried at 110°C for 25 hours
- measured heating value :

  wood Ho ≃ 19,5 kJ/gram

  charcoal ≃ 30 kJ/gram

- fir
- size : 20x20x70 mm
- dried at 110°C for 24 hours
- measured heating values :

  wood Ho ≃ 20,3 kJ/gram

  charcoal Ho ≃ 31,5 kJ/gram

  unit charges : about 100 g added as soon as flames from former charge disappeared.

From the above : ratio $\frac{\text{Ho (charcoal)}}{\text{Ho (wood)}} \simeq 1{,}5$

Only high power tests were done very much like a short boiling test (SBT). In fact the test procedure itself was tested at the same time.*

---

* The fire was started with a gas burner, which is very convenient in a lab. For the field test some kerosene might be the best way out. (cfr. Field test draft 3, Arlington.)

Together with weighings and temperature measurements gas analysis was included (CO_2, CO, O_2). Puzzling results came out showing a very high concentration up to 29% ($CO_2 + O_2$). This appeared to be due to a leakage in the pump of the O_2 analyser (suction side). Therefore the O_2 indications were wrong. After repair the figures switched to "normal" levels.

1.3 Observations on combustion

Test series 1

A conical grid in 1 mm steelsheet was used, as shown in figure 1.1. In the central area, 38 primary air holes Ø 10 mm were drilled, at the periphery 36 holes Ø 10 mm for secondary air. The air flow was - to some extent - controlled through obstruction with sand of the triangular air intake Ao with 5280 mm^2 max. cross section.

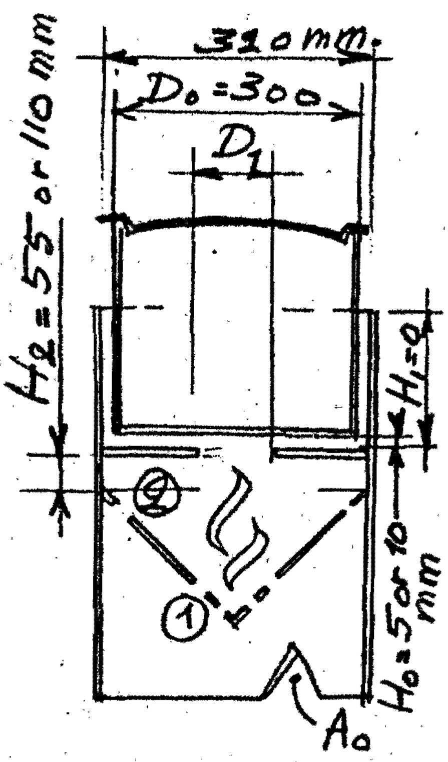


Figure 1.1: Test stove, with radiation shield under pan bottom

1 primary air holes (38 Ø 10)
2 secondary air holes (36 Ø 10)
D_1 = 50, 100, 150, 200, 250 or 320 mm.

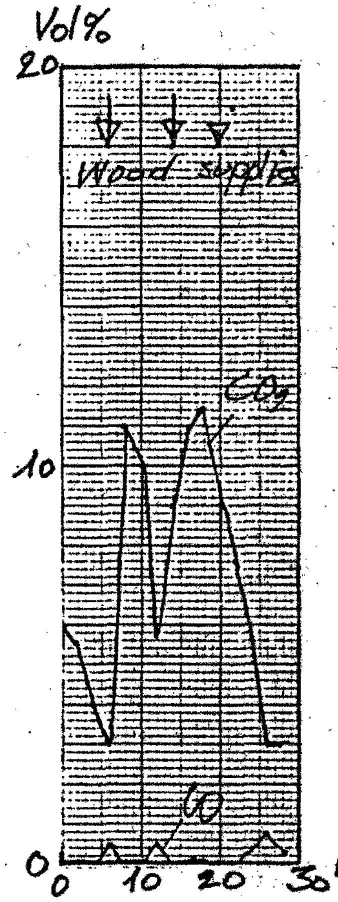


Figure 1.2: Typical gas analysis; grate with secondary air holes; D_1 = 250 mm

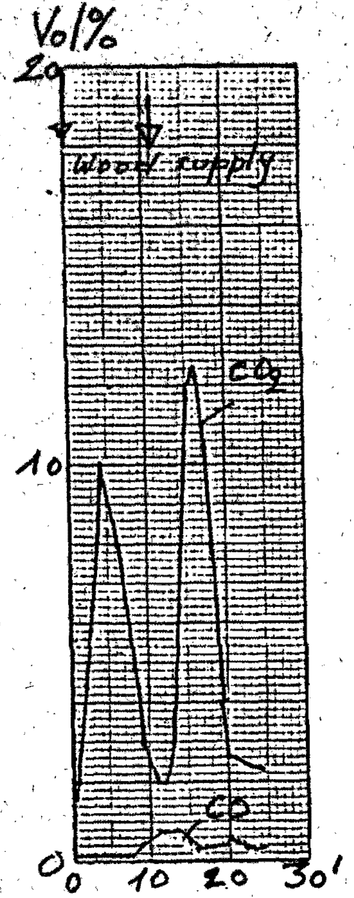
When starting a wood fire, stable volatiles flames lodged on the secondary air holes. A fair combustion was achieved with low CO concentration (< 0,5). The CO_2 concentration went up after adding wood (up to 10..12%). (figure 1.2)
However after some time a charcoal bed built up, and a serious drop in the stove power and efficiency was observed. The power (rate of combustion) went down, probably due to the fact that with the given stove geometry and fire structure there was a poor air distribution. With too low a power level the heat losses from the stove and pan systems to the surroundings are relatively higher when compared with the net heat input into the pan. Even there is some power level where the pan temperature is just maintained, without any further increase: at this level the efficiency in fact is zero.

Obviously cumulative obstruction of the primary air holes and bypassing of the air through the secondary air holes resulted in a shortage of combustion air in the charcoal bed.

Test series 2

To prevent charcoal build up, a grate without secondary air holes was tried. It was expected that in this case any of the limited air supply available would go through the central grate area and thus through the charcoal bed. The charcoal did completely burn indeed. The CO_2 and CO concentration peaks increased (figure 1.3). A lot of smoke was generated, but the power and efficiency were not seriously affected except with small D_1 diameters. In this case CO_2 went up to 18%, CO up to 3% (figure 1.4). Obviously the charcoal combustion was "paid for" with a poor combustion of the volatiles. From the high CO_2 content one could conclude that the stove was overloaded.

On one hand the high CO content indicates poor combustion, but on the other hand the very high CO_2 concentration suggests that this is due not to poor mixing or too low combustion temperatures, but rather to shortage of air input (air excess factor about 1.2). In other therms: an excess of wood...



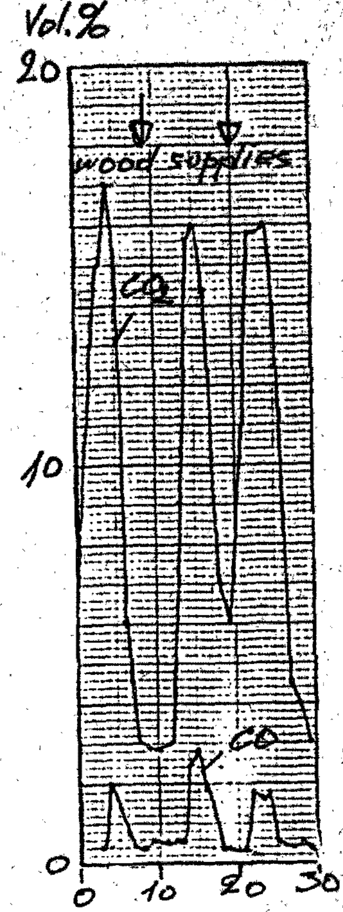


Figure 1.3: Typical gas analysis; grate without secondary air holes; D_1 = 250 mm

Figure 1.4: Typical gas analysis; grate without secondary air holes; D_1 = 50 mm

Tests at lower power level or with a different grate might give the answer.

Anyhow, it seems that in this stove, at the power levels used, (3 ..4 kW), on one hand fair combustion of volatiles requires ample secondary air holes whereas charcoal burnout requires to obstruct them. The air control damper upstream from the grate (Ao) has a poor effect on the process. The question arises if active control of the primary to secondary air ratio is necessary.*

* Former tests in Eindhoven by P. Visser with a very similar stove (but with a flat grate) reportedly have shown excellent overall efficiencies together with ugly combustion (smoke and/or charcoal build up).

1.4 Air supply

Beyond the air supply in sufficient quantities there is another problem about mixing of secondary air and volatiles. In a low draft system turbulency can hardly be expected. It has been observed that laminar flows in parallel direction do not mix easily, as diffusion is the main transport phenomenon involved, which is rather slow. However, cross flow arrangements perform visibly better, which is not astonishing at all. Cross flow occurs for example where volatiles circulate along a sheet, while combustion air comes in at a right angle, through holes in the said sheet. Of course the higher the available draft the higher the combustion air speed and throw, and the better the mixing will be.

Here a conflict arises.
Other studies (De Lepeleire, 1982) have shown that for good convection heat transfer pressure drops should concentrate in the heat transfer area, not in the chimney or in air dampers; here it seems that some draft is to be consumed in the combustion zone. In the end we have to look for a fair compromise, and spend some of the limited draft available in the combustion zone, and some in the heat transfer area. In any case it seems that pressure drops outside the combustion or heat transfer zones (for example in separate air dampers upstream) should be avoided or kept at a minimum.

These are tentative conclusions to be checked by further experiments.
A next step therefore will be to try a grate with built in and separate flow controls for primary and secondary air. Let's stress the "built in" idea: in this way the available draft is converted into kinetic energy in the firebed, and available for mixing combustion air and volatiles. Another philosophy behind it has to do with the stove flexibility. It is well known that in general a stove has a limited power range (P_{min}-P_{max}).

Power can be written:

$$P = (P/A) \cdot A \text{ kW}$$

where (P/A) is a specific power (kW/m^2). A is the grate area. Now, the power range can be influenced through (P/A) (which is pretty difficult) or through (A). The "built in" concept will include a variation of the active grate area..

1.5 Observations on heat transfer

Similar stoves have been tested, which were different only in the diameter D_1 in a horizontal shield parallel with the pan bottom at a distance Ho (5 or 10 mm) (figure 1.1). In fact if D_1 is equal to D_o for example, the firebed radiates directly to the pan bottom, and probably takes an important share in the heat transfer. However, when D_1 is small as compared with D_o, (for example 1/3) the pan bottom area exposed to direct radiation from the firebed is reduced to a bit more than 1/9; the remainder now receives indirect radiation from the shield, and increased convection inputs. The convection input is increased due to the higher gas velocities as imposed by the small gap width Ho.
The question was to see how the heat transfer efficiency changes as a function of (D_1, Ho), or in other terms whether radiation or convection heat transfer processes are the important ones. Test results are summarised in table 1.1.

The stove was operated as explained above: about 100 g of dry wood (size 20x20x70 mm) was added when the flames of the former charge disappeared.
Some tests were done once, others twice, whereas eleven tests with (D_1 = 250, H_2 = 55 mm) are done. These eleven tests range in power from 2.24 kW to 3.87 kW. In fact the disappearance of volatiles flames is an unstable criterion for adding wood. Note that when short, square wood is added (in fact it has to be thrown in) many different firebed structures may result. Probably this explains why the power varied so much.

	H_2 = 5.5			H_2 = 11			H_2 = 5.5		
	H_o = 1.0			H_o = 0.5			H_o = 0.5		
D_1	test nrs	eff. (average)	power kW	test nrs	eff. (average)	power kW	test nrs	eff. (average)	power kW
5	2 32 44	.274	3.15	1 14	.317	2.38	-	-	-
10	-	-	-	3	.382	3.27	15 25	.42	2.58
15	37 39 40	.3913	3.4157	4	.396	3.26	16	.40	4.50
20	35 38	.395	3.40	6 11	.40	3.97	17	.408	3.17
25	34	.417	3.11	5 9 10 47	.3994	3.8475	18 19 23 24 26 27 33 48 49 50 51	.4219 max. min.	3.17 .508/2.24 .35/3.20
32				7	.4246	4.66	20 22 41	.445	3.6

Table 1.1: Overview of the tests performed.

On the other hand the power was relatively low.
As a conclusion, it seems that the procedure used to control the fire power level, mainly adding wood when flames dissapears, is not the best one. Looking at the efficiency: there seems to be a tendency for efficiency decrease at higher power. However, the power range and the number of tests looks too small for final conclusions.

Let's compare now the complete set of results, and look for the influence of the diameter D_1 of the hole in the radiation shield under the pan bottom. The influence of D_1 is rather small in the range 100-250 mm, when the efficiency is considered, and a little more visible for the power level. This is not astonishing, as certainly the stove with emphasis on convection has higher flow resistance and a similar draft..
On the other hand it appears that shielding the pan bottom with D_1 = 100 mm does not destruct the heat transfer efficiency. In other terms: a drastic reduction of direct radiation is compensated by convection heat transfer and indirect radiation. However: when D_1 is further decreased to D_1 = 50 mm both the power and the efficiency suffer a lot. When no radiation convection shield is used at all (D_1 = 32 cm), the fire power increases (as expected) and so does slightly the efficiency.

From the foregoing the following first conclusions are suggested:
- A radiation-convection shield is not recommended for a shielded stove without a chimney, as it does not increase the efficiency, whereas the possible power level is reduced.
- However: the forced flow convection heat transfer was strong enough to maintain the efficiency. It might be useful in stoves with a chimney where more draft is available, and where more emphasis can be put on convection by narrowing the gap width (Ho). This will be investigated later.

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G. De Lepeleire, 1982
Some stove design rules as derived from a convection stove model.
K.U. Leuven.

2 NOTES ON GAS ANALYSIS

by
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2.1 Introduction

One of the basic requirements for drawing mass and energy balances in a wood stove is the quantity of air flowing through the stove. Direct measurement of air flow through a small naturally aspirated device like a wood stove is not an easy experimental task. Therefore the group has relied on the carbon balance technique to infer the air flow through the stove using measurements of gas composition, fuel quantity and the ultimate analysis of the fuel. The method while being reasonable did produce off and on irreconcilable results between the oxygen percentages inferred from the carbon balance technique and the oxygen meter records. This was traced to a calibration problem of the oxygen meter. A second confusing factor is the oxygen present in the wood, which also participates in the combustion process along with the oxygen in the air.

In view of these difficulties, it was decided during the course of the work to obtain an independent check on the gas analysis and results obtained therefrom. The obvious candidate for this was to measure nitrogen, since it is present in air only, always at the same concentration and passes through the stove without any chemical interaction.

The most accessible way of measuring nitrogen is by means of a gas/chromatograph which is discussed in section 2.2 (a test stand for measuring nitrogen in fluegases of woodfires).

In section 2.3 of this chapter the Ostwald diagram will be discussed. The diagram provides a quick check on the excess air flowing through a stove by just knowing the CO and CO_2 content in the combustion products. For work on stove design development we expect that this would be a very useful tool which avoids the laborious process adopted by the Woodburning Stove Group. Wood contains carbon, hydrogen and oxygen. These constituents vary with the different wood species and therefore will affect the combustion process. The difference can be easily depicted in the Ostwald diagram.

2.2 A test stand for measuring nitrogen in flue gases of woodfires

The measuring circuit to determine nitrogen includes a gas chromatograph, an integrator, a stripchart recorder and further necessary accessories (see figure 2.1a). This setup is preceded by a gas handling train to clean the flue gases (see figure 2.1b). For the apparatus used see the detailed instrumentation list in appendix A.

The principle of gas chromatography

The chromatographic separation of different gases on a solid adsorbent the so-called Gas Solid Chromatography (GSC) is a result of different adsorption coefficients of the gases interacting with the solid phase. These different adsorption coefficients led to different retention volumes (this is the volume in the column required for elution of the sample. When a mixture of gases is lead through a column at a constant flow, different retention times are attained. In this case the column is streamed with helium at a constant flow, this is the so-called carrier gas. By means of a sampling valve a plug of purified flue gas is inserted in the carrier gas flow and is lead through the column. In the column, the plug is unraveled into separate components which leave the column one after the other and are led through the katharometer. As these components have different c.q. lower thermal conductivities compared to that of the helium they cause a temperature rise at the filaments of the katharometer. This temperature difference results in a change in resistance of the metal wire and is measured by means of a Wheatstone bridge. The katharometer is linked to a stripchart recorder and an integrator.

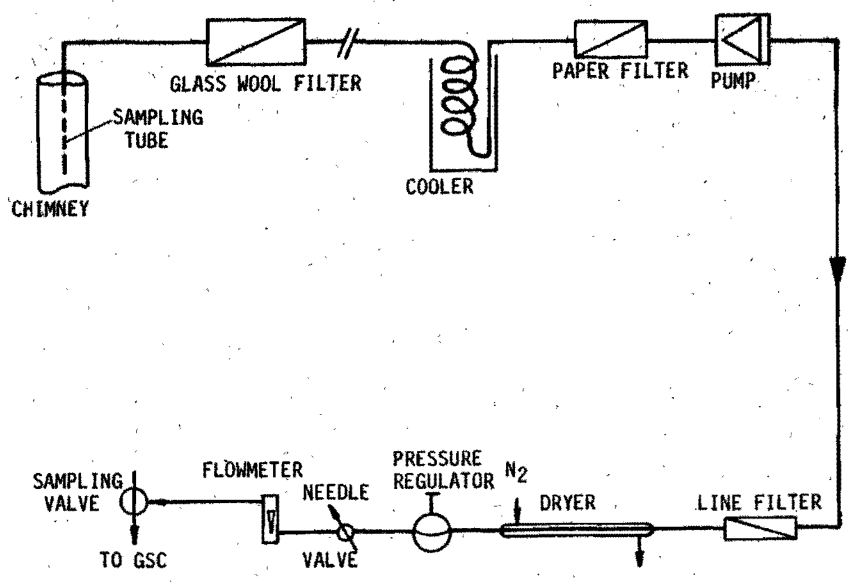

Fig. 2.1b Gas handling train to clean the flue gas.

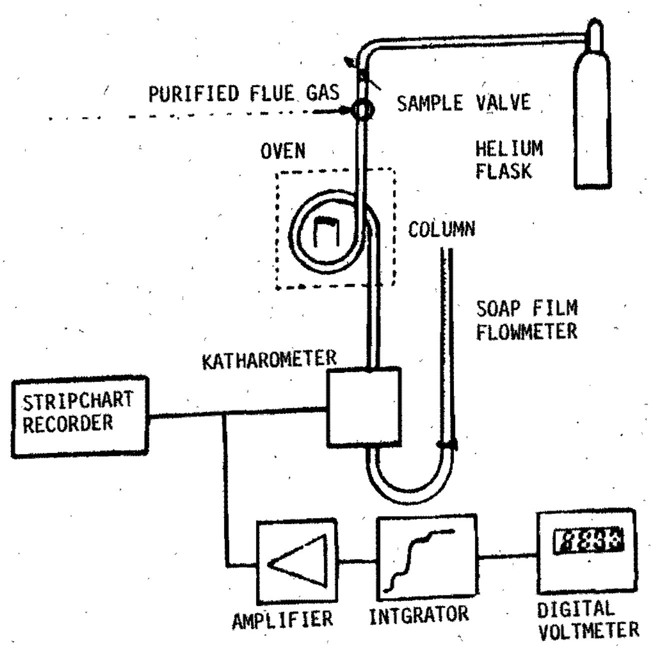

Fig. 2.1a Measuring circuit to determine nitrogen.

Since the signal is too small for the integrator to handle, it is first amplified. The column is of the molsieve type and is capable of determining nitrogen, oxygen, carbon monoxide and methane. Carbon dioxide is too big a molecule and therefore it cannot pass the column. Although we are interested mainly in the nitrogen concentration also oxygen and carbon monoxide are quantitatively measured. And at certain stages of the experiment c.q. just after a new charge, methane is qualitatively determined. Carbon monoxide because of its low concentration compared to those of oxygen and nitrogen showed a great deviation even at the calibrating stage and is therefore not taken into account.

Measurements

At a carrier gas flow of 22 ml helium per minute it takes 4 minutes for a sample to pass the column with oxygen having the smallest retention time of 97 seconds and carbon monoxide having the longest of 189 seconds. Taking the running start and the run out of the components into account, the time elapsed between the first and the last peak is about two minutes. Because all the operations had to be done by hand and to have a safe margin in which a sample can be analyzed the time interval of three minutes was derived. This means that when a sample is still leaving the column and entering the katharometer a following sample is already injected into the column. Before and after an experiment of about two hours which includes 40 measurements, the chromatograph is calibrated with air, pure nitrogen and a calibration gas. The latter contains 0,5% CO, 6% CO_2, 7,5% O_1 and 86% N_2. The peak areas are measured with an integrator and the output is displayed on a voltmeter. From the calibration gases and their corresponding peak areas the conversion factors can be calculated. With these factors the peak areas of the measured samples can be converted into concentrations.

Flue gas handling

Before a fluegas sample can be led into the column of the gas-chromatograph it has to be freed of all its solid particles and water vapour. Therefore a gas handling train was constructed to accomplish this (see figure 2.1b). It consists of the following components. A tube filled with glass wool at the sample inlet filters away the course soot, tar and ash particles, then the gas is led through an icecooler where the main part of the water vapour is condensed. Further downstream there are a paper filter, a pump and a line filter (boro silicate) with a retention efficiency of 99,95% for particles of 0,6 micron and even a higher efficiency for particles above and beyond this size. Then the gas flows past a dryer. This is a thin walled tube of 1,2 m long with a diameter of about 3 mm that adsorbs and permeates water only. This tube is sealed in an impermeable shell through which a dry gas (nitrogen) flows counter current to the sample flow. At the end a precision pressure regulator provides a constant sample flow of $100^{ml}/min$ to the gas chromatograph.

Experiments

During tests with the Metal Experimental Stove the flue gases were measured for their nitrogen content. For a description of the Metal Experimental Stove see Chapter 6. Three tests were performed. The first two were identical with a 10% primary air inlet opening and the third with a 100% inlet opening. In all cases the tests were carried out with white fir, no secondary air was admitted, the grate-panbottom distance was 80 mm and the fire was driven at a nominal power of 6 kW. During every experiment the CO-, CO_2- and O_2-content of the flue gases and the temperatures at various spots of the stove were recorded.

For purposes of comparison formulas have been given to calculate the oxygen and nitrogen content and the excess air factor from the CO and CO_2 content. The derivation of these formulas is shown in Section 2.3 of this chapter. The formulas are:

$$[N_2] = 0,79 - 0,40\ [CO] - 0,004\ [CO_2]$$

$$[O_2] = 0,21 - 0,601\ [CO] - 0,996\ [CO_2]$$

$$\lambda = \frac{1 - [CO] - [CO_2]}{2,36\ [CO] + 4,74\ [CO_2]} + 0,21$$

where [CO], [CO_2], [O_2] and [N_2] are concentrations of the respective gases and λ is the excess air factor, which is defined as the ratio of the total amount of air drawn into the stove and the stoichiometric amount of air required for the combustion process.

Measurements and their results

Before and after each experiment the gas chromatograph was calibrated. The gases applied for calibration were: air consisting of 20,95 ± 0,01% O_2 and 78,08 ± 0,01% N_2. Nitrogen of technical quality with 99,9 ± 0,1% N_2 and a calibration gas containing 0,49 ± 0,05% CO, 6.00 ± 0,1% CO_2, 7,16 ± 0,1% O_2 and 86,35 ± 0,2% N_2.

The results of the nitrogen measurements are plotted on top of the CO-, CO_2- and O_2-graph. And the oxygen results are presented as little crosses in the graph (see figures 2.2, 2.3 and 2.4). In the tables 2.1, 2.2 and 2.3 a number of experimental results as well as the calculated values are listed. The calculated values are derived through the formulas mentioned above with values of CO and CO_2, which are read from the graphs at distinct moments. These calculated values can be compared with the experimental results. With the same CO and CO_2 concentrations also the excess air factor for that particular moment can be calculated. As is shown from the tables the excess air factors in the tail end of the tests are very high. The reason for this is that the gasstream through the stove, determined by the temperature in the chimney, is much larger then required.
For purposes of comparison in a general sense some fictitious values of CO, CO_2 and the calculated values for O_2, N_2 and λ are listed in table 2.4. For CO concentrations between 0% and 2% the nitrogen percentage shows a slight variation ranging between 79% and 78,1% N_2.

First experiment

Before and after the experiment two calibrations had been done. The relative error in both the oxygen and nitrogen value is 2%. As can be seen from figure 2.2 and table 2.1 the nitrogen contents show too low a value compared to the calculated ones. This is probably due to the limited calibration.

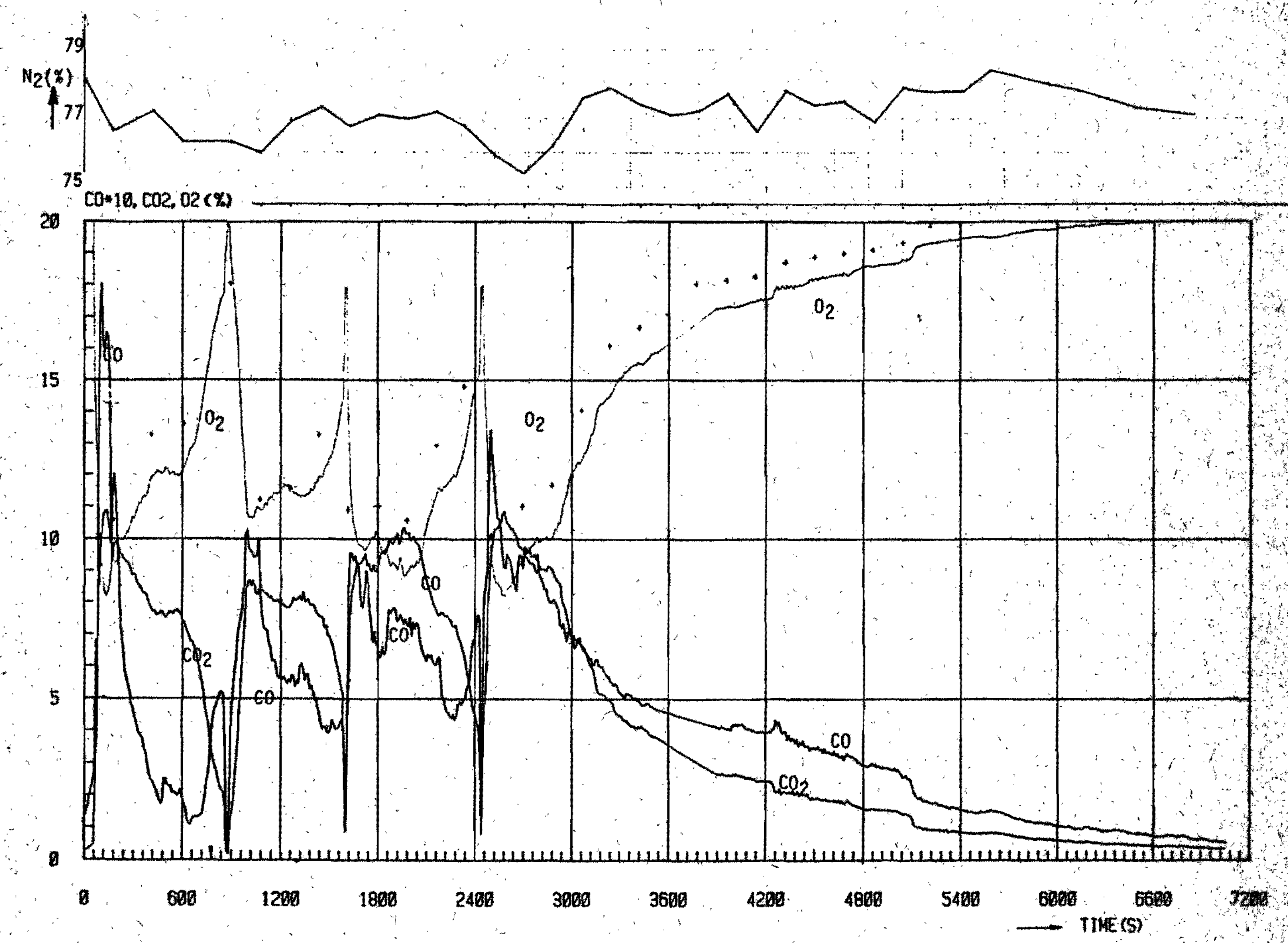

Fig. 2.2. Gas analysis chart of experiment 1

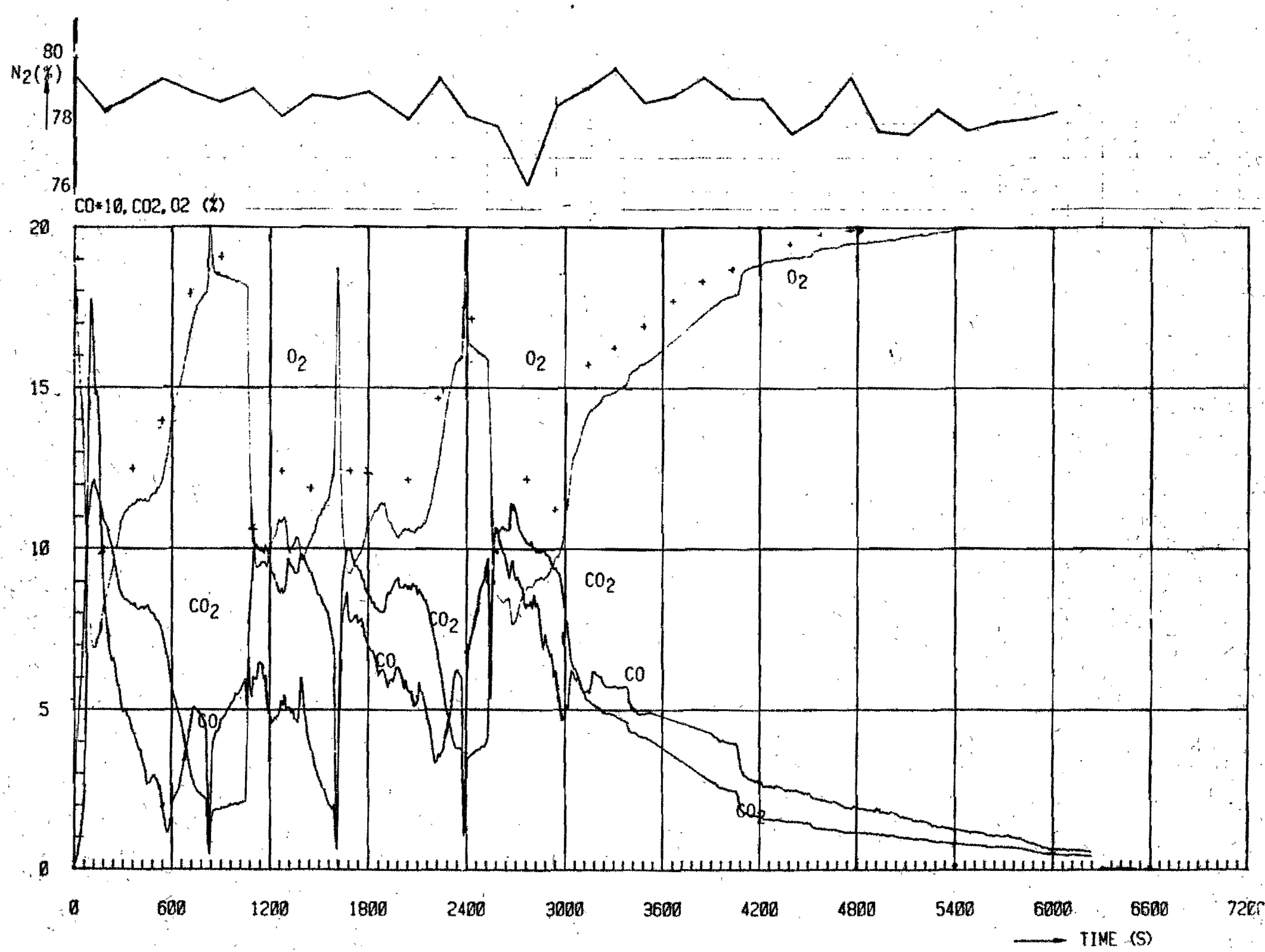

Fig. 2.3. Gas analysis chart of experiment 2.

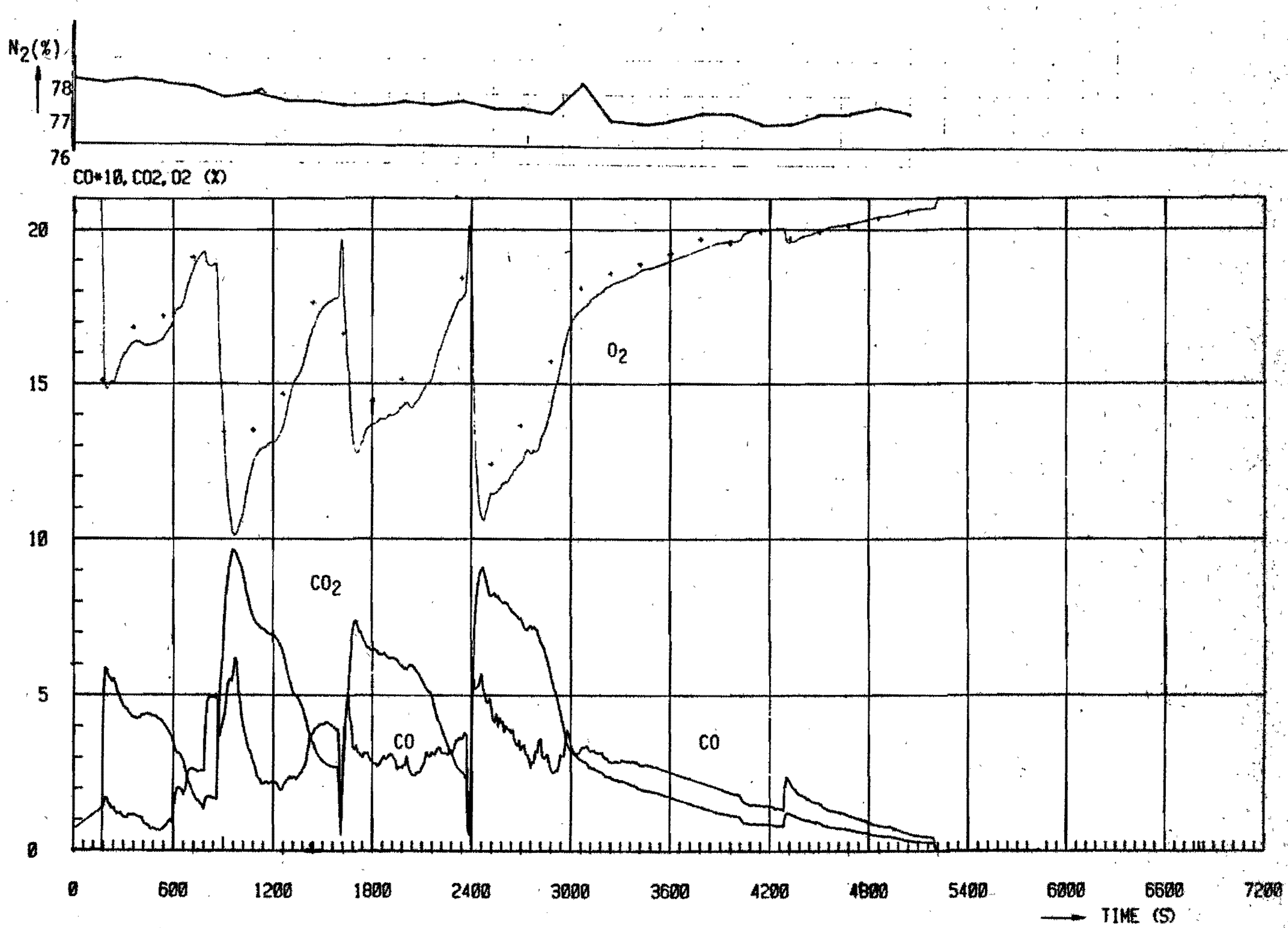

Fig. 2.4. Gas analysis chart of experiment 3.

The oxygen content shows 1-1,6% higher values compared to the paramagnetic measurement of oxygen, but it confirms with the calculated value.

Second experiment

As said before this experiment is identical to the previous one. But the calibration of the gas chromatograph was more extensive. The relative errors for the oxygen and nitrogen are respectively 2% and 1%. This time the nitrogen measurements shown in figure 2.3 match reasonably well with the calculated values. Still the paramagnetic measurement of the oxygen is 1-1,5% too low (see table 2.2).

Third experiment

This is the experiment with the primary air inlet completely open, so higher oxygen values could be expected as can be seen from figure 2.4. The paramagnetic measuring device was recalibrated but now for the range of 7-21% instead of 0-21%. This resulted in some improvement.
Again the nitrogen measurements showed too low a value of about 1%-2% (see table 2.3). The excess air factors show considerably higher values compared to the previous experiments as was to be expected.

The conclusion drawn from these experiments is that the carbon balance technique of estimating mass balances in a stove is adequate for most stove design development work.

2.3 The Ostwald combustion diagram for wood

We will start by considering wood as a fuel in some detail. The basic ingredients for forming wood are carbon dioxide and water. By means of photosynthesis CO_2 and H_2O are converted into three major constituents of wood namely cellulose $(C_6H_{10}O_5)_n$, lignin $(C_9H_{10}O_3(CH_3O)_{0,9-1,7})_m$ and hemicellulose $(C_5H_8O_4)_L$. These constituents vary with wood species. In general hard woods contain about 43% cellulose, 22% lignin and 35% hemicellulose, while soft woods have approximately 43% cellulose, 29% lignin, and 28% hemicellulose (Shafizadeh and De Groot, 1976). The different chemical compositions are responsible for varying combustion values among wood fuels. For example holo cellulose (cellulose and hemicellulose) has a combustion value of 17,46 MJ/kg whereas lignin accounts for 26,63 MJ/kg. Therefore it can be stated generally that wood with a higher lignin content also has a higher heat content (Tillman, 1978).

The composition of wood also determines how it releases its energy. Wood combustion starts with pyrolysis. In pyrolysis holo cellulose principally promotes the release of volatiles. While lignin also releases volatiles, it primarily promotes char formation (Shafizadeh and De Groot, 1976). This can be more or less derived from the chemical formula of lignin which shows a higher relative proportion of carbon compared to holo cellulose.

For purposes of calculating air requirements and combustion gas production it is sufficient to know the ultimate analysis of the fuel. This analysis provides the proportions of carbon, hydrogen and oxygen present in the wood. These constituents determine the quantities of CO_2, CO and H_2O produced by the combustion. The relative proportions of CO_2, CO and O_2 are depicted in the Ostwald diagram indicating the overall course of combustion. To realize this kind of a graphical display of the combustion process it is necessary to make some assumptions which state:

- The wood entirely burns to CO, CO_2 and H_2O, thus no hydrocarbons, soot particles etc. are formed.

- In the first instance no difference is made between the flame phase in which the hydrocarbons, in the form of volatiles, burn and the charcoal phase in which only carbon burns.

This last assumption will be relaxed at a later stage.

Quantitative determinations of soot particles and C_xH_y by TNO (Claus et al, 1981) proved that these fractions account for a very small part in the mass balance (0,1%-2%) and heat balance (0,1%-2,5%). So the first assumption is valid within reasonable limits.

When we want to drawn a relationship between the constituents C, H_2 and O_2 in the wood and the production of CO, CO_2 and H_2O in a general way, we state that wood consists of p% carbon, x% hydrogen, (8x + y)% oxygen and r% ash. These fractions are all mass fractions. Noting down the oxygen fraction in this particular form has the following reason. By weight one part of hydrogen requires eight parts of oxygen to form water. Dependent on the wood species this ratio of oxygen and hydrogen is somewhere around eight. In other words there is a deficit or a surplus of oxygen in the wood, or the oxygen just balances the hydrogen. So the value of y can be negative, positive or zero. Converting the mass fractions into molar fractions they become p/12 mol C, x/2 mol H_2 and (8x + y)/32 mol O_2, where 12, m 2 and 32 respectively are the molar masses of carbon, hydrogen and oxygen. If f is the fraction of the wood that is converted into CO it becomes:

$$f = \frac{\text{molCO}}{\text{molCO} + \text{molCO}_2} = \frac{[CO]}{[CO] + [CO_2]}$$

The volumes of the several flue gas components expressed in moles, taking 100 g. of wood as a basis, will then be:

$$\begin{array}{lll}
V_{CO} & = f \cdot p/12 & \text{molCO} \\
V_{CO_2} & = (1 - f)\, p/12 & \text{molCO}_2 \\
V_{H_2O} & = x/2 & \text{molH}_2\text{O} \\
V_{O_2} & = 0{,}21\ (V_a - V_{a.st.}) & \text{molO}_2 \\
V_{N_2} & = 0{,}79 \cdot V_a = 0{,}79 \cdot \lambda \cdot V_{a.st.} & \text{molN}_2
\end{array}$$

where $V_{a.st.}$ is the stoichiometric amount of air and V_a the total amount of air drawn into the stove. The excess air factor λ is defined as the quotient of V_a and $V_{a.st.}$.
The stoichiometric amount of combustion air then becomes:

$$V_{a.st} = \frac{1}{0.21}\left(\frac{1}{2} V_{CO} + V_{CO_2} + \frac{1}{2} V_{H_2O} - V_{O_2 \text{ in wood}}\right) \text{ mol}$$

$$V_{a.st} = \frac{1}{0,21}\left(\frac{1}{2} \cdot f \cdot \frac{p}{12} + (1 - f) \cdot \frac{p}{12} + \frac{1}{2} \cdot \frac{x}{2} - \frac{8x + y}{32}\right) \text{ mol}$$

$$V_{a.st} = \frac{1}{0,21}\left[\left(1 - \frac{1}{2}f\right) \frac{p}{12} - \frac{y}{32}\right] \quad \text{mol} \tag{2.1}$$

While the CO-, CO_2- and O_2-meters can only cope with dry flue gas, the derived concentrations are also based upon dry flue gases. Therefore we can state

$$\left.\begin{aligned} [CO] &= \frac{V_{CO}}{V'_{fl}} = \frac{f\frac{p}{12}}{V'_{fl}} \\ [CO_2] &= \frac{V_{CO_2}}{V'_{fl}} = \frac{(1-f)\frac{p}{12}}{V'_{fl}} \end{aligned}\right\} \quad V'_{fl} = \frac{\frac{p}{12}}{[CO] + [CO_2]} \quad \text{mol} \tag{2.2}$$

V'_{fl} is the amount of dry flue gas.

$$V'_{fl} = V_{CO} + V_{CO_2} + V_{O_2} + V_{N_2}$$
$$V'_{fl} = f \cdot \frac{p}{12} + (1-f) \cdot \frac{p}{12} + 0,21.(V_a - V_{a.st}) + 0,79 \cdot V_a$$
$$V'_{fl} = \frac{p}{12} + (\lambda - 0,21)\left\{\frac{1}{0,21} \quad \left(1 - \frac{1}{2}f\right).\frac{p}{12} - \frac{y}{32}\right\} \tag{2.3}$$

Combining the two equations (2.2) and (2.3) leads to:

$$\frac{\frac{p}{12}}{[CO] + [CO_2]} = \frac{p}{12} + \left(\frac{\lambda-0,21}{0,21}\right)\left(\left(1 - \frac{1}{2}f\right) \frac{p}{12} - \frac{y}{32}\right) \tag{2.4}$$

dividing by $\frac{p}{12}$ and expressing in terms of λ results in:

$$\lambda = 0,21 \left(\frac{1 - [CO] - [CO_2]}{\left(\frac{1}{2} - \frac{3}{8}\frac{y}{p}\right) [CO] + \left(1 - \frac{3}{8}\frac{y}{p}\right) [CO_2]}\right) + 0,21 \tag{2.5}$$

making [CO] explicit the same equation (2.4) results in:

$$[CO] = \frac{0{,}21 - 0{,}21\,\frac{3}{8}\frac{y}{p}\,[CO_2] - (1 - \frac{3}{8}\frac{y}{p})\,\lambda\,[CO_2]}{0{,}21(\frac{1}{2} + \frac{3}{8}\frac{y}{p}) + \lambda\,(\frac{1}{2} - \frac{3}{8}\frac{y}{p})} \qquad (2.6)$$

Drawing a similar equation including the oxygen concentration will be as follows:

$$[O_2] = \frac{V_{O_2}}{V'_{fl}} = \frac{0{,}21\,(V_a - V_{a.st})}{V'_{fl}}$$

Substituting V_a, $V_{a.st}$ and V'_{fl} will give

$$[O_2] = 0{,}21.(\lambda - 1).\frac{1}{0{,}21}.((1 - \frac{1}{2}f)\,\frac{p}{12} - \frac{y}{32})\,\frac{[CO] + [CO_2]}{\frac{p}{12}}$$

$$[O_2] = (\lambda - 1)\,((\frac{1}{2} - \frac{3}{8}\frac{y}{p})\,[CO] + (1 - \frac{3}{8}\frac{y}{p})\,[CO_2])$$

Replacing λ with equation (2.5) and writing [CO] and $[CO_2]$ as separate terms leads to:

$$[O_2] = 0{,}21 - (0{,}605 - \frac{3}{8}\frac{y}{p}\,.\,0{,}79)\,[CO] - (1 - \frac{3}{8}\frac{y}{p}\,.\,0{,}79)\,[CO_2] \qquad (2.7)$$

To get an impression of how the Ostwald diagram works, we simply state the oxygen content in the wood matches with the hydrogen in it. So y is zero. Then the equations (2.6) and (2.7) turn into the following expressions

$$[CO] = \frac{0{,}42 - 2\lambda\,[CO_2]}{\lambda + 0{,}21} \qquad (2.8)$$

$$[O_2] = 0{,}21 - 0{,}605\,[CO] - [CO_2] \qquad (2.9)$$

These equations (2.8) and (2.9) are plotted in figure 2.5. As an overall picture this diagram will tell us as a first approximation in which region the combustion process will stay.

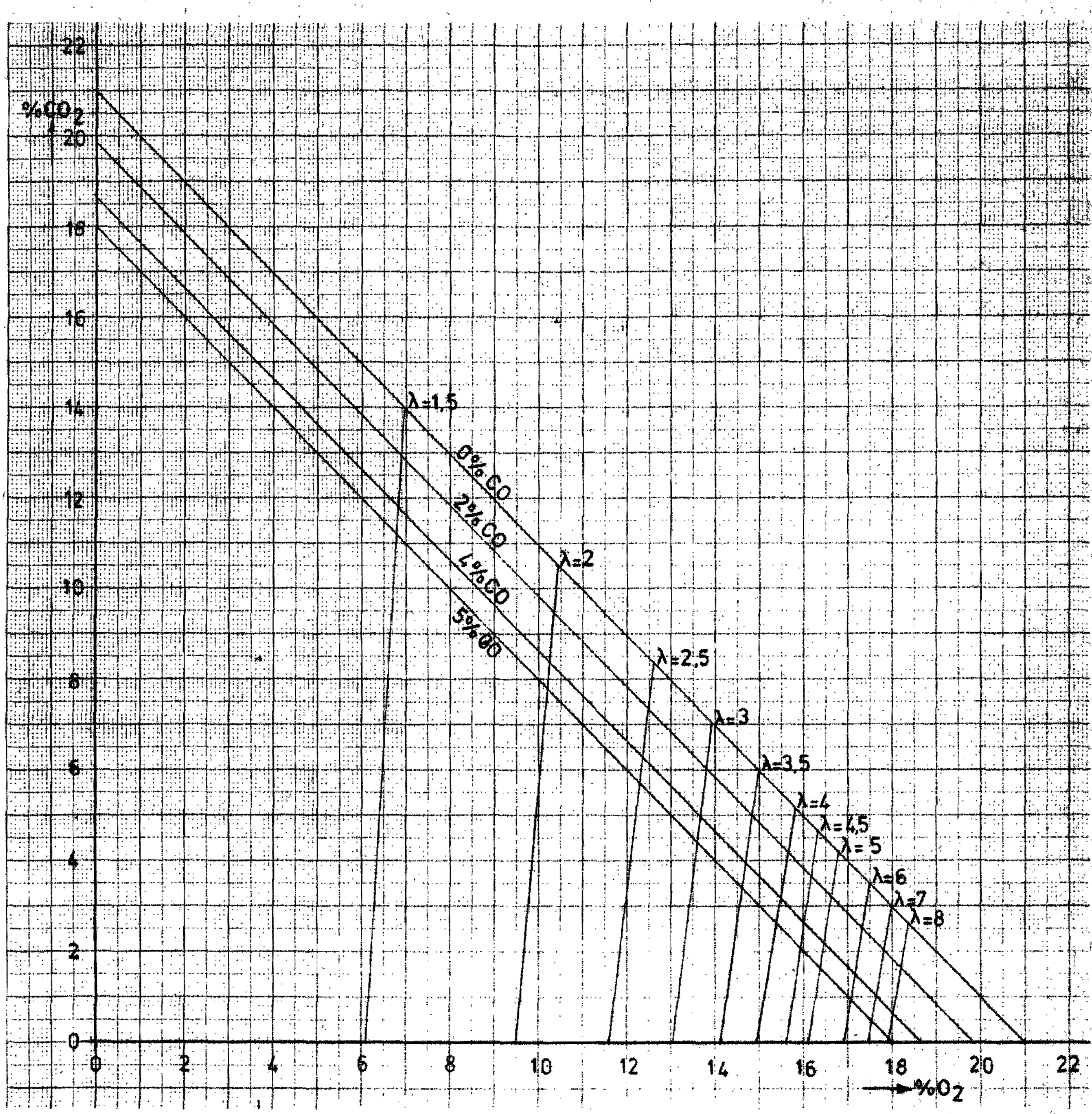

Fig. 2.5. The Ostwald diagram for white fir.

One can see that the combustion process especially applicable to wood stoves is concentrated in the shaded area bordered by [CO] lines corresponding to 0% and 2%, the line at which λ = 1,5 at the top and the [CO_2] = 0%-line at the bottom. Since this band is too narrow to enable quantitative reading, a more spacious presentation of the Ostwald diagram would be preferable. This will be done for the examples of white fir, Detarium microcarpum and the flame phase of Detarium microcarpum.

The two wood species white fir a soft wood and Detarium microcarpum, an african hardwood are taken as examples because they represent the cases of surplus and insufficient oxygen to account for the combustion of hydrogen in the wood. According to the ultimate analysis carried out by TNO-Apeldoorn the samples consist of:

White fir (wf)		Detarium microcarpum (dm)	
0,9	% ash	1,9	% ash
50,7	% C	48,8	% C
5,3	% H_2	6,1	% H_2
43,1	% O_2	43,2	% O_2
18,7	MJ/kg combustion value	20,35	MJ/kg combustion value

One can expect that due to the presence of lignin, which has a deficit of oxygen, the oxygen-hydrogen ratio of 1:2 would be disturbed. In most cases this is true, like in Detarium microcarpum, but in the case of white fir it is not. In white fir the oxygen-hydrogen ratio is very close to 1:2 this is probably due to the presence of resins in the wood which make up for the shortage of oxygen.

From hereon we will derive the formulas for the two wood species. Therefore we have to determine y, which represents the difference between the oxygen and hydrogen content in the wood. y = % O_2 - 8 . % H_2. So for white fir y = 0,7 and for detarium m. y = - 5,6. The latter clearly indicating a lack of oxygen in the wood.

Substituting these values plus the values for the carbon fraction into the equations (2.6) and (2.7) we obtain for white fir

$$[CO] = \frac{0,424 - 0,002\ [CO_2] - 2,011\ \lambda\ [CO_2]}{\lambda + 0,214} \tag{2.10}$$

and

$$[O_2] = 0,21 - 0,601\ [CO] - 0,996\ [CO_2] \tag{2.11}$$

or

$$[CO_2] = 0,211 - 0,603\ [CO] - 1,004\ [O_2] \tag{2,12}$$

for Detarium microcarpum

$$[CO] = \frac{0,387 + 0,017\ [CO_2] - 1,921\ \lambda\ [CO_2]}{\lambda + 0,117} \tag{2.13}$$

and

$$[CO_2] = 0,203 - 0,618\ [CO] - 0,967\ [O_2] \tag{2.14}$$

The equations (2.10) and (2.12) are presented in figure 2.6 and the equations (2.13) and (2.14) in figure 2.7.

Finally we set a side the assumption in which is stated that no difference is made between the flame phase and the charcoal phase. The charcoal phase which is considered to have no oxygen and hydrogen at all is described by equation (2.8) and (2.9). In the case of white fir the situation between flame and charcoal phase is very much alike. Comparing the equation (2.8) with (2.10) and (2.9) with (2.11) one can see that they agree with each other fairly well. This is obvious because the value of y is small which means that oxygen and hydrogen are nearly complementary to one another. And because oxygen and hydrogen happen to be only in the volatiles, thus flame phase, the same situation occurs as with white fir as a whole. In contrast to this Detarium microcarpum has a surplus of hydrogen in the volatiles, which has to be burned by external combustion air. Assuming that charcoal and volatiles account approximately for respectively 20% and 80% (Brame and King, 1967).

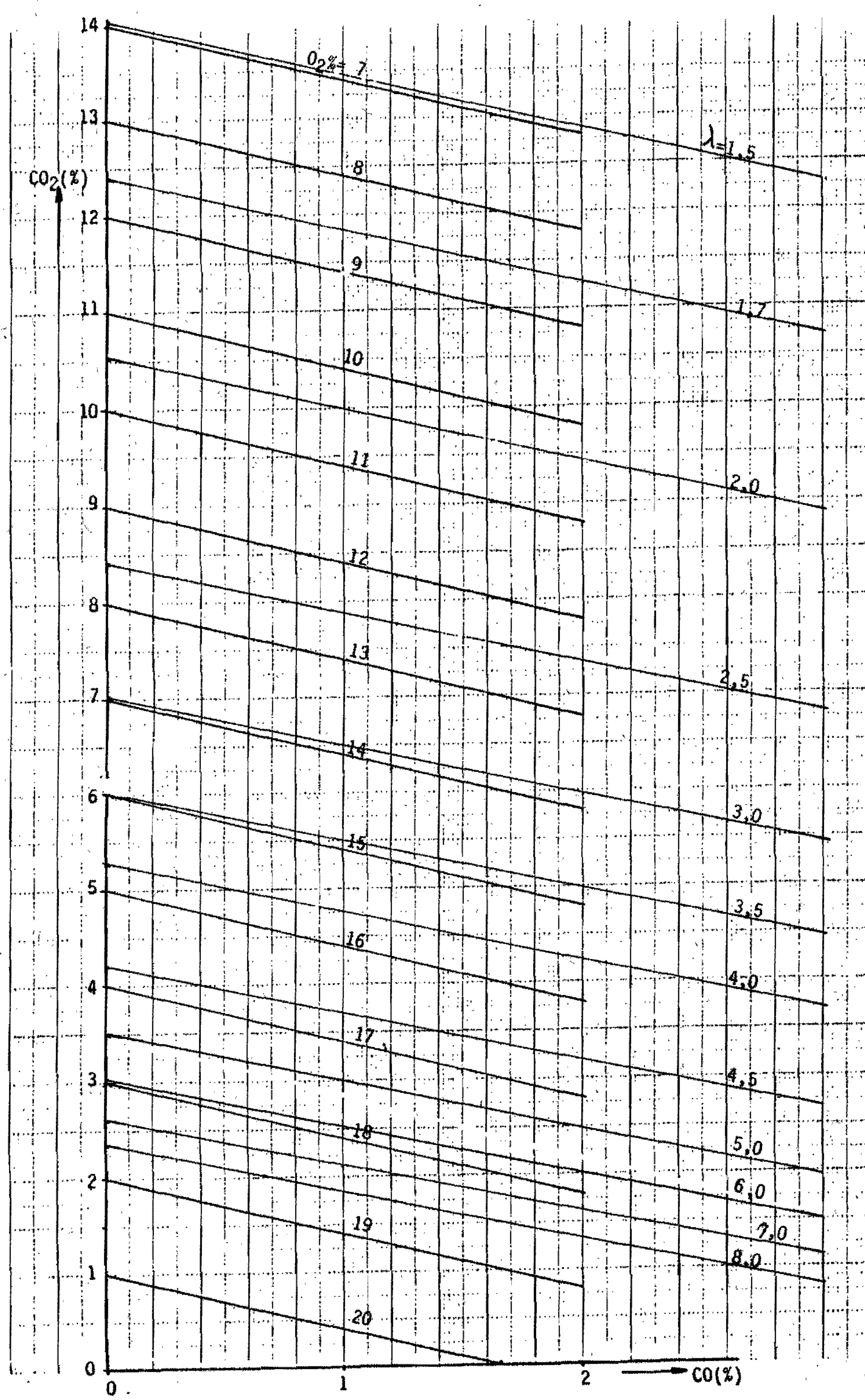

Fig. 2.6. The Ostwald diagram for white fir in the range between 0% and 1% CO.

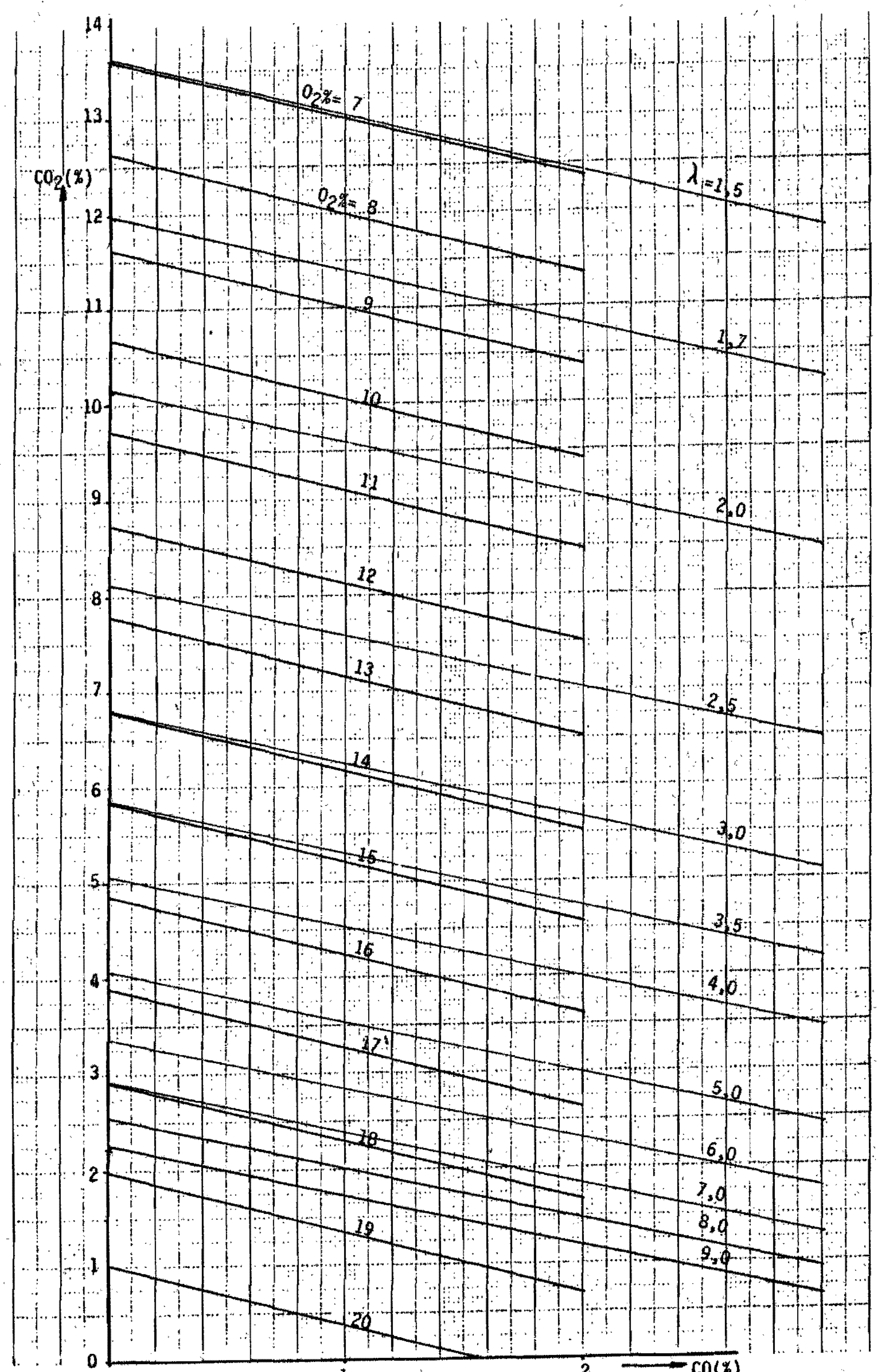

Fig. 2.7. The Ostwald diagram for Detarium m in the CO range of 0 - 2%.

The wood consists of:

$$\left.\begin{array}{l} 1{,}9\ \%\ \text{ash} \\ 28{,}8\ \%\ \text{carbon} \\ 6{,}1\ \%\ \text{hydrogen} \\ 43{,}1\ \%\ \text{oxygen} \end{array}\right\} \rightarrow \text{volatiles}$$

$$20{,}0\ \%\ \text{carbon} \rightarrow \text{charcoal}$$

The volatile parts have to be reworked into weight fractions again for calculation y and the carbon fraction p. We obtain:

$$\left.\begin{array}{l} 36{,}9\ \%\ C \\ 7{,}8\ \%\ H_2 \\ 55{,}3\ \%\ O_2 \end{array}\right\} \rightarrow y = -7{,}1$$

substituting these values into equations (2.6) and (2.7) results in:

$$[CO] = \frac{0{,}367 + 0{,}026\,[CO_2] - 1{,}874\,\lambda\,[CO_2]}{\lambda + 0{,}157} \qquad (2.15)$$

and

$$[CO_2] = 0{,}199 - 0{,}626\,[CO] - 0{,}946\,[O_2] \qquad (2.16)$$

These equations are presented in figure 2.8.

Discussion

If one has no access to sophisticated apparatus like infrared analysers or gas chromatographs but still desires to draw a heat balance of the stove at hand, the Ostwald diagram can be of use. One can attain to different levels of accuracy. The crudest one does not even require the use of the Ostwald diagram and it defines the excess air factor as $\lambda = \frac{0{,}21}{[CO_2]}$; this follows directly from equation (2.5), assuming that there is no CO in the flue gas and hydrogen and oxygen in the wood match each other. By means of the following examples the sensible heatloss is determined. For this one needs to know the stoichiometric amount of air and this amount is dependent on the CO concentration. In the graph of figure 2.9 this dependence is shown by means of equation (2.1). To burn 1 kg of white fir and assuming there is no CO thus f = 0, the stoichiometric amount is 4,5 m^3 air.

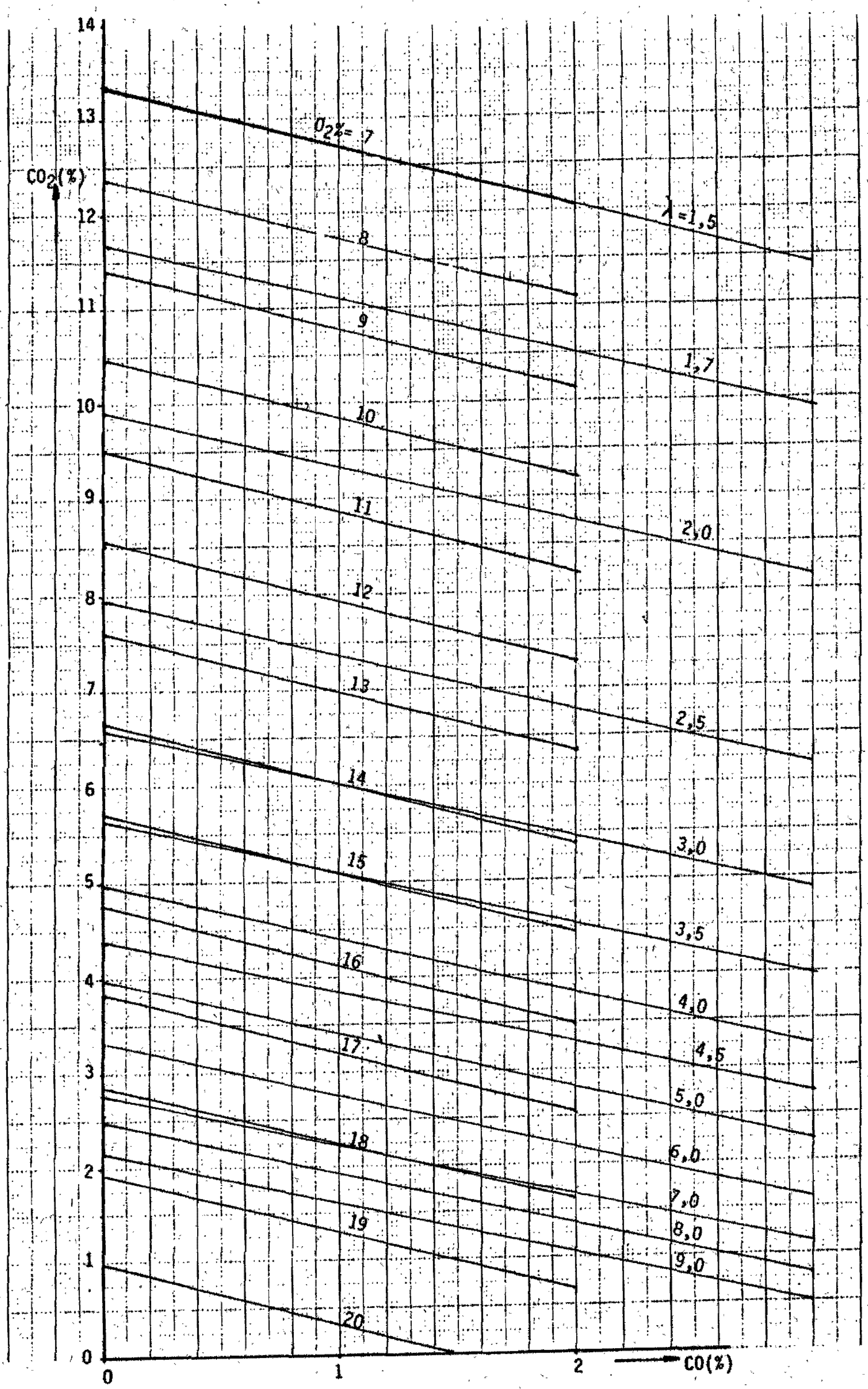

Fig. 2.8. The Ostwald diagram for the flame phase of Detarium m in the CO range of 0 - 2%.

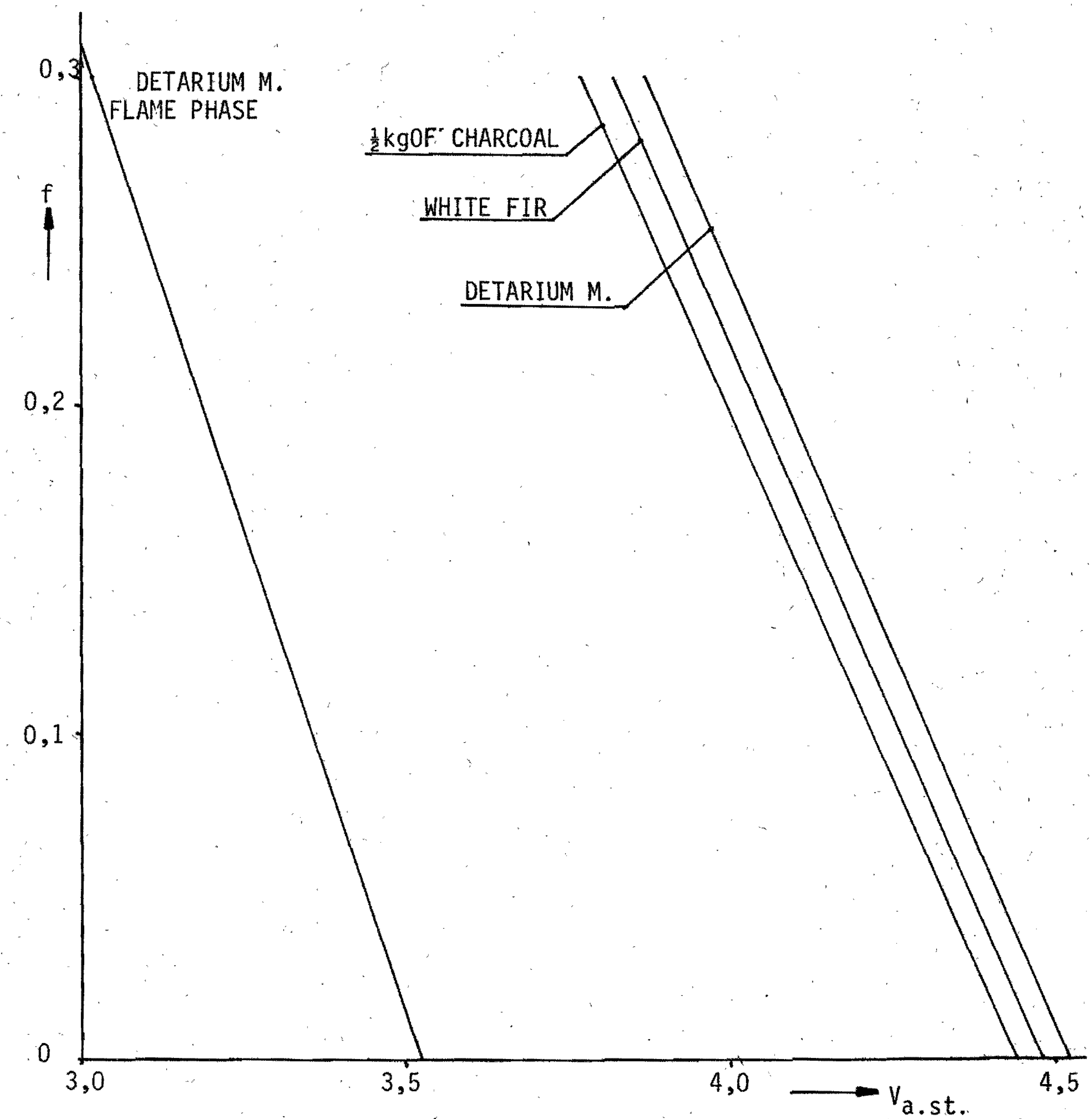

Fig. 2.9. The stochiometric amount of air as a function of the CO concentration.

With a CO_2 concentration of 10% $\lambda = \frac{0,21}{0,1} = 2,1$. The total amount of air drawn into the stove is then 2,1 . 4,5 = 9,5 m^3 of air with a specific density of 1,3 kg/m^3. The total mass going through the chimney is 2,1 . 4,5 . 1,3 + 1 = 13,3 kg of fluegas. Assuming that the flue gas has a temperature of 150^{o}C and a specific heat of 1,05 $\frac{kJ}{kg.K}$, the heat carried away by the flue gas becomes 2,1 . 10^3 kJ. Relating this loss to the heat input which is the combustion value of white fir (18,73 . 10^3 kJ/kg). The sensible heat loss accounts for 11%. If we are able to measure carbon monoxide and we measure 1% CO, then $\lambda = 2$ (see figure 2.6) and $V_{a.st} = 4,3$ m^3 (see figure 2.9). Following the same lines like the sample above the sensible heat loss accounts for 10,2%. If we add to that the latent heat loss due to CO formation, which is in the order of 6,3% (1,8 . 10^3 kJ) the total heat loss from the flue gases becomes 16,5%.

When knowing the ultimate analysis of a certain wood species, for instance Detarium microcarpum one can adapt the Ostwald diagram and depict λ more accurately and the same applies for the stoichiometric volume as a function of f. For the concentration of CO_2 = 10% and CO = 1%, $\lambda = 1,9$ and $V_{a.st.}$ becomes a little over 4,3 cm^3. The sensible heat carried away now is (4,3 . 1,9 . 1,3 + 1) . 150 . 1,05 = 1830 kJ. With the combustion value of 20350 kJ/kg for Detarium microcarpum the relative heat loss is 9%. At last we calculate the heat loss in the flame phase only. For [CO] = 1% and [CO_2] = 10% we find $\lambda = 1,87$ and $V_{a.st.} = 3,37$ m^3. Q_{sens} = 1448 kJ per 1 kg of volatiles. However burning 1 kg of wood approximately 0,8 kg of volatiles is generated. Correcting the heat loss for this amount relating it to 1 kg of wood results in a heat loss of 5,7%. The heat carried away during the charcoal burning becomes $[(\frac{2}{5} . 4,24 . 2 . 1,3) + 0,2] . 150 . 1,05 = 727$ kJ $\rightarrow$ 3,6%.

Another purpose served by the Ostwald diagram is that one can determine instantaneous excess air factors when one knows the CO and CO_2 concentrations at certain moments. Examining gas analyses of experiments it became evident that at the last stage of an experiment very high excess air factors where attained ($\lambda > 6$).

Although the burning rate of this end phase is very low, chimney temperatures at this stage do not drop considerably and hence the air sucked into the stove reduces not that much also. The air flow in this stage can be diminished by means of dampers, whether this will improve the performance of the stove needs closer investigation.

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time (min)	% CO IR	% CO_2 IR	% O_2 PM	% O_2 GSC	% O_2 CALC	% N_2 GSC	% N_2 CALC	λ CALC
3	1,20	9,9	9,4	10,6	10,4	75,7	78,5	2,6
7	0,20	7,7	12,0	13,3	13,2	76,4	78,9	3,5
10	0,20	7,5	12,1	13,5	13,4	75,3	78,9	3,6
15	0,51	1,6	18,4	17,9	19,1	75,3	78,8	14,5
18	0,78	8,4	10,9	12,3	12,1	74,9	78,7	3,1
21	0,56	7,9	11,6	12,6	12,8	76,0	78,7	3,3
24	0,43	7,5	12,0	13,2	13,2	76,6	78,8	3,5
27	0,95	7,1	12,1	12,0	13,6	75,8	78,8	3,7
30	0,65	9,1	10,0	11,0	11,5	76,3	78,7	2,9
33	0,72	10,2	9,0	10,6	10,4	76,1	78,7	2,6
36	0,62	7,6	11,5	13,0	13,0	76,4	78,7	3,4
39	0,52	6,05	13,0	4,7	14,6	75,9	78,8	4,3
42	1,14	10,3	8,7	10,0	10,0	74,8	78,5	2,5
45	0,97	9,5	9,4	11,0	10,9	74,2	78,6	2,7
48	0,81	8,9	10,2	11,7	11,6	75,2	78,6	2,9
51	0,66	6,6	12,6	14,0	14,0	76,9	78,7	3,9
54	0,55	4,9	14,5	16,0	15,8	77,3	78,8	5,3
57	0,48	3,6	15,5	16,5	17,1	76,6	78,8	7,1
66	0,41	2,7	17,3	18,1	18,1	77,0	78,8	9,6
69	0,39	2,5	17,5	18,2	18,3	75,7	78,8	10,1
72	0,39	2,1	18,0	18,6	18,7	77,1	78,8	11,9
75	0,35	2,0	18,2	18,7	18,8	76,6	78,9	12,6

Table 2.1 : Measured and calculated values belonging to experiment 1.
IR = Infrared Gas Analyzer; GSC = Gas Solid Chromatography;
PM = Magnetic Oxygen Meter; CALC = Calculated values.

time (min)	% CO IR	% CO_2 IR	% O_2 PM	% O_2 GSC	% O_2 CALC	% N_2 GSC	% N_2 CALC	λ CALC
21,15	0,5	8,7	10,9	12,3	12,0	78,2	78,8	2,4
15	0,5	2,0	18,5	19,4	18,7	78,6	78,8	9,4
24,15	0,4	9,3	10,3	11,8	11,5	78,8	78,8	2,2
30	0,7	8,6	10,7	12,2	12,0	78,9	78,7	2,4
34	0,6	8,9	10,6	12,0	11,7	78,1	78,7	2,3
37,15	0,4	6,1	13,2	14,7	14,7	79,3	78,8	3,4
43	1,1	10,5	8,5	9,8	9,8	77,9	78,5	1,9
46	0,8	10,1	8,9	12,3	10,4		78,6	2,0
49	0,6	9,4	9,7	11,2	11,2	78,5	78,7	2,2
52,10	0,6	5,3	14,2	15,6	15,3	79,0	78,7	3,8
58	0,5	4,1	15,8	16,9	16,6	78,6	78,8	4,9
67	0,4	2,5	17,8	18,7	18,2	78,7	78,8	7,8
73	0,2	1,5	19,0	19,5	19,4	77,7	78,9	13,2

Table 2.2 : Measured and calculated values belonging to experiment 2.
IR = Infrared Gas Analyzer; GSC = Gas Solid Chromatography;
PM = Magnetic Oxygen Meter; CALC = Calculated values.

time (min)	% CO IR	% CO_2 IR	% O_2 PM	% O_2 GSC	% O_2 CALC	% N_2 GSC	% N_2 CALC	λ CALC
3	0,2	5,7	16,2	15,0	15,2	77,8	78,9	3,6
6	0,1	4,3	16,4	16,7	16,6	77,8	78,9	4,8
9	0,1	4,0	16,5	17,0	17,0	77,8	79,0	5,2
12	0,3	1,7	18,9	19,0	19,1	77,7	78,9	11,5
15	0,4	6,8	13,7	13,2	13,9	77,4	78,8	3,0
18	0,3	7,4	12,5	13,4	13,4	77,5	78,9	2,8
21	0,2	6,3	13,7	14,4	14,6	77,3	78,9	3,3
24	0,4	3,4	16,8	17,5	17,4	77,3	78,8	5,9
27	0,4	2,7	17,8	16,5	18,1	77,2	78,8	7,3
30	0,3	6,4	13,7	14,3	14,4	77,2	78,9	3,2
33	0,3	5,9	14,3	15,0	14,9	77,3	78,9	3,5
36	0,3	4,9	15,2	-	15,9	77,2	78,9	4,2
39	0,4	2,5	17,8	18,3	18,3	77,3	78,8	7,9
48	0,4	8,2	11,4	12,3	12,5	77,1	78,8	2,0
45	0,3	7,5	12,5	13,5	13,3	77,1	78,8	2,7
48	0,3	5,4	14,4	15,5	15,4	77,0	78,9	3,8
51	0,3	2,9	17,4	18,0	17,9	77,8	78,9	6,9
54	0,3	2,3	18,2	18,4	18,5	76,8	78,9	8,6
57	0,3	1,9	18,7	18,7	18,9	76,7	78,9	10,4
60	0,3	1,7	19,0	19,0	19,1	76,8	78,9	4,6

Table 2.3 : Measured and calculated values belonging to experiment 3.
IR = Infrared Gas Analyzer; GSC = Gas Solid Chromatography;
PM = Magnetic Oxygen Meter; CALC = Calculated values.

% CO	% CO_2	% O_2	% N_2	λ
0	0	21,0	79,0	8
0	1	20,0	79,0	21,2
0	4	17,0	79,0	5,3
0	8	13,0	79,0	2,6
1	1	19,4	78,6	14,1
1	4	16,4	78,6	4,7
1	8	12,4	78,6	2,5
2	2	17,8	78,2	7,0
2	10	9,8	78,2	1,9
2	15	4,8	78,2	1,3
34	0,4	0,0	65,4	1,0

Table 2.4 : Fictitious values of CO and CO_2 and the consequent values for O_2, N_2 and λ.

Appendix A

Instrumentation for Gas Solid Chromatography.

- F&M (Hewlett & Packard) Scientific 700 laboratory chromatograph mode: 700-0119F, serialno.: 809-B01938.
- Control cabinet
  model: 700-0119F, serialno.: 809-T01938.
- HP stripchart recorder, 1 channel, model: 7127A
- Integrator, homemade
- Digital Voltmeter Fluke
  model: 8800A, digital multimeter
- Columns
  MOLSIEVE 13X80 - 100 Mesh 3,6 m * 3,2 mm ϕ
  PORAPAK Q 80 - 100 Mesh 0,5 m * 3,2 mm ϕ
- Detector
  Thermal conductivity cell (Katharometer)
- Amplifier 0-1000X
  Electro instr. inc., model: A20B-2
- Carrier Gas: Helium
  Gas flow 20 ml/min
- Dryer
  model: Mini Dryer 125-48P, 1,2 m * 3,2 mm ϕ
- Flowmeters Porter Instrumentation Comp. Inc.
  model: 65A-125-3 with regulator
  max. flow 130 ml/min (air)
- Pressure regulator Porter
  model: 8286
- Sample Valve 8-port sampling valve
- Soap-film flowmeter.

3 THE EFFECT OF BAFFLES ON THE PERFORMANCE OF THE NOUNA WOOD STOVE

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3.1 Introduction

To see what effect could be expected from baffles a number of experiments have been carried out with different baffle constructions in the existing Nouna Wood Stove. The aim of these experiments was to improve the efficiency of the Nouna Stove. On the other hand an attempt is made to deliver new concepts for consideration by the fieldworkers in Upper Volta. This report shows the set up of the baffle experiments and it shows that the efficiency of a stove can be improved with changes of the construction.
Special attention is given to reduce the standard minimum heat output of the existing Nouna Wood Stove without a remarkable deterioration of the combustion quality. For the best modification the influence of several variables on the efficiency and the combustion performance is studied.

3.2 The effects of baffles

3.2.1 Design of the stove

A complete description of the Nouna Stove has already been published in Prasad (1981).
For the baffle experiments, however, this stove is modified in various ways. The set up of the modifications introduced in the original stove design are presented in figure 3.1.
The various modifications are specified by the lettres A, B and C. The discussion of the results wil mainly be restricted to the "C" modification. Figure 3.2 shows a complete drawing of the "C" stove. The baffle is constructed out of fire bricks and concrete and can easily be removed out of the stove. The gap between the bottomside of the second pot and the baffle is 0.02 m.
The length of the chimney is still 1 meter and is provided with a chimney damper which can adequately control the draft.
An impression of the "C" stove is given by means of a picture in figure 3.3.

3.2.2 Experimental details

The performance of the modified stove is determined for a number of experiments specified by A and B.
With these experiments for the first and second pan 4 l and 2 l of water were used respectively.
However, for the experiments specified by "C", another pan was placed in the heating chamber. The diameter was 0.26 m and the height 0.155 m. The amount of water for these experiments was 4 l for both the pans. The experimental set up of the measurements was similar to that of the original Nouna Wood Stove and is given in Prasad (1981). These tests included the water boiling tests and the flue gas analysis. The fuel used in the majority of the experiments was planed oven-dry white fir with a size of 0.02 * 0.03 * 0.2 m and with a density of about 350 kg/m^3. The duration of the experiments varied between 0.5 and 2 hours.

3.2.3 Effect of various baffle constructions on the efficiency

To see what effect could be expected from baffles a number of experiments have been carried out with different baffle constructions in the Nouna stove. The results of the experiments are summarized in tables 3.1 and 3.2.
To see the progress in efficiency of the stove, by introducing the various modifications, the average values of the experiments A, B and C are presented in table 3.3 and plotted in figure 3.4. The figure shows that in general the stove efficiency can be improved by introducing a baffle.

Modification "A"

The reduction of the flue gas channel with 65% for the A modification results in a higher efficiency especially for the first pan. This can be attributed to a higher combustion temperature, caused by the lower amount of air that is flowing through the stove. Consequently more radiation heat is transferred to the pan and also the convective heat part is more important. Apart from the higher flue gas temperature the convective heat transfer has improved because the baffle prevents the flames to turn off the first pan.

The efficiency of the second pan is influenced by the alteration of the flue gas pattern, which results in an increased convective heat transfer.

Modification "B"

The experiments with the B modification show another improvement of the efficiency of the stove. A point that has to be mentioned in this case is the removing of the flue gas duct and the sand which filled up the space between the inner and outer wall of the stove. Removing the flue gas duct created the possibility to increase the height of the baffle.

Both radiation and convection are positively influenced by the increased height of the baffle. In addition to this one side of the first pan may receive convective heat now. This explains the higher efficiency of the first pan.

Undoubtedly the smaller gap between the baffle and the second pan is responsible for the increase in efficiency of this pan.

Modification "C"

The best results are obtained with this modification. The most important aspect of this modification is the replacement of the second pan by a pan with a larger diameter. The new pan has a 30% larger diameter, corresponding with 0.26 m. As a consequence of this replacement, the diameter of the heating chamber changed to 0.275 m.
The adapted baffle under the second pan is made of a concrete cylinder while the gaps are filled up with sand, especially at the connection with the part of the baffle in the flue gas duct.
The gap between the bottom of the pan and the baffle is about 0.02 m. As to the importance of this gap nothing can be concluded yet. However, experiments have showed that a smaller gap influences the combustion quality in the negative sense. On the other hand it can be expected that the convective heat transfer will increase by diminishing the gap.

Compared with the original Nouna stove the "C" Nouna stove efficiency has improved with 50% for the first pan and even with 180% for the second pan in the relative sense.

The improvement of the efficiency of the first pan can be attributed to a higher combustion temperature. This is because less combustion air is flowing through the stove now. Figure 3.7 illustrates this. The figure shows the excess air factor versus the heat output for the original and modified Nouna Stove.

The remarkable improvement of the second pan efficiency can undoubtedly be attributed to the increased pan diameter.

The obtained data from the experiments with the "B" and "C" stove present the opportunity to make an exercise concerning the relationship between the efficiency and the heat transfer surface of the pan. There is a suggestion that this relationship for the second pan might be a proportional one, within a certain range of relative surface enlargment (Claus et al 1982). A difference between the present case and the results of the earlier work is that the surface area increase in the latter was obtained by sinking the pan in the stove body, while in the present work the area increase was achieved by increasing the pan diameter. In spite of this difference we use the result of Sulilatu and Claus to convert the measured efficiency for the smaller second pan ("B") into the expected efficiency due to surface enlargement. This calculation results in an efficiency of 11%, while the measured efficiency for the bigger second pan ("C") is 10.8% (table 3.3).
Although the relative surface enlargement lies at the top of the lineair interval, there is a good agreement between expected and measured efficiency. This exercise therefore sustains the above mentioned idea.

The performance of the "C" Nouna Stove is studied in more detail in the following chapter.

3.2.4 The "C" Nouna Stove

3.2.4.1 *Effect of the heat output of the fire on the efficiency*

To investigate the influence of the heat output a number of experiments were performed in the way as described in Prasad 1981. These experiment have been carried out with a damper position of 25%. The data summarized in table 3.1 are plotted in figure 3.5. It shows the total efficiency as well as the efficiency for the first and second pan as a function of the heat output of the fire. Under the conditions investigated, the efficiency of the stove varies from 30% to 39%. The efficiency of the first pan is about 2 times higher than the efficiency of the second pan. Under normal conditions the heat output of the fire varies between about 4 and 8 kW.

However, the minimum heat output can be reduced to about 2 kW when the combustion chamber is fully closed and draft is reduced.
For these experiments see table 3.1 exp. nr. 23, 13 and 26.
Experiment nr. 23 started with a hot stove and with a reduced draft of about 1.5 Pa. Under these conditions and a fully closed combustion air damper a heat output of 3 kW can be realised.
The experiments nr. 13 and 26 present the results when the heat output is reduced to about 2 kW. The experiment nr. 13 started with a charge of 4 pieces of wood in 10 min. corresponding to 6.3 kW. After three charges and the burning away of the remaining charcoal the evaporated water was measured by weighing and replaced into the stove. Now the experiment was continued with 2 pieces of wood at a charge time of 15 minutes. The damper was fully closed while the draft was reduced from 3.4 Pa to 0.98 Pa. The results of this experiment are summarized in table 3.4. It shows the evaporated amount of water during the first and second period in g/min. It is rather satisfactory that the efficiency does not drop. The table shows that the efficiency of the first pan increases while the efficiency of the second pan drops. An explanation for this can be that because of the low burning velocity of the wood, most of the convective heat is transferred to the first pan.
Consequently less heat is left for the second pan.

3.2.4.2 *Effect of the combustion air damper on the efficiency*

To show the influence of the combustion air damper position on the efficiency of the Nouna Wood Stove, the experimental data of table 3.1 at the same chimney damper positions are plotted in figure 3.6.
The graph shows the efficiency of the first and second pan, as well as the total efficiency. In contradiction to experiments reported in (Prasad 1981), the recent experiments point out that the damper position hardly influences the efficiency. The variation in efficiency for 12.5% to fully open air intake is in the order of 4 percentage points.
The insensitivity of the efficiency for the damper position can probably be attributed to the dimensions and the construction of the stove. In contradiction to the original Nouna Wood Stove the

main pressure drop occurs over the baffle. This is a really important aspect of the improvement of the Nouna Wood Stove. Most probably this is also valid for other existing stoves of the same type as the Nouna Wood Stove.

3.2.4.3 *Effects of the chimney draft on the efficiency*

Earlier experiments, described in (Prasad 1981), already indicated that efficiency can be influenced by draft. In spite of the limited experiments with variable draft, it can be carefullu concluded that the efficiency increases at a reduced draft. The experiments also indicate that a variable chimney damper is very useful. The minimum output of 2 kW could only be achieved by adjusting this damper.
The usual way to establisch the minimum draft is to reduce the chimney damper position until a point has been reached at which smoke is emitted through the front side of the stove. These manipulations hardly influence the combustion quality.

Some experiments show that it is even possible to influence the ratio of the total generated heat over the pans by means of regulation with the chimney damper, see the boiling time of the two pans for the experiments 21 and 22. It would be worthwhile to perform further experiments to clarify this effect.

3.2.4.4 *Effect of the initial condition of the stove on the efficiency*

In order to test the influence of the initial condition of the stove on the efficiency the "C" Nouna Stove was operated several times in cold condition and two times in hot condition. The first two tests - 28 and 28' - were carried out on the same day at an interval of 4 hours. 28 was at high power and 28' was at low power, the latter showing an efficiency increase of about 7 percentage points. Much of this increase in efficiency should be attributed to the change in power ratter than the change in initial condition (see figure 3.5).

The second set 29 and 29' followed the same pattern as above but at the same power level, and this time no change in efficiency was observed. The conclusion drawn for this is that the time constant for the stove was approximately three hours and as such the tests 28' and 29' should be considered to correspond to cold condition.

In order to clearly demonstrate the effect of a hot stove on the efficiency, two more tests - 30 and 30' - were conducted under real hot conditions. This condition was realized by operating the stove for 3 hours at the same power before starting the experiment. The efficiency in this case was 6 percentage points higher than the results for the tests 28', 29 and 29'.

The data of the flue gas composition and the average fluegas temperature for the experiments with a cold stove show, that the sensible heat loss is approximately 11% of the generated heat, see section 3.2.6. For the hot stove this is about 17%.

3.2.5 Combustion performance of the "C" Nouna Wood Stove

3.2.5.1 *Effect of the heat output on the combustion performance*

Figure 3.7 shows the excess air factor as a function of the heat output for the "C" and original Nouna Wood Stove. Over a heat output range from 4 to 9 kW the excess air factor of the "C" Nouna Wood Stove decreases from 3 to 1.5. The original Nouna Stove, on the other hand, draws in much higher quantities of air into the stove and moreover the excess air factor is only mildly influenced by the power level of the fire. Figure 3.8 compares the CO production (9/kg of wood burned) in the two stoves. While both the stoves show approximately the same CO content at the low power end, the modified "C" stove shows a rapidly increasing CO formation with increasing power of the fire. This poses an important design dilemma. The modified "C" stove shows a total efficiency of about 33% in comparison with an efficiency of 13% for the original design (see figure 3.4). The resolution of this dilemma needs further investigation.

The "C" stove as it stands now needs to be operated at low power outputs and figure 3.5 shows that this procedure will not have an adverse influence on the efficiency.

To gain more insight into the problem, further results are presented in figures 3.9 and 3.10. Figure 3.9 shows the CO_2 and CO concentrations as a function of the power level of the fire. The CO_2 concentration increases from 6 to 11% and the CO concentration from 0.3 to 1.3%. Unless the chimney gases are led outside from the kitchen, the higher values of CO can pose a health hazard. Figure 3.10 shows that the efficiency is more or less independent of the CO production.

The last result suggests that the loss in efficiency due to the formation of CO is compensated by the reduction in heat loss in some other part of the heat balance in the stove. This suggestion was tested by examining the results in table 3.2 on combustion quality corresponding to 25% damper position. There appears to be a general indication in support of the statement that sensible heat loss reduces with increases in CO content. However we must caution that this is an ambiguous result because small changes in power level produce significant changes in the results (see table 3.6).

It is interesting to compare the combustion quality of the Nouna "C" stove with more sophisticated domestic heating stoves marketed in Holland which are presently being tested in TNO labs at Apeldoorn. The CO-content in the Nouna "C" lies in the range of 40-130 g/kg while the domestic heating stove shows a range of 45-110 g/kg for the same power level of the fire.

It has te be noted that the presented CO concentrations are the average values over an experiment. A real impression of the instantaneous concentration is given in the figures 3.13 and 3.16. These are the results of experiments 26 and 27. Figure 3.13 shows the emission at the maximum and minimum heat output that can be realised with the "C" stove. This corresponds with experiment nr. 26. It shows that for the first part of the experiment a CO concentration as high as 3% can be reached while the average value is 1.1%. Figure 3.15 shows the emission of the flue gases at a reduced heat output of 4.5 kW. This corresponds with experiment nr. 27. The maximum CO concentration of 0.7% is low relatively speaking though still unacceptable from health point of view.

3.2.5.2 *Effect of the combustion air damper on the combustion performance*

Figure 3.11 shows the excess air factor versus the damper position. The figure shows that the excess air factor varies from 2 to 3.5 between air intake at 12.5% of the damper opening and fully open air intake. The figure further shows that the damper appears to control the combustion air between 12.5% and 50% of the damper opening. Figure 3.12 shows a decreasing CO concentration in this range. Above 50% damper opening hardly more combustion air is sucked through the stove even without the complete steel sheet, while the CO concentration still drops. Figure 3.6 has indicated already that the damper opening hardly effects the efficiency. So the conclusion can be drawn, that firing the stove with a damper position above 50% diminishes the CO concentration without loss of efficiency.

The minimum heat output of 2 kW, however, can only be reached by reducing the draft as mentioned in section 3.2.4.3 in combination with a fully closed combustion air damper.

3.2.6 <u>Heat balance of the "C" Nouna Stove</u>

For the experiments that have been carried out, no complete heat balance can be made because the wall temperatures and unburnt hydrocarbons have not been measured.
Nevertheless, the data of the flue gas composition present the opportunity to calculate the approximate sensible heat losses and unburnt constituents based on the carbon monoxyde production. For a few experiments these calculations have been carried out. They are summarized in table 3.7.

Although the data for the approximate heat balance are obtained from experiments with a non-stationary stove condition, they provide an indication of the difference between the original and "C" Nouna Stove.

The results for both the cold and the hot "C" Nouna Stove point out about 15 percentage points less sensible heat loss compared to the original Nouna Stove. As the loss due to unburned constituents does not change remarkably and the efficiency increases with about 15%, this gain in the sensible heat losses must at least partly be transferred to the pans.
Whether the accumulated heat loss is less for the "C" than for the original Nouna Stove can not be concluded from this data. However, the difference between the data for the cold and the hot "C" Nouna Stove indicates, that this heat still is an important factor. Decreasing this factor may realise even higher efficiencies.

3.2.7 Determination of the minimum and maximum heat output

The maximum and minimum heat output were established by means of the charge time. In general the experiments have been carried out with four pieces of wood.
The charge time for the maximum heat output is considered to be the shortest possible time between two charges, at which no building up of the fuel bed is observed.
The charge time for the minimum heat output is considered to be the largest possible time between two charges at which the new supplied charge of wood still can be ignited without an external pilot flame.

3.3 Conclusions

From the experiments with the modified Nouna "C" Wood Stove the following conclusions can be drawn.

* A baffle construction improves the cooking efficiency with about 7 percentage points. Increasing the pan diameter of the second pot gives another improvement of the efficiency with about 8%. Compared with the original Nouna Wood Stove the final improvement of the efficiency is 80% in the relative sense, which is 15% absolute.

* The efficiency varies between 30% and 40%.

* The stove can be operated with a minimum heat output of 2 kW and a maximum heat output of 9 kW.

* The minimum heat output of 2 kW can only be reached with a closed combustion air damper and a strongly reduced draft.

* The heat output of the fire hardly affects the efficiency.

* The combustion air damper position, taken between 12.5% and fully open air intake (steel sheet removed), has no influence on the efficiency.

* The initial condition of the stove, in the sense of hot or cold, affects the efficiency in the following way; at a heat output of 5 kW the efficiency of the hot stove is about 6 percentage points higher than for the cold stove.

* The concentration of carbon monoxide rises with a factor four by increasing the heat output from 4 to 9 kW.
  Operating the stove at a damper position above 50%, has a favourable effect on the CO-concentration. Because in both cases the efficiency does not drop remarkably the stove should by preference be fired at a low heat output and an open air intake.

* The CO-concentration varies between 03% and 1.1%. Although these percentage are still rather high the stove can compete with the existing domestic heating stoves.

* The stove gives a high users comfort in the sense of easy wood ignition, little smoke production and a boiling time for the first pan of 20 minutes.

* The approximate heat balance shows that the sensible heat loss is reduced from about 30% for the original stove to about 15% for the "C" stove. It can be excepted that this heat is absorbed by the pans.

* For existing heavy stoves the application of a baffle can be recommended for improving their efficiency.

3.4 References

K. Krishna Prasad (1981)
Some studies on open fires, shielded fires and heavy stoves.
Report from the Woodburning Stove Group.

J. Claus, W.F. Sulilatu (1982)
A comparison of the performance of Three Wood stoves.
To be published in: Proceedings of The Indian Academy of Sciences.

Table 3.1 Efficiency of the Nouna wood stove as a function of the heat output of the fire and the damper position for various baffle constructions.

size of the wood pieces: 0.02 x 0.03 x 0.2 m
depth of pans in stove : 0.11 m
initial amount of water: first pan 4.0 kg
second pan 2.0 kg

Symbols:

d.p. - combustion air damper position [%]
Δm_f - mass of charge [kg]
Δt - time between the two charges [min]
$\dot{Q}$ - heat output of fire [kW]
n_i - number of charges [1]
m_f - total amount of wood used [kg]
t_t - total burning time [min]
T_i - initial temperature of the water [°C]

t_b - time to boiling : t_{b1} - of pan 1 [min]
t_{b2} - of pan 2 [min]
m_s - amount of water evaporated: m_{s1} - of pan 1 [kg]
m_{s2} - of pan 2 [kg]
η - efficiency : η_1 - of pan 1 [%]
η_2 - of pan 2 [%]
η_t - total [%]

Type and run no.	d.p. [%]	m_f [kg]	Δt [min]	$\dot{Q}$ [kW]	n_i [1]	Δm_f [kg]	t_t [min]	T_i [°C]	t_b [min] t_{b1}	t_{b2}	m_s [kg] m_{s1}	m_{s2}	η [%] η_1	η_2	η_t
Original															
1 42	25	.912	8	8.9	4	0.228	50	22	24	[97°]	0.4509	-	13.6	3.7	17.3
2 37 [1]	"	2.329	"	9.1	10	0.233	108	20	26	44	2.362	0.436	15.4	3.9	19.3
A															
3	25	1.3033	"	8.5	6	0.2172	68	18	21	33.5	1.1464	0.2102	16.22	4.75	21.0
4	"	0.9087	"	8.9	4	0.227	48	23	18	27	0.7096	0.1026	16.97	5.14	22.1
B															
5	* open	0.852	"	8.3	4	0.2125	50	21	18	25.5	0.864	0.1315	20.5	6	26.5
6	25	0.8406	"	8.2	4	0.210	51	16	22	26	0.575	0.1476	17.2	6.6	23.8
7	25	0.898	15	4.7	4	0.224	49	18	24	38	0.8696	0.0877	19.6	5.31	24.9

* Open air intake. Steel sheet removed.

Table 3.1 (continued)

initial amount of water: first pan 4.0 kg
second pan 4.0 kg

Type and run no.	d.p. [%]	m_f [kg]	Δt [min]	$\dot{Q}$ [kW]	n_i [1]	Δm_f [kg]	t_t [min]	T_i [°C]	t_b [min]		m_s [kg]		η [%]		
									t_{b1}	t_{b2}	m_{s1}	m_{s2}	η_1	η_2	η_t
C															
8	* open	0.6892	10	7.2	3	0.230	45	20	22	27	0.4841	0.0944	19	12	31
9	100	0.6818	10	7.1	3	0.227	55	20/14	19	28	0.6248	0.1293	21.5	13.6	35.1
10	25	0.6706	10	7	3	0.223	48	20	15	26	0.8044	(100°)	25.11	10.65	35.8
11	12.5	0.7049	10	7.3	3	0.235	58	18/12	18.5	[95°]	0.8252	-	24.5	10.5	35
12	* open	1.0023	10	7.8.	4	0.2505	-	20/17	19	27.5	0.9311	0.4748	18.32	13.1	31.4
13	100	0.6048	10	6.3	3	0.2016	38	25/24	22	26	0.3324	0.0991	17.7	13.2	31
14	50	0.6651	10	6.92	3	0.222	50	22	17	27	0.588	0.115	21.2	12.55	34
15	* open	0.6597	10	6.86	3	0.220	49	25	18.5	23.5	0.5787	0.2074	20.72	13.94	34.7
16	25	0.6581	15	4.56	3	0.219	55	13/19	24	37	0.6209	0.0743	23	12.35	35.3
17	25	0.873	8	8.5	4	0.218	-	14	24	30	0.7326	0.2312	19	12	31
18	25	0.9087	8	8.9	4	0.227	54	19/16	16	28	1.3092	0.1993	24	10.5	35.5
19	25	0.6977	12	6	3	0.232	45	13	19.5	34	0.9109	0.0853	26.9	12.61	39.5
26	25	0.965	8	9.4	4	0.241	60	20/19	21	26	0.899	0.3084	18.63	11.35	30
27	25	0.6745	15	4.7	3	0.225	57	21	23	34	0.6913	0.2043	22.81	14.11	37
C															
20	25	0.7071	12	6.13	3	0.2357	72	23/17	19	29	0.949	0.3653	25.9	16.71	42.6

* Open air intake. Steel sheet removed.

Table 3.1 (continued)

initial amount of water: first pan 4.0 kg
second pan 4.0 kg

Type and run no.	d.p. [%]	m_f [kg]	Δt [min]	$\dot{Q}$ [kW]	n_i [1]	Δm_f [kg]	t_t [min]	T_i [°C]	t_b [min]		m_s [kg]		η [%]		
									t_{b1}	t_{b2}	m_{s1}	m_{s2}	η_1	η_2	η_t
C															
21	* open	0.6941	10	7.2	3	0.231	47	18	24	25.5	0.311	0.245	16	15	31
22	closed	0.677	10	7	3	0.225	55	17/20	22	22.5	0.535	0.285	20.5	15.6	36.1
								4 pieces							
23	closed	0.4604	24	3	2	0.2302	42	22	21	[70°]	0.5392	-	29.24	9.31	38.5
								4 pieces							
13	100	0.6048	10	6.3	3	0.2016	38	25/24	22	26	0.3324	0.0991	17.7	13.2	31.2
								2 pieces							
13[1]	closed	0.2142	15	2.2	2	0.1071	44	-	-	-	0.8856	0.1598	31	3.4	34.4
								4 pieces							
26	25	0.965	8	9.4	4	0.241	60	20/19	21	26	0.899	0.3084	18.6	11.35	30
								2 pieces							
26[1]	closed	0.338	15	2.3	3	0.112	60	-	-	-	1.7288	0.5527	29.6	8.71	38.3

* Open air intake. Steel sheet removed.

Table 3.2 Combustion performance of the modificated Nouna Wood stove as a function of the heat output of the fire and the combustion air damper position for various baffle constructions.

Symbols:
d.p. - combustion air damper position [% open]
n_i - number of charges [1]
$\dot{Q}$ - heat output of the fire [kW]
T_g - flue gas temperature [°C]

initial amount of water
A/B pan 1 : 4 kg; pan 2: 2 kg
C pan 1 : 4 kg; pan 2: 4 kg

Run no.	d.p. [%]	n_i [1]	$\dot{Q}$ [kW]	Flue gas composition CO_2 [%]	CO [%]	CO [g/kg]	O_2 [%]	T [°C]	Excess air factor [-]	Draft [Pa]
1	25	4	8.9	6.73	0.15	26	13.9	313	3	4.2
2	25	4	9.1	6.80	0.24	40.4	13.98	276	3	"
A										
3	25	4	8.5	9.11	0.31	39	11.05	362	2.21	4.2
4	25	4	8.9	8.54	0.27	36.2	12	297	2.4	"
B										
5	* open	4	8.3	8.12	0.37	51.5	12.9	231	2.45	4.2
6	25	4	8.2	9.01	0.59	72.6	12.3	107	2.2	"
7	25	4	4.7	5.5	0.25	51.3	16	141	3.61	"
C										
8	* open	3	7.2	7.18	0.31	48.9	13.94	113	2.8	3.4
9	100	3	7.1	8.8	0.46	58.7	11.84	119	2.25	"
10	25	3	7	8.7	0.68	85.6	11.95	139	2.22	"
11	12.5	3	7.3	9.5	1.06	118.6	11.21	80	1.97	"
12	* open	4	7.8	-	-	-	-	-	-	"
13	100	3	6.3	5.81	0.28	54.3	15.85	129.7	3.41	"
14	50	3	6.92	6.8	0.19	32.1	14.72	156	2.92	"
15	* open	3	6.86	6.4	0.22	39	15.26	177	3.15	"

* Open air intake. Steel sheet removed.

Table 3.2 (continued)

Run no.	d.p.	n_i	$\dot{Q}$	Flue gas composition					Excess air factor	Draft
				CO_2	CO	CO	O_2	T		
	[%]	[1]	[kW]	[%]	[%]	[g/kg]	[%]	[°C]	[-]	[Pa]
C										
16	25	3	4.56	6	0.32	60	15.3	126	3.3	3.4
17	25	4	8.5	-	-	-	-	122	-	"
18	25	4	8.9	12.24	1.5	129	7.86	108	1.51	"
19	25	3	6	7.77	0.61	86	12.76	137	2.5	"
26	25	4	9.4	9.7	1.11	121	11.51	156	1.92	"
27	25	3	4.7	6.85	0.29	48	14.57	158	2.91	"
20	25	3	6.13	9	0.51	63.6	11.35	187	2.2	1
21	open	3	7.2	6.4	0.19	34	15	195	3.15	4
22	closed	3	7	8	0.34	50	13.6	200	2.5	4
23	closed	2	3	7.5	0.57	83	13.13	83	2.6	1.5
13	100	3	6.3	5.81	0.28	54	15.85	129.7	3.4	3.4
13[1]	closed	2	2.2	-	-	-	-	-	-	1
26	25	4	9.4	9.7	1.11	121	11.51	156	1.92	3.4
26[1]	closed	3	2.3	6.25	0.45	79	14.98	125	3.1	1

Table 3.3 Effect of introduced baffles in the Nouna wood stove on efficiency of the stove.

damper position: 25%
heat output : 8.5 kW

Type of stove	Efficiency			Improved efficiency in the relative sense		
	η_1 [%]	η_2 [%]	η_{total} [%]	η_1 [%]	η_2 [%]	η_{total} [%]
original design	14.5	3.8	18.3	-	-	-
A	16.6	4.9	21.5	14.5	29	18
B	18.9	6.6	25.5	31	74	40
C	22.1	10.8	32.9	53	184	80

Table 3.4 Effect of lowering the heat output on efficiency. Exp. 13 and 26.

Exp. No.	pieces [-]	charge time [min]	Q [kW]	draft [Pa]	Efficiency [%]			Evaporated water [gr/min]	
					η_1	η_2	η_{total}	pan 1	pan 2
13	4	10	6.3	3.4	18	13.2	31.2	21	12.6
13'	2	15	2	1	31.12	3.4	34.5	6	1.3
26	4	8	9.4	3.4	18.6	11.35	30	23	9
26'	2	15	2.3	1	29.6	8.7	38.3	14	4

Table 3.5 Influence of the initial condition of the stove on the efficiency

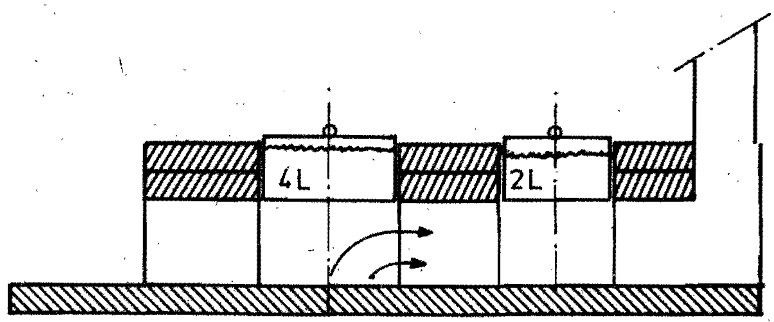
exp.	damper position [%]	Δt [min]	n_i [l]	$\dot{Q}$ [kW]	t_{b1} [min]	t_{b2} [min]	η_1 [%]	η_2 [%]	η_{tot} [%]	condition of the stove
28	25	8	4	9.4	21	26	18.6	11.4	30.0	cold
28'	25	15	3	4.7	23	34	22.8	14.0	36.8	cold
29	25	15	4	4.8	22	35	21.3	14.0	35.3	cold
29'	25	15	3	5.2	21	34	21.7	13.5	35.2	cold
30	25	15	4	5.2	20	21	22.8	18.8	41.6	hot
30'	25	15	4	5.4	20.5	20.5	22.2	18.8	41.0	hot

Table 3.6 Chimney losses of the "C" Nouna Stove for various power levels.

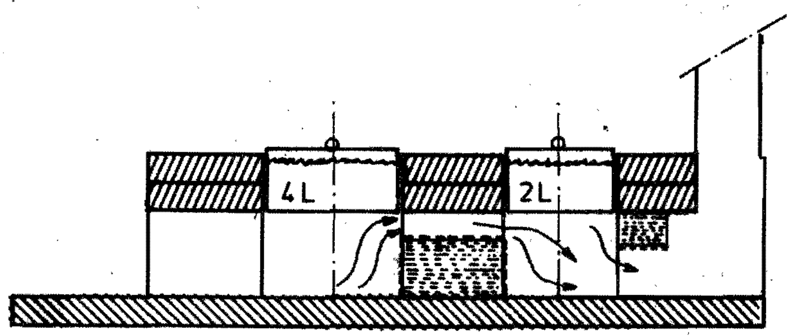
Run no.	Heat output [kW]	Latent heat loss [kJ/kg]	Sensible heat loss [kJ/kg]	Total chimney loss [kJ/kg]
16	4.56	566	2071	2637
27	4.7	453	2377	2830
19	6.0	811	1732	2543
20	6.13	600	2175	2775
10	7.0	807	1564	2371
18	8.9	1216	787	2003
26	9.4	1141	1546	2687

Table 3.7 Approximate heat balance of the "C" Nouna wood stove

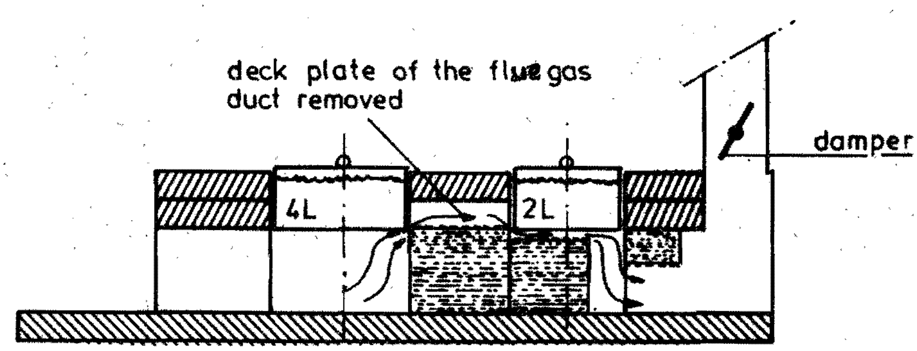
Run no.	Heat output [kW]	Total efficiency [%]	Sensible heat in flue gas [%] ±1%	Unburnt constituents CO [%] ±1%	Stove condition
26[1]	2.3	38.3	11	4	cold
16	4.6	35.4	12	3	cold
19	6	39.5	10	5	cold
10	7	35.8	9	5	cold
26	9.4	30	11	7	cold
30	5.2	41.6	18	4	hot
30'	5.4	41.0	17	5	hot



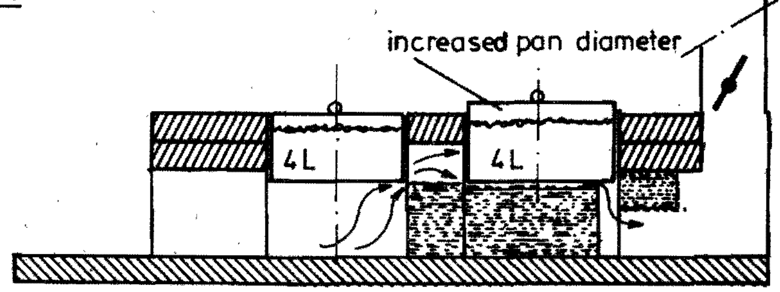
Normal situation



Modification A

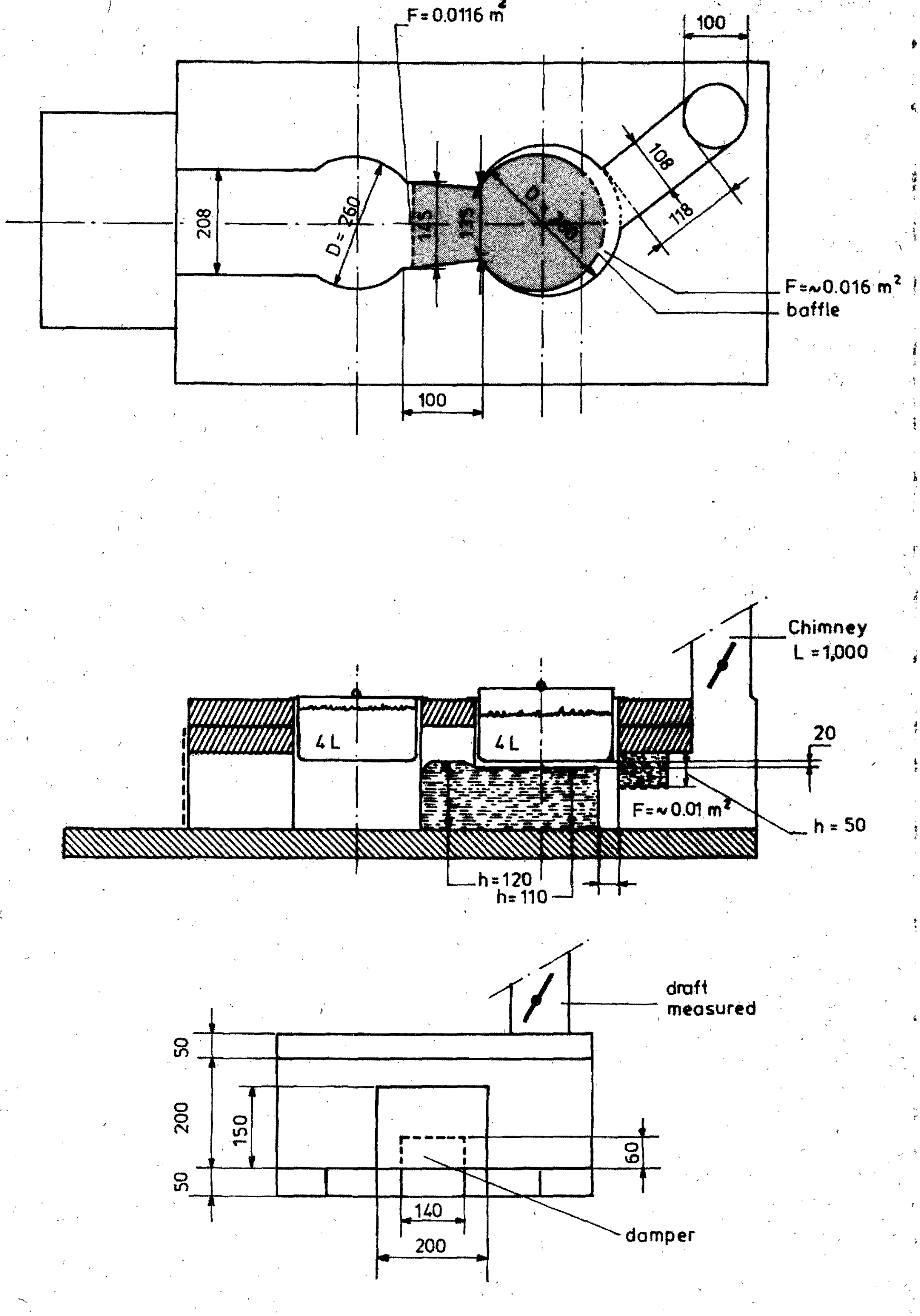


Modification B



Modification C

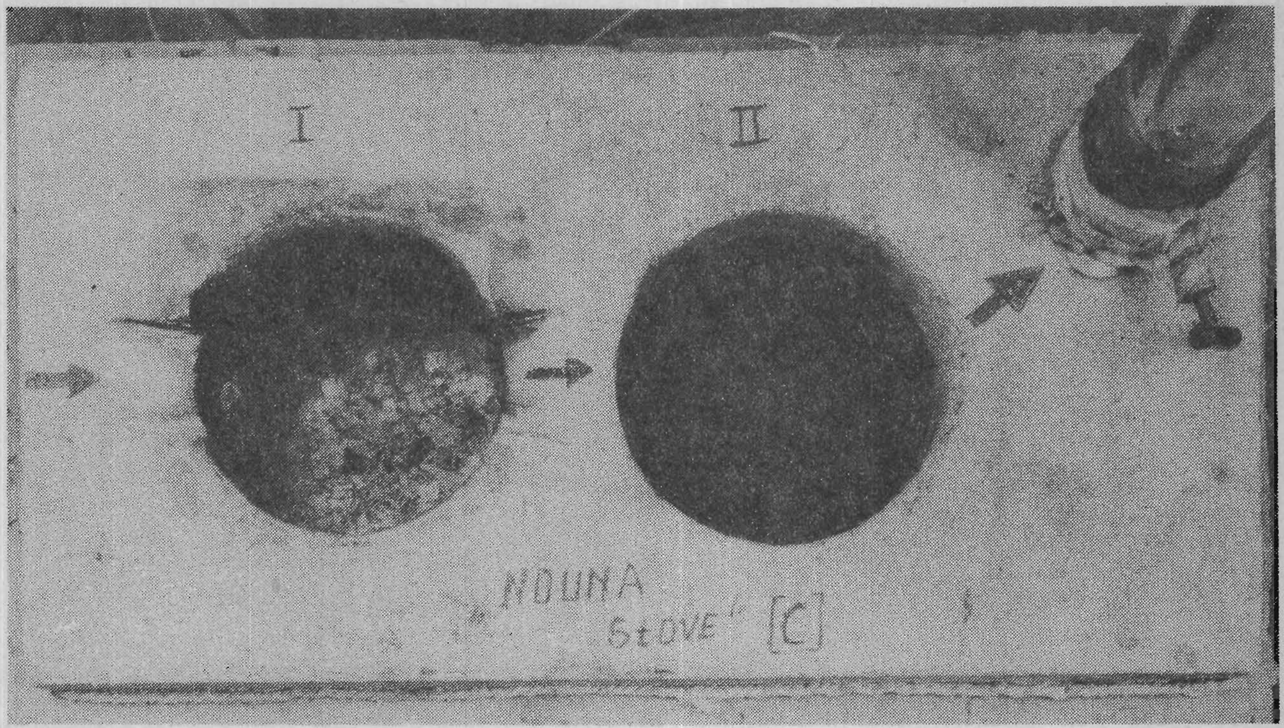
Construction of the original Nouna stove and the modifications A, B and C	MT_TNO 84940 Fig. 3.1



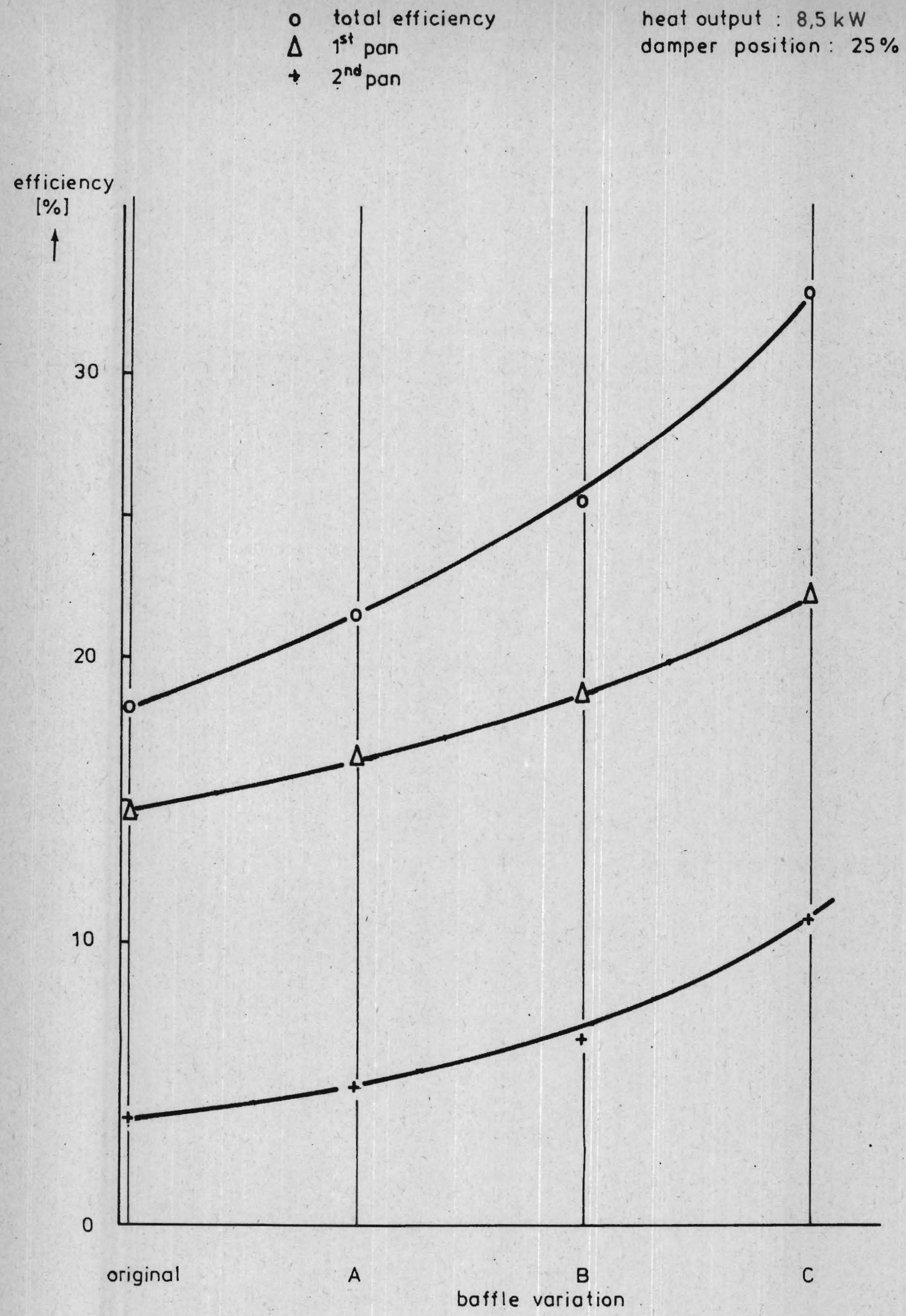
Schematic design of the Nouna wood stove

Modification "C" scale 1 : 10

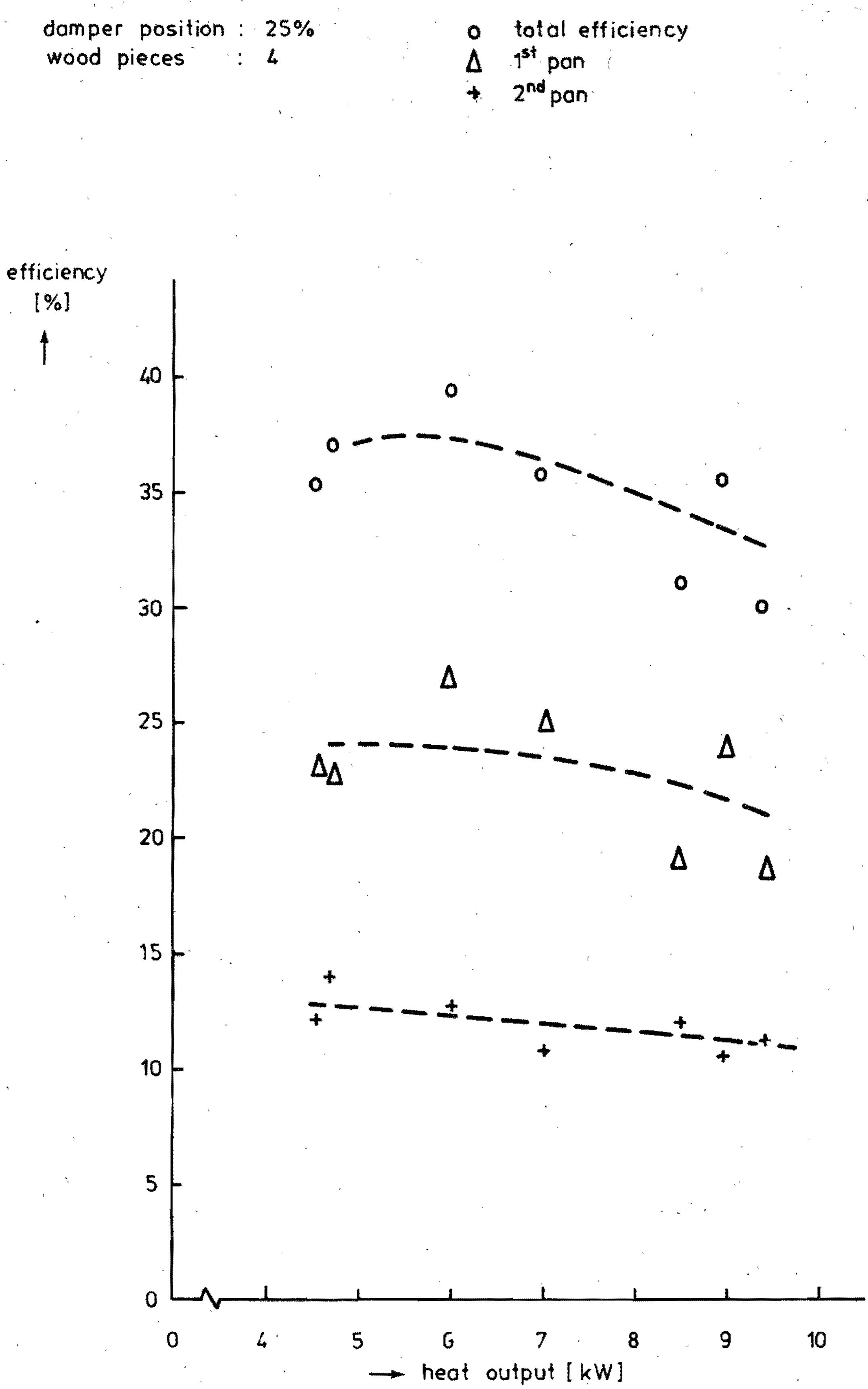
MT_TNO
84940
Fig. 3.2



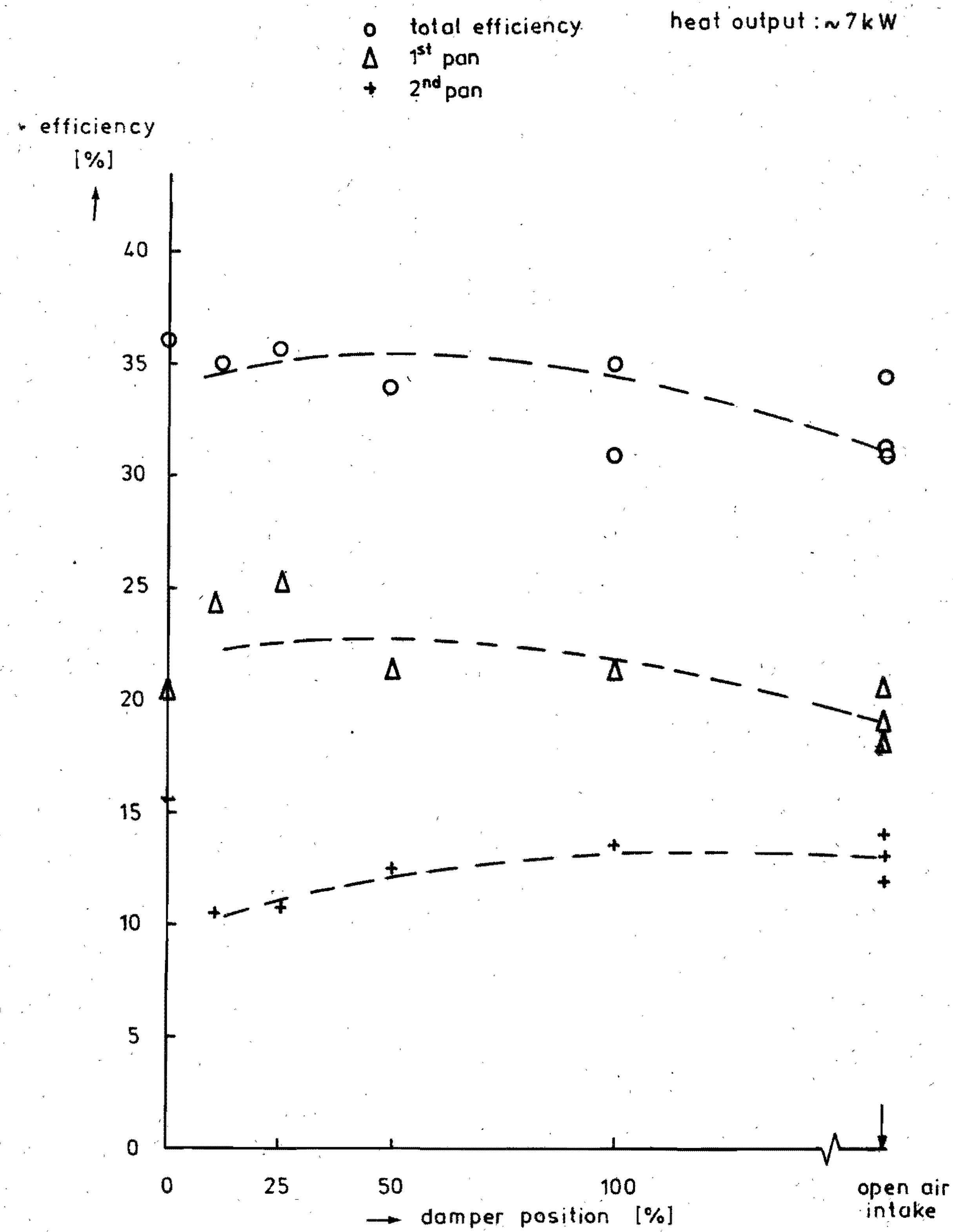
The Nouna wood stove Modification "C"	MT_TNO 84940 Fig. 3.3



Influence on efficiency by introduction of the baffle constructions in the Nouna wood stove	MT_TNO 84940 Fig. 3.4



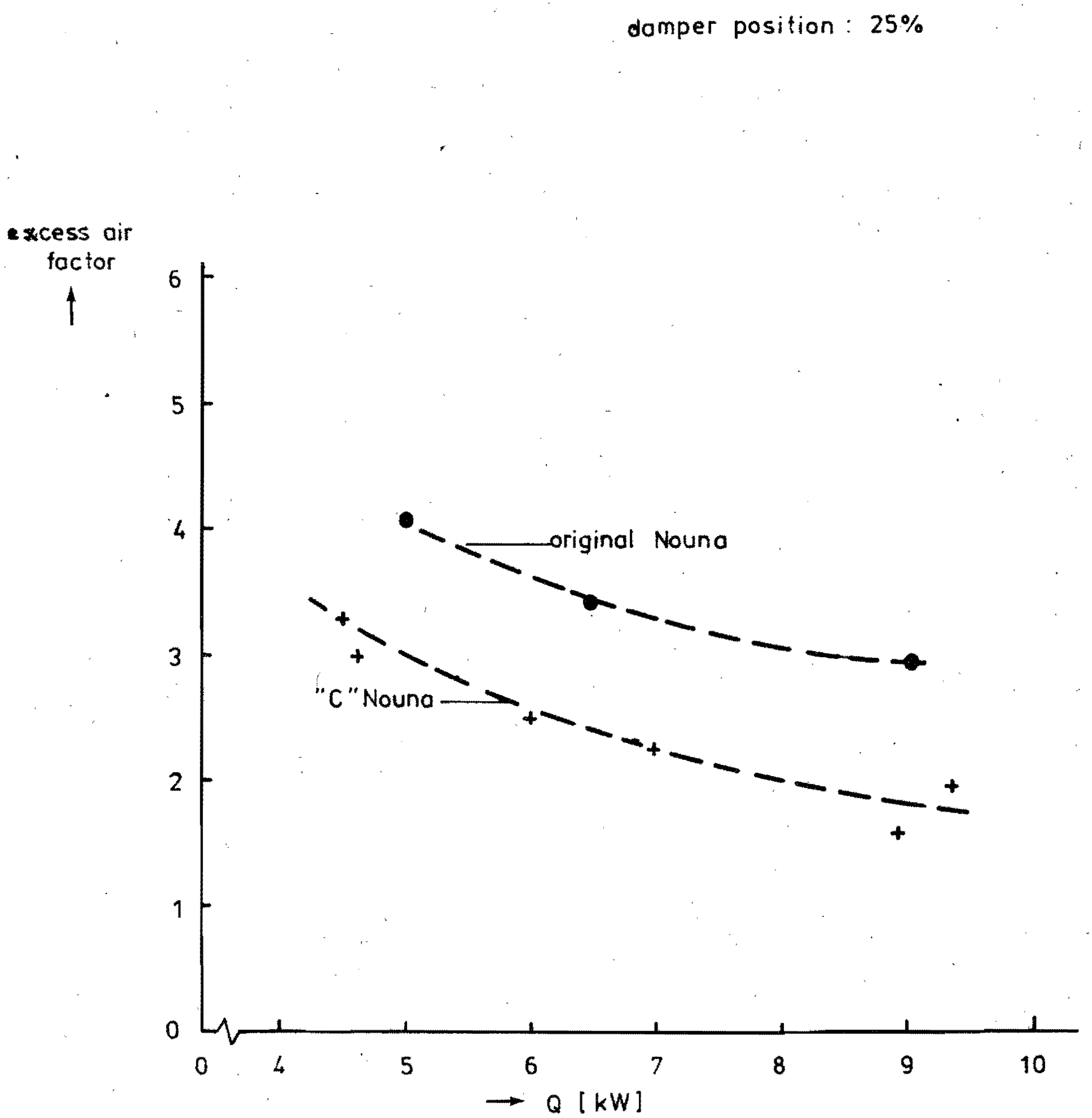
Efficiency as a function of the heat output of the modificated Nouna wood stove	MT_TNO 84940 Fig. 3·5



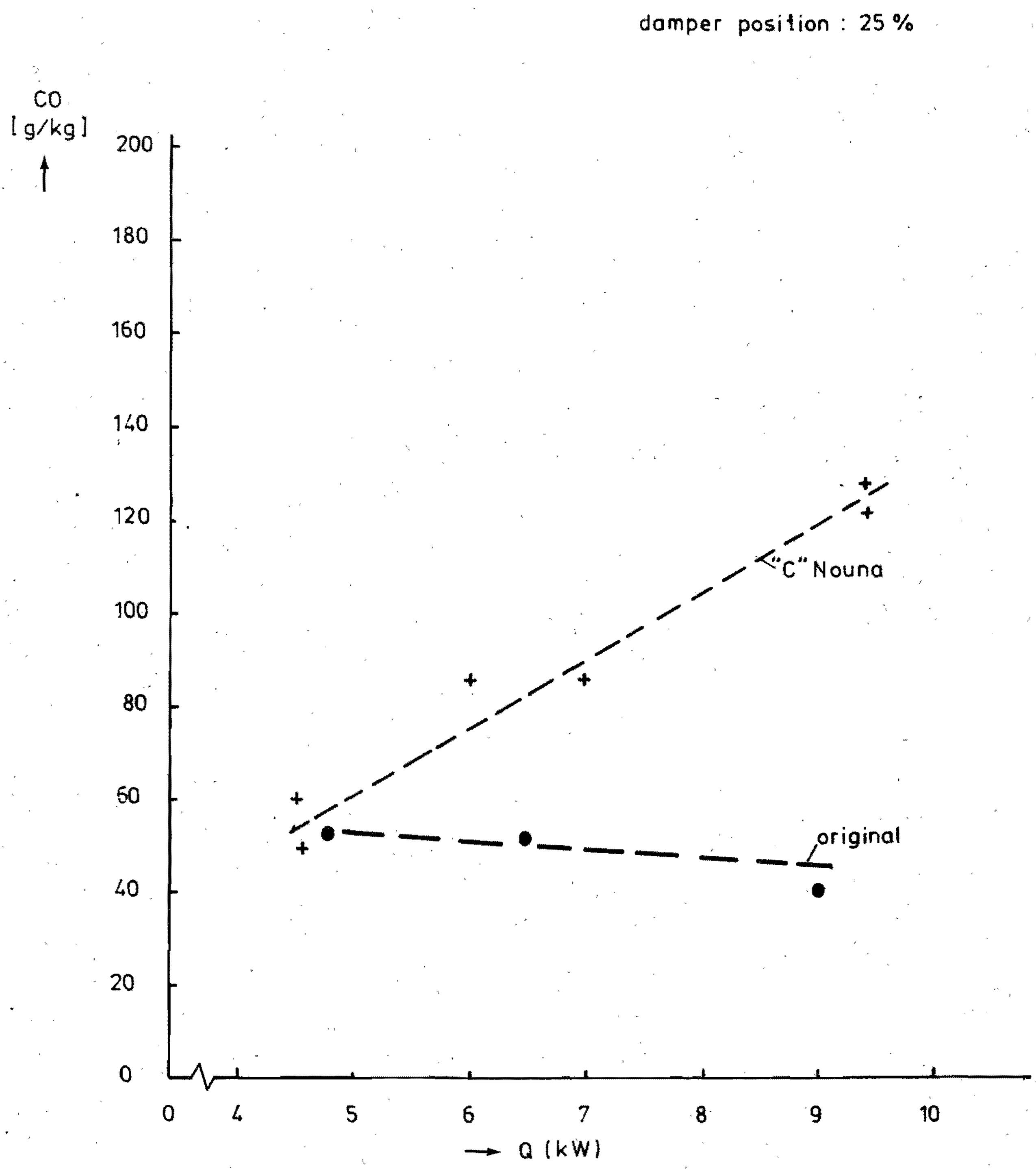
Efficiency as a function of the damper position of the Nouna wood stove (C)

MT_TNO 84940

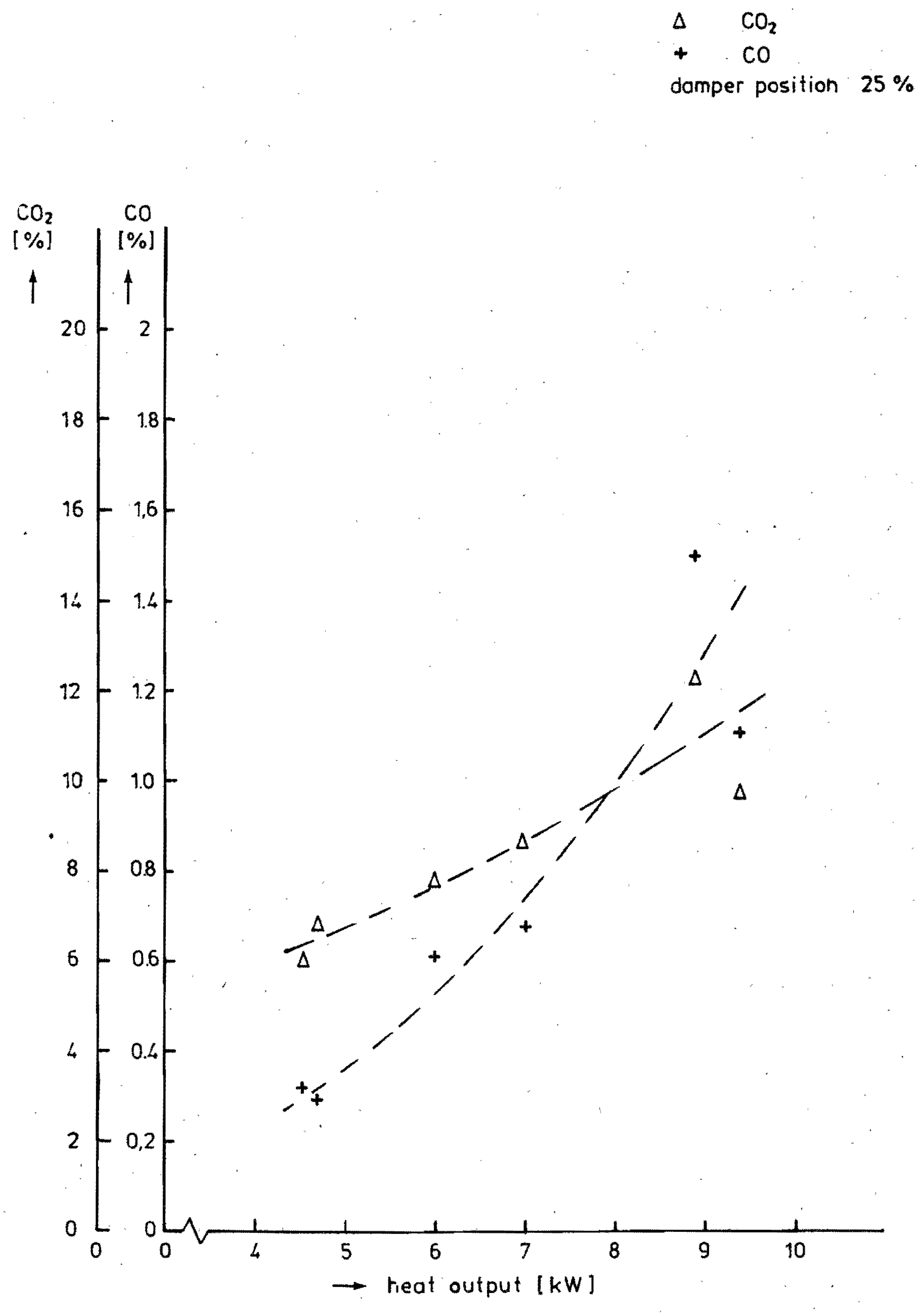
Fig. 3·6



Excess air factor as a function of the heat output of the fire. For the original Nouna and "C" Nouna stove	MT-TNO 84940 Fig. 3.7

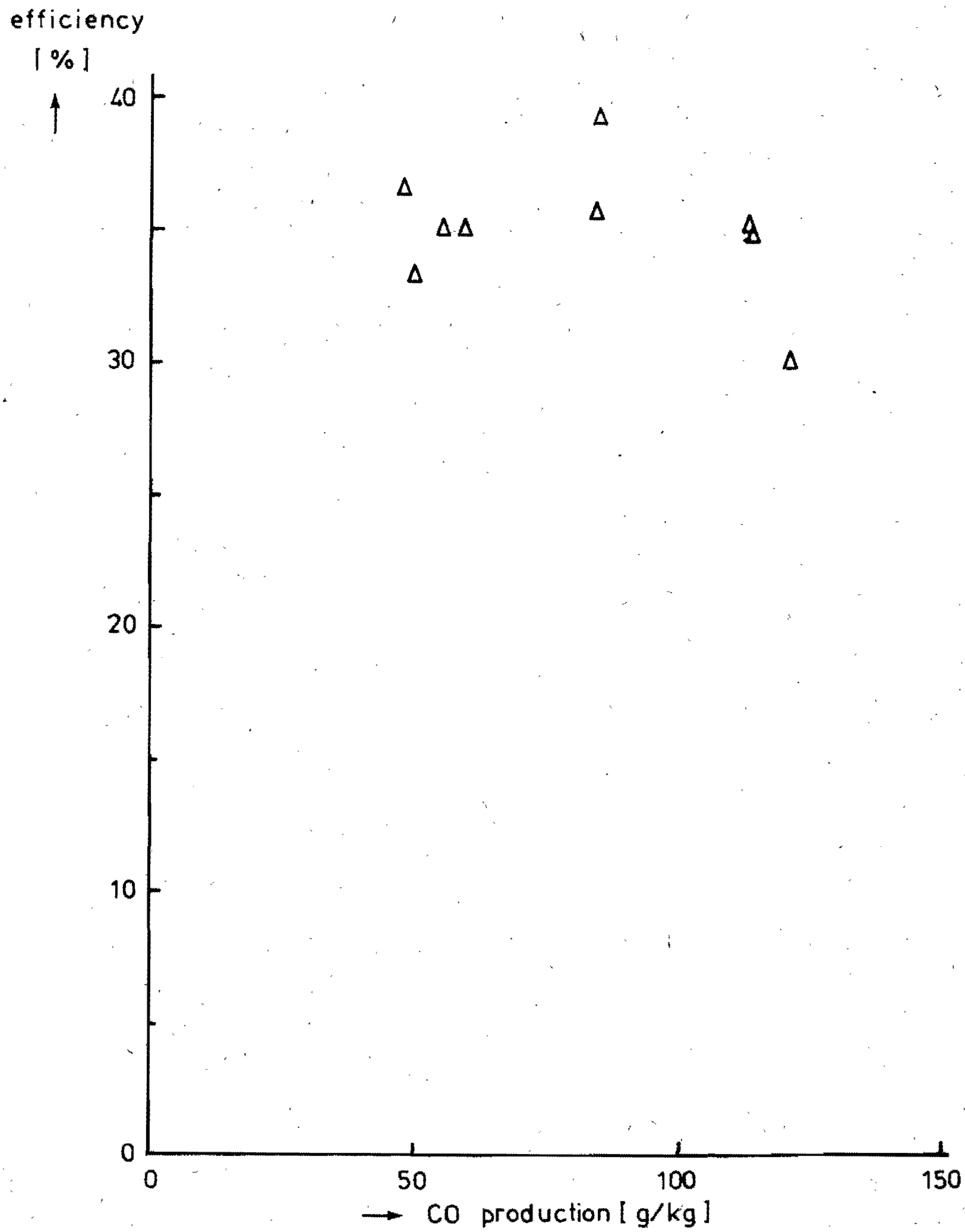


CO production in g/kg wood burnt as a function of the heat output of the fire for the original Nouna and "C" Nouna wood stove	MT-TNO 84940 Fig. 3.8

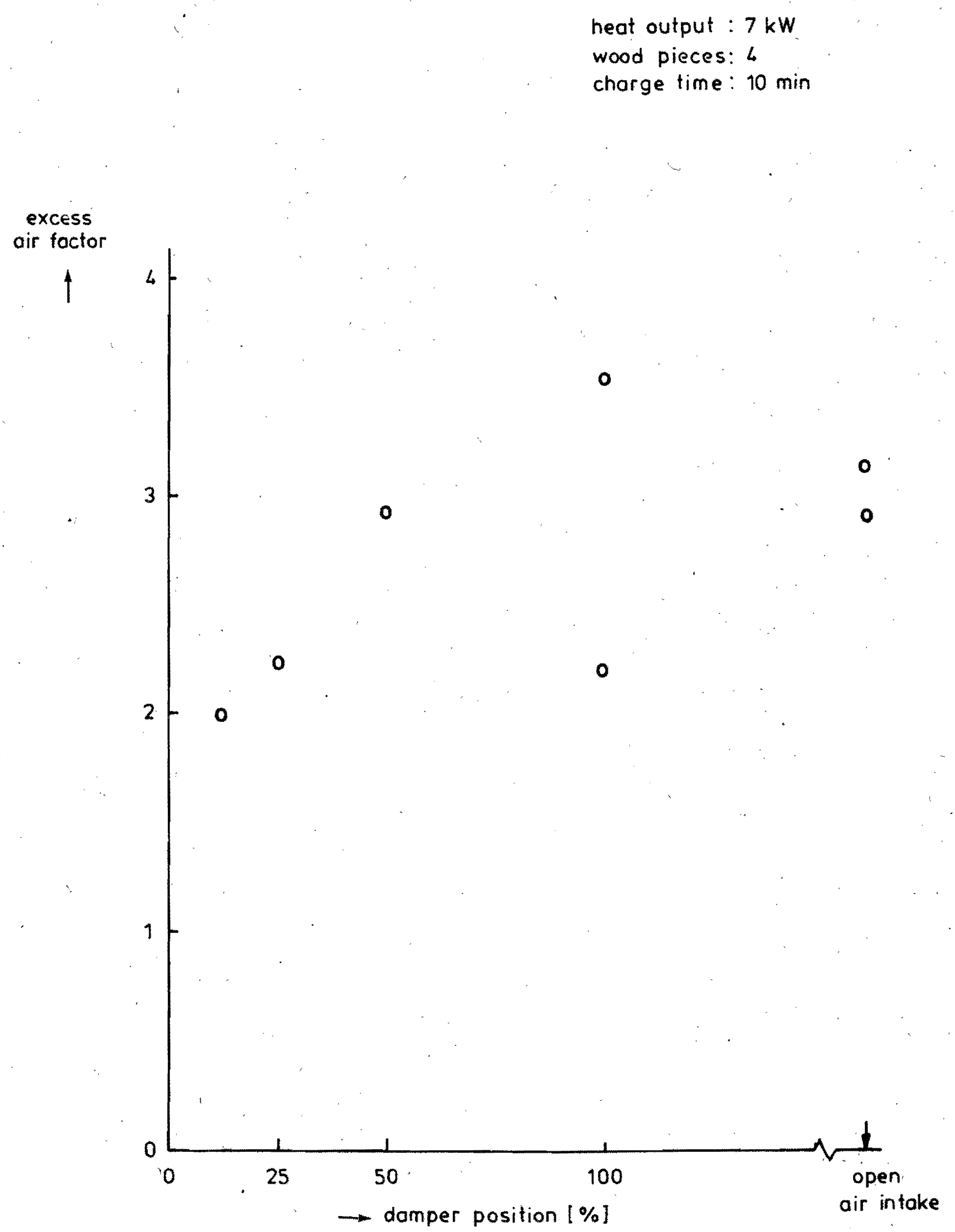


CO_2, CO concentration as a function of the heat output

MT-TNO
84940
Fig. 3.9

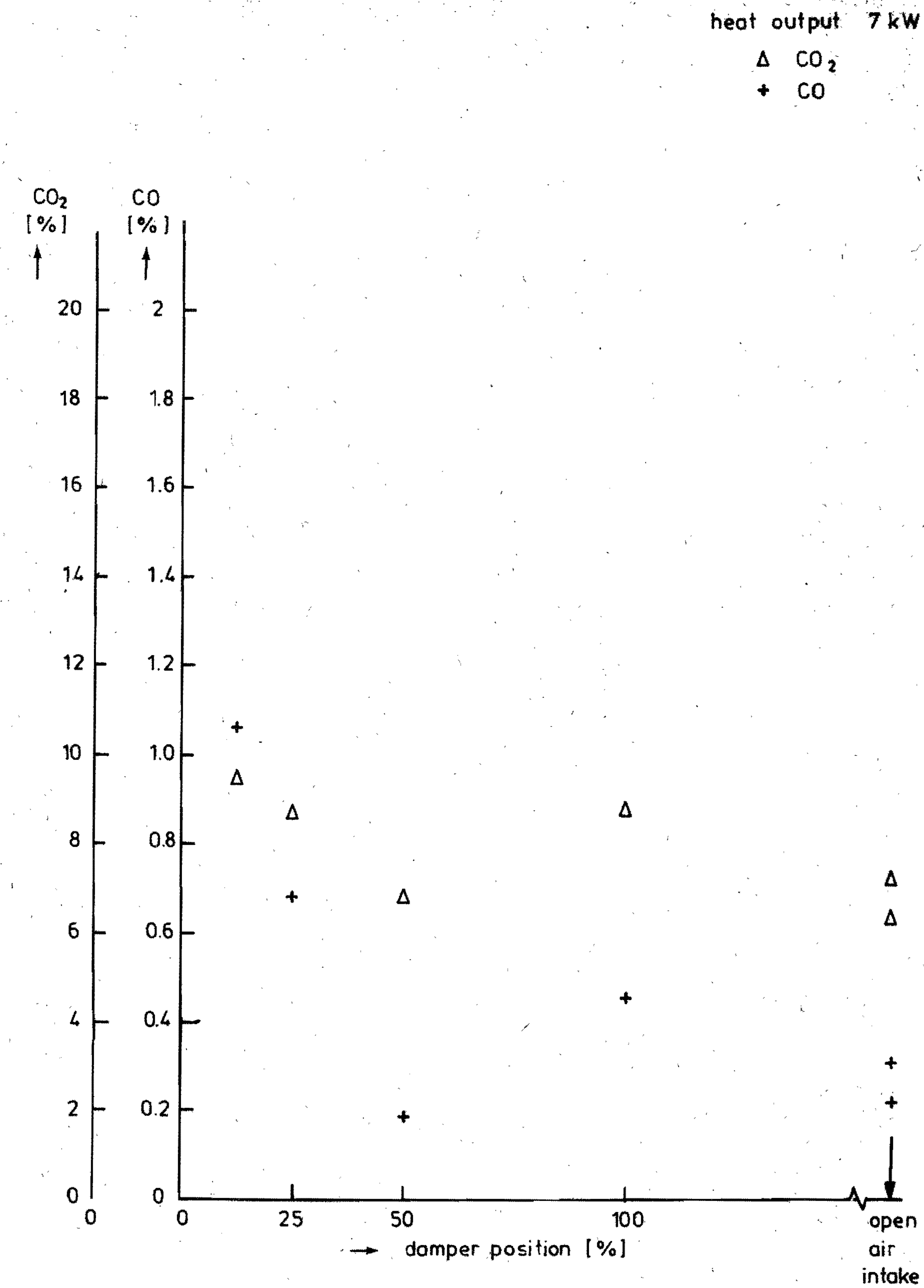


The efficiency as a function of the CO production for various heat outputs and a damper position of 25 %	MT-TNO 84940 Fig. 3.10



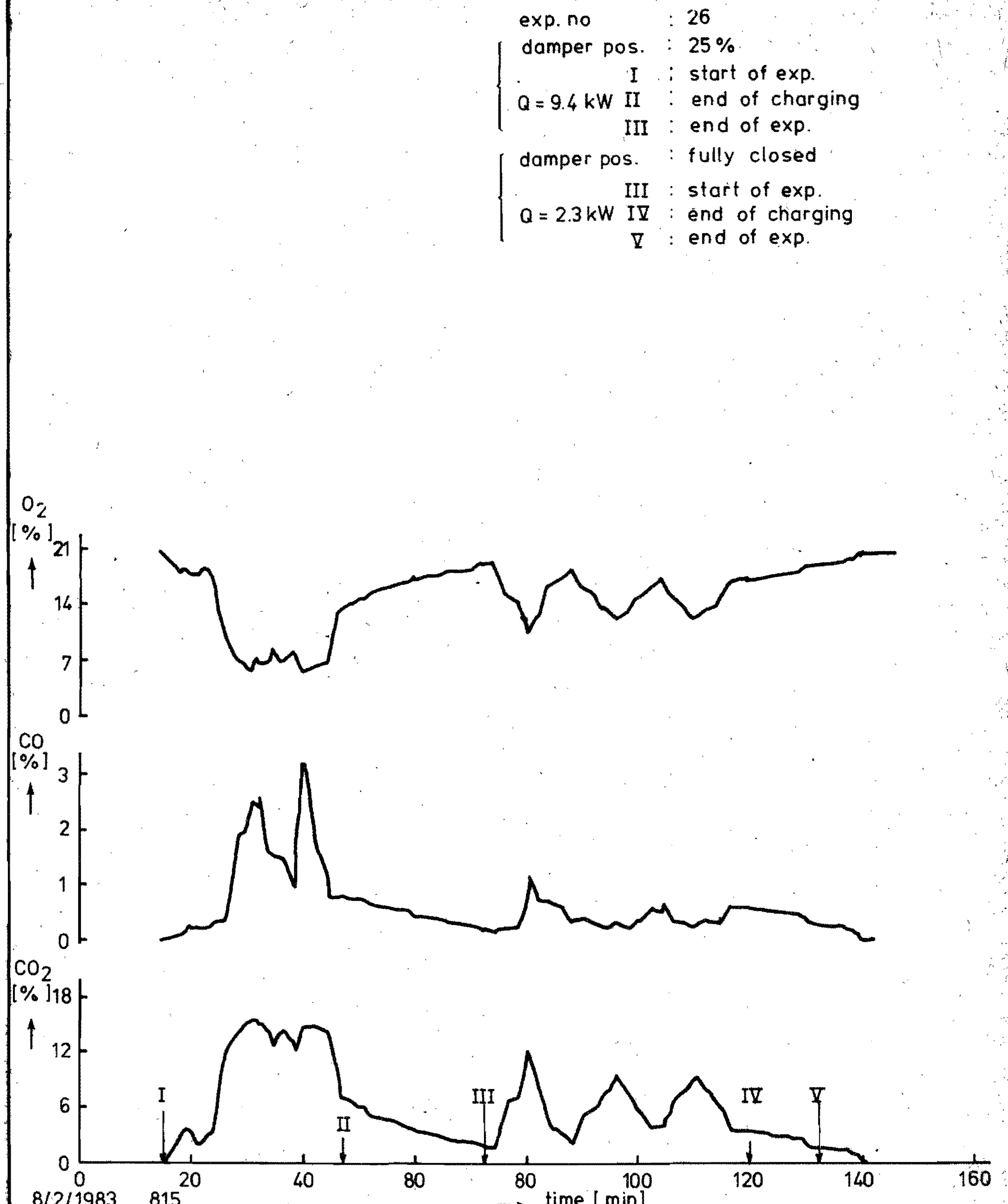
Excess air factor as a function of the damper position of the Nouna wood stove (C)

MT-TNO
84940
Fig. 3-11



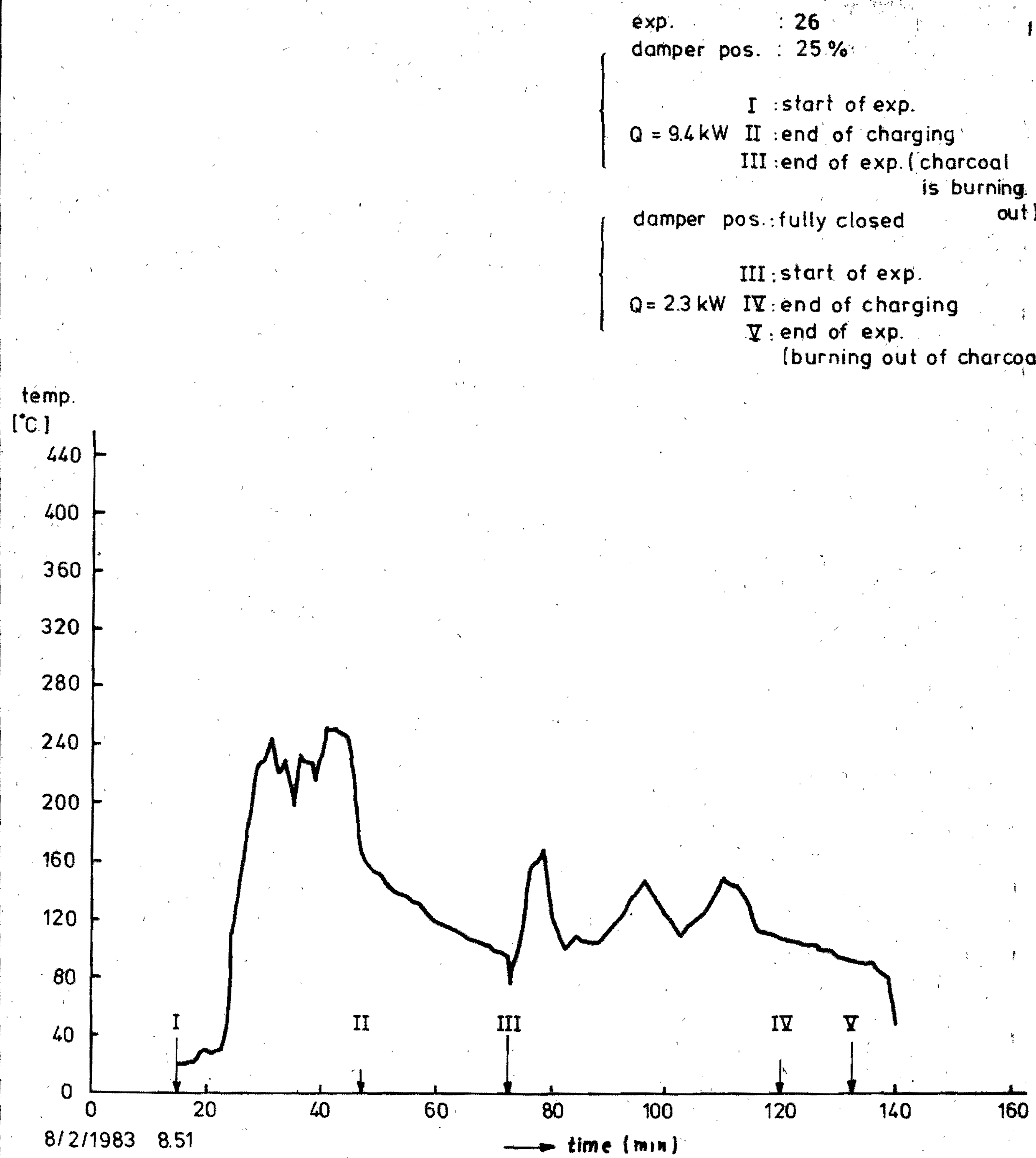
CO_2, CO concentration as a function of the damper position of the "C" Nouna wood stove

MT_TNO
84940
Fig. 3·12

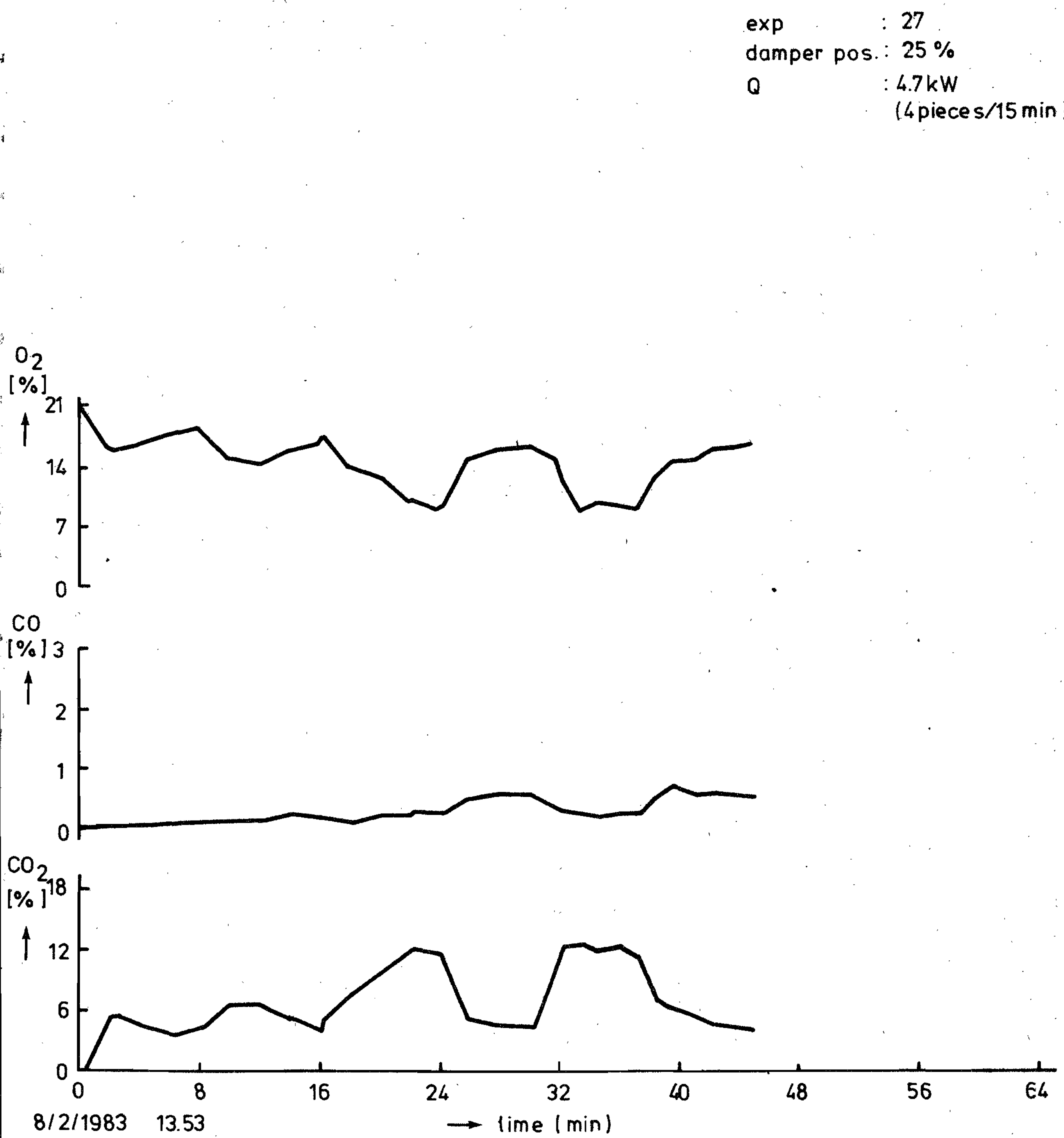


The flue gas concentration as a function of time

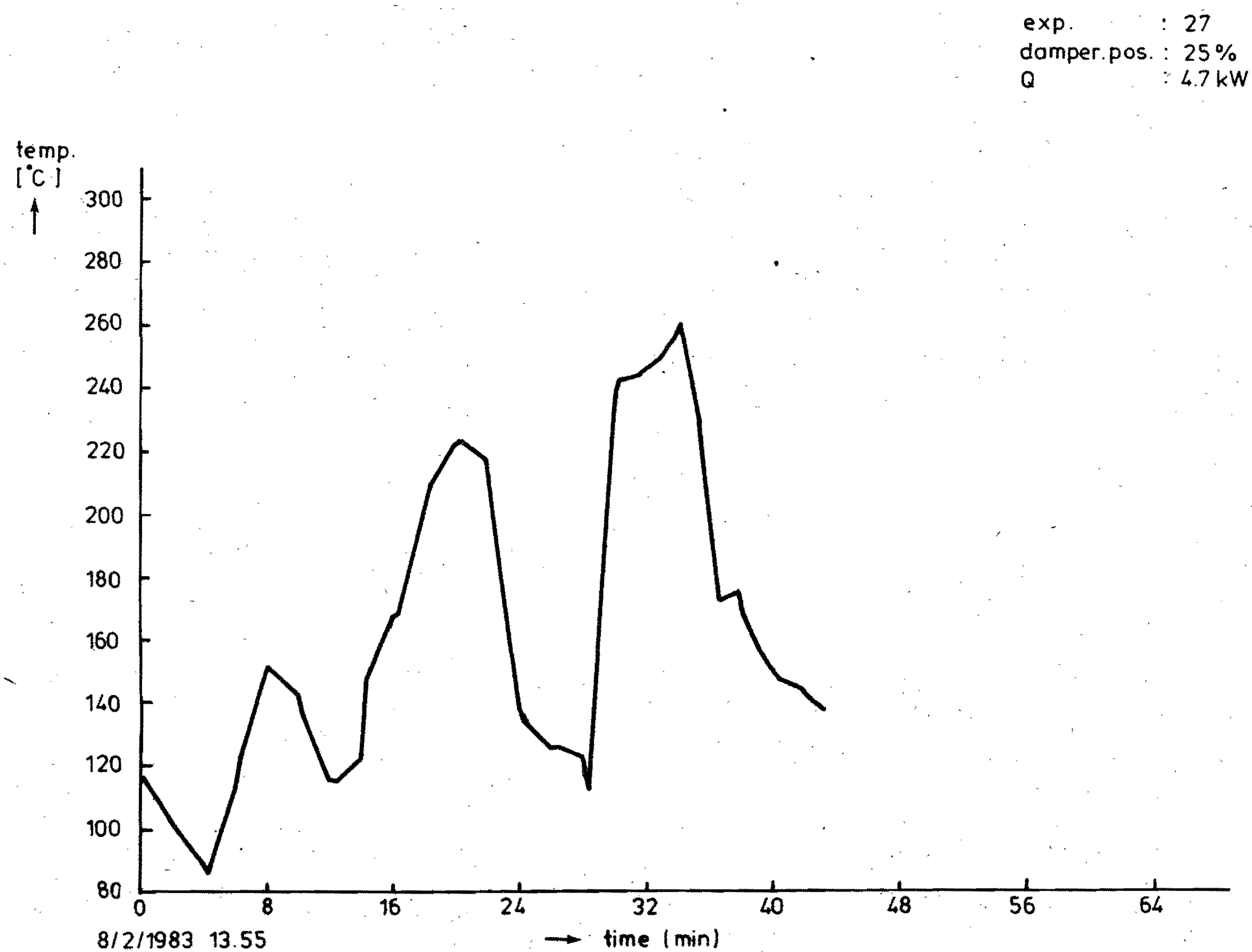
MT_TNO
84940
Fig. 3.13



The flue gas temperature as a function of time	MT_TNO 84940 Fig. 3-14



The flue gas concentration as a function of time	MT_TNO 84940 Fig. 3·15



The flue gas temperature as a function of time

MT-TNO
84940
Fig. 3-16

4. HEAT TRANSFER CHARACTERISTICS OF METAL, CERAMIC AND CLAY STOVES

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4.1 Introduction

The wood stove literature abounds with statements that generally go as follows:

(i) metal stoves, unlike clay stoves, lose a great deal of heat to the surroundings; the implication is that metal stoves show poor efficiencies due to this reason;

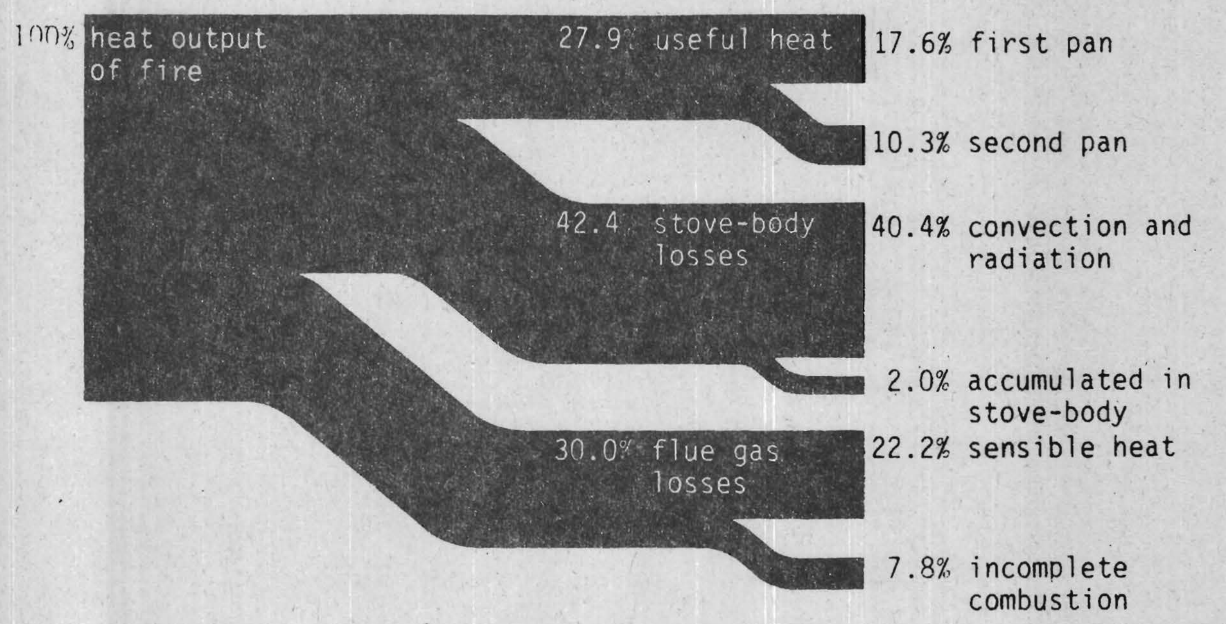
or more sophisticatedly,

(ii) metal stoves are better for short intermittent operation while clay stoves are better for long durations of operation.

These are really opinions based on intuition rather than on evidence in a strict scientific sense.

We first take a look at some experimental evidence (figure 4.1) obtained by TNO on a metal and brick stove (see Prasad 1982).
A superficial look at these heat balance diagrams indeed support the first part of the first statement. However the efficiency of the metal stove is very much higher than that of the brick stove. Thus it is clear that the second part of the first statement is not necessarily true. Moreover, a closer examination of the heat balances shows that there is nearly 12% of heat unaccounted in the estimations provided for the brick stove. Drawing heat balances for a wood stove in general is a complex task and it is more so in a heavy stove - be it made of clay or bricks. The difficulty in the heavy stoves arises due to the large number of temperatures that need to be measured. This is not easy to arrange and thus the estimates of heat accumulation in the stove body could be in error by significant margins.

A METAL STOVE (De Lepeleire/Van Daele stove)



A HEAVY STOVE (Nouna stove)

Fig. 4.1. Effect of construction material on heat balances in stoves

It is quite concervable that a substantial proportion of unaccounted heat could be due to the underestimate of the accumulated heat. From these, it is reasonable to suppose that the stove body losses are not very different for the two stoves. In particular it seems that as far as obtaining high efficiencies from cook stoves are concerned there is no evidence available to suggest that the material of construction is a serious limiting factor.

The present note presents a relatively simple analysis to provide a quantitative support to the neuristic discussion above. In addition, it is hoped that the analysis will indicate the directions for further experimental work, that will remove some of the restrictive assumptions made in the analysis.

4.2 The stove model

Figure 4.2 shows the prototype of a stove model considered in the study. It consists of a square cavity of side a. At the bottom of the cavity is the fuelbed and the height of the cavity above the fuelbed is h. The pan is located at the top of the cavity. The walls of the cavity and the pan receive heat from the fuelbed and the flames

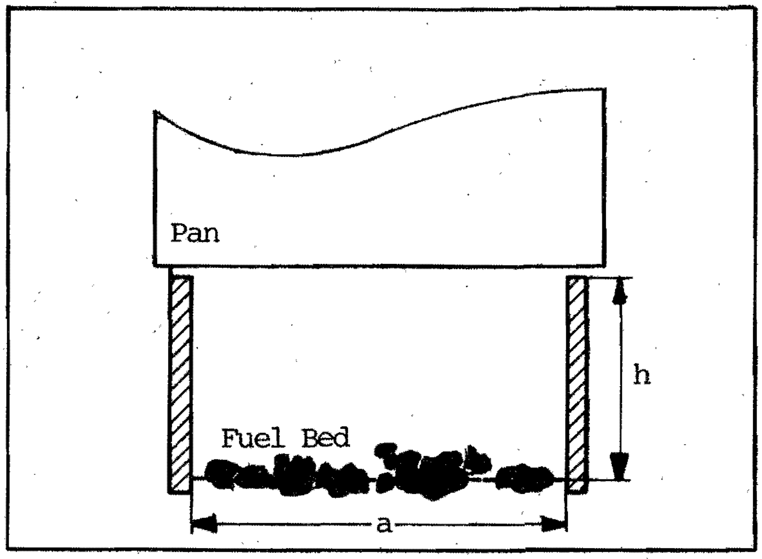


Figure 4.2: The stove model

on top of the fuelbed by radiation and convection. The purpose of the present excercise is to calculated the heat flow through the walls of the cavity as a function of time for a given power output of the fire.

4.3 The physical model

The modelling of heat transfer from a wood fire is too complex and demands a knowledge of the heat liberation from the fire as a function of time. This knowledge is in the process of being accumulated by the group. For the purposes of the present study we represent the fire by a constant heat source. The heat source itself is assumed to be symmetric in the cavity so that each wall of the cavity receives the same amount of heat. In addition we ignore the presence of corners and the effect of the finite height of cavity. With these assumptions the problem reduces to one of heat conduction in a slab. The physical situation is depicted in figure 4.3.

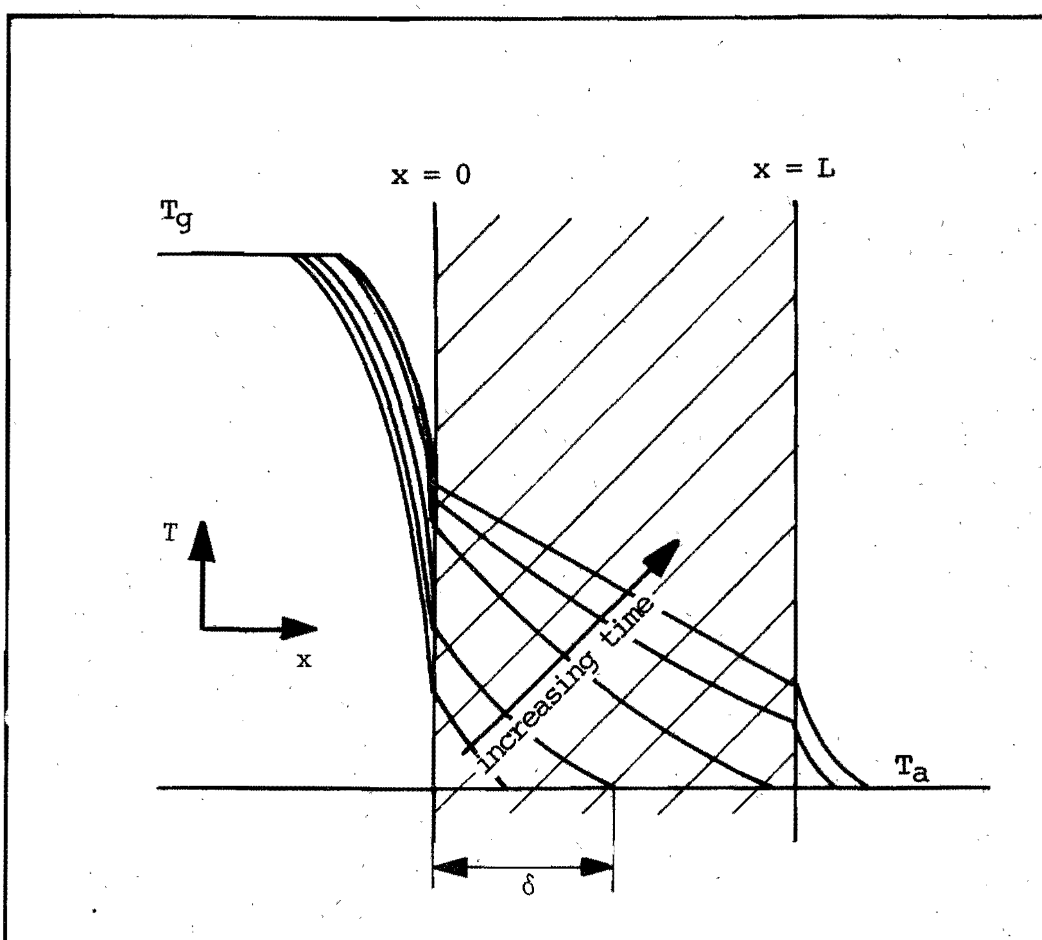


Figure 4.3: The physical model: heat conduction in a slab.

The face $x = 0$ is exposed to the fire suddenly at time $t = 0$ which according to our assumption is at a uniform temperature T_g thereafter. The face $x = L$ is exposed to the environment at temperature T_a. The temperature distribution in the slab varies with time and is depicted qualitatively in the figure. The analysis will result in the determination of heat flow through the wall at $x = 0$ as a function of time.

4.4 The mathematical model

The physical problem of the previous section can be described mathematically by the following equations.

$$\frac{\partial T}{\partial t} = \kappa \frac{\partial^2 T}{\partial x^2} \quad (t > 0,\ 0 < x < L) \tag{4.1}$$

$$T(x,\ 0) = T_a \tag{4.2}$$

$$-\lambda \left.\frac{\partial T}{\partial x}\right|_{x=0} = \alpha_g \{T_g - T(0,t)\} \tag{4.3}$$

$$-\lambda \left.\frac{\partial T}{\partial x}\right|_{x=L} = \alpha_a \{T(L,t) - T_a\} \tag{4.4}$$

where T = the temperature, K
t = the time, s
X = the distance, m
κ = the thermal diffusivity, m^2/s
λ = the thermal conductivity, W/mK
α = the heat transfer coefficient, W/m^2K
and the subscripts g corresponds to gas and a corresponds to air.

This is the classic diffusion problem. For constant α_g and α_a, its solution is given in terms of an infinite series of exponential functions. (see for example Carslaw & Jaeger) The series is rather slowly convergent and cumbersome to evaluate in practical situations where there is a need to investigate a large number of parameters. In addition the procedure is not easy to adopt to the real problem where α_g and α_a are functions of temperature, and the problem becomes nonlinear. While we restrict ourselves to the linear problem in this study, we adopt a more general but approximate procedure of solution.

4.5 The solution technique

We adopt a solution technique called the integral technique (see Özişik 1980 for details of the procedure). The technique divides the solution in two phases, which can be seen from figure 4.3. For small times a distance δ can be identified beyond which the slab does not notice the presence of the fire at x = 0. In other words only part of the slab experiences the temperature increase. This fact will be easily observed by anybody working with heavy stoves by simply touching its outer skin. It will not feel warm to the touch for considerable periods of time after a fire has been started. It is to be noted that δ is a function of time and the purpose of the solution technique is to determine δ. The second phase of the solution starts when δ = L and the temperature of the outer skin increases till a steady state is established, after which the temperatures do not vary with time.

The technique consists in integrating equation (4.1) over x from 0 to δ for the first phase. This results in

$$\frac{d}{dt} \left[\int_o^\delta T(x,t)\, dx - T_a\, \delta(t) \right] = \kappa \frac{\partial T}{\partial x} (0,t) \tag{4.5}$$

The function T(x,t) is approximated by an assumed profile. We use a parabolic approximation

$$T(x,t) = A + Bx + Cx^2 \tag{4.6}$$

The coefficients - which are functions of time - are evaluated by the following conditions

$$\text{At } x = 0, \; - \lambda \frac{\partial T}{\partial x} = \alpha_g \, [T_g - T(0,t)] \tag{4.7}$$

$$\text{At } x = \delta,\ T[\delta(t),t] = T_a \tag{4.8}$$

$$\left.\frac{\partial T}{\partial x}\right|_{x=\delta} = 0 \tag{4.9}$$

The resulting expression is substituted into equation (4.5). The indicated integration and differentiation are then carried. The procedure leads after some algebraic manipulation to the following differential equation

$$\frac{2\ \delta^* + \delta^{*2}}{1 + \delta^*} \quad \frac{d\delta^*}{dt^*} = 6 \tag{4.10}$$

where $\delta^* = \dfrac{\delta}{2\lambda/\alpha_g}$

$$t^* = \frac{t}{\left(\frac{2\lambda}{\alpha_g}\right)^2 \frac{1}{\kappa}}$$

Equation (4.10) is sowed with the initial condition

$$\delta^*\ (0) = 0 \tag{4.11}$$

The solution for (4.10) and (4.11) is given implicity by

$$t^* = \frac{1}{6}\left[\ \frac{1}{2}\ \{(1 + \delta^*)^2 - 1\} - \log\ (1 + \delta^*)\right] \tag{4.12}$$

We now proceed to construct the solution for phase 2. The integration of equation (4.1) for this phase is carried from 0 to L over x and results in

$$\frac{d}{dt}\int_0^L T\ dx = K\left(\left.\frac{\partial T}{\partial x}\right|_{x=L} - \left.\frac{\partial T}{\partial x}\right|_{x=0}\right) \text{ for } t > t_1 \tag{4.13}$$

where t_1 is the time determined by equation (4.12) when $\delta = L$.

Again we assume a parabolic profile for the temperature

$$T = a + bx + cx^2 \tag{4.14}$$

The coefficients this time are determined by equations (4.3), (4.4) and

$$T(0,t) = T_1(t). \tag{4.15}$$

In this phase of solution T_1 is treated as the unknown. Once a, b and c are determined, the resulting temperature profile is substituted into equation (4.13). We then get

$$\frac{d\,\theta_1}{dt^*} + \phi\,\theta_1 = \psi \tag{4.16}$$

where $\theta_1 \equiv \dfrac{T_1 - T_a}{T_g - T_a}$

$$\phi \equiv \frac{2(1+H)/(H-L^*)}{L^*[1 + L^*\{\frac{2}{3} - \frac{1+L^*}{3(H-L^*)}\}]}$$

$$\psi \equiv \frac{2H/(H-L^*)}{L^*[1 + L^*\{\frac{2}{3} - \frac{1+L^*}{3(H-L^*)}\}]}$$

$$L \equiv \frac{\alpha_g L}{2\lambda}$$

and $H \equiv 2L^* + \dfrac{\alpha_g}{\alpha_a}$

The initial condition for equation (4.16) is

$$\theta_1\,(t_1^*) = \theta_{1,i} \tag{4.17}$$

$\theta_{1,i}$ has to be obtained from the first phase solution. The solution for equations (4.16) and (4.17) are given by

$$\theta_1(t^*) = \frac{\psi}{\phi} - (\frac{\psi}{\phi} - \theta_{1,i})\exp\{-\phi(t^* - t_1^*)\} \tag{4.18}$$

It is interesting to note that ψ/ϕ is the steady state solution of the problem as can be seen by either setting $d\theta_1/dt^* = 0$ in equation (4.16) or by taking the limit of expression (4.18) as $t^* \to \infty$.

Denoting this by $\theta_{1,st}$. We can write (4.18) as

$$\theta_1(t^*) = \theta_{1,st} - (\theta_{1,st} - \theta_{1,i}) \exp\{-\theta(t^* - t_1^*)\} \qquad (4.19)$$

$\theta_{1,st}$ can also be expressed by

$$\theta_{1,st} = \frac{H}{1+H}$$

4.6 Solution output

Three of practical results can be expected from the solutions given by expressions (4.12) and (4.19).

(a) Temperature distributions as a function of time: this result is not very useful from a practical point of view. However the maximum temperature in the system occurs always at $x = 0$. The rate of growth of this temperature would be useful in estimating the radiant heat transfer to the pan. Since $T_{1,st}$ is the maximum temperature recorded, it will provide indications whether a given material could be used with confidence for the application under consideration.
The relevant expressions for the temperature are given by:
for $t^* < t_1$,

$$\theta_1(t^*) = \frac{\delta^*}{1 + \delta^*} \qquad (4.20)$$

where δ^* is given by Eq. (4.12);
for $t^* \geq t_1$, $\theta_1(t^*)$ is given by Eq. (4.18).

(b) Time at which steady state is reached: this will indicate the relative importance of stored heat in the wall and heat loss from the outer skin of the wall

(c) Heat flow through the inner skin of the wall: this is calculated by

$$q = \int_0 \alpha_g \{T_g - T_1(t)\}\, dt$$

per unit area of the slab.
This is the information that will be useful in determining the stove construction material from the point of view of heat loss from the stove.

4.7 Results and discussion

We make a comparative study of three principal contendors for materials of construction of stove.
These are:
(a) clay (air-dried);
(b) ceramic;
(c) metal.

We need to specify several parameters before we can use the solutions to obtain quantitative information that could be effectively used for purposes of design. These parameters are T_g, α_g, α_a, L, and the material properties λ and κ. A major limitation on the usefulness of a theory is the knowledge of these parameters. In particular T_g and α_g can not be specified with any confidence for wood-burning systems. Appendix 1 discusses the problem in greater detail and provides a basis for the values chosen in this work.

Figure 4.4 presents the growth of the wall temperature at x = 0 with time. The parameter values used to obtain these results are shown as an inset in the diagram. The inner wall temperature of the clay wall raises slowest while the metal wall responds fastest - a well-known result. The steady state temperatures are 465, 566 and 655 C for metal, ceramic and clay walls respectively. The times, at which steady states are attained, are approximately 20 and 60 minutes for the metal and ceramic walls respectively. The clay wall does not attain steady state at all during the computational period of 2 hours. Finally, other temperature results that are useful are the times at which the outerwalls begin to sense the presence of the fire at x = 0. These times are less than 1 minute, about 5 minutes and about an hour for the metal, ceramic and the clay walls respectively

Table 4.1 gives the cumulative heat flow results for the same conditions used in fig. 4.4 for three representative times,

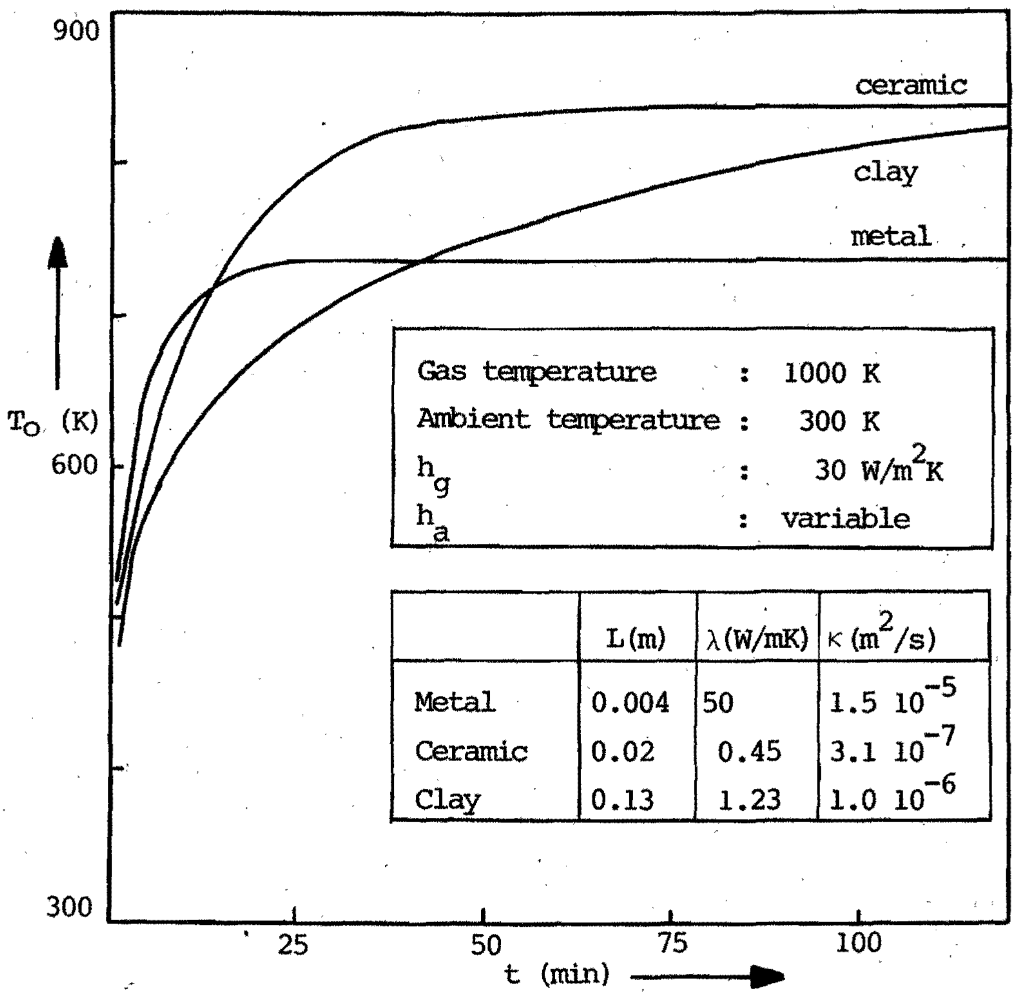


Figure 4.4: Growth of the wall temperature with time for the various stove walls.

Table 4.1 Heat flow results in MJ/m^2

Time in hours / Material	½	1	2
Metal	15.2	29.3	57.6
Ceramic	14.6	23.6	41.2
Clay	19.7	33.2	54.7

½, 1 and 2 h. The table shows that the ceramic is the material that loses the least amount of heat. In particular the common statement that the metal stoves lose a lot more heat than the clay stoves is not supported by these calculations. However, it should be pointed out that the results in Table 4.1 are the heat flows through the inner skin of the stove. There is no doubt that a substantial proportion of this heat is stored in the clay stove. However, it is to be doubted whether such stored heat could be used for any significant cooking activity.

Because of the considerable uncertainty involved in the evaluation of α_g, we present some results in Table 4.2 on its effect on the results obtained. For the configurations studied here, a factor of 3 variation in α_h produces a change in heat loss over a period of 2 hours of 74.4, 42 and 52% for metal, ceramic and clay respectively. In general the metal stoves require a more careful estimation of α_g than the ceramic or clay stoves. Another influence of change in α_g is the time required to attain steady state. For the metal it changes from 48 minutes to about 25 minutes when α_g changes from 15 to 45 kW/m^2K. The corresponding results for the ceramic wall are 120 and 90 minutes.

Next, Table 4.3 presents the effect of T_g on the heat flow. As can be gathered this is an important parameter for obtaining reasonable estimates of heat flow. T_g does not significantly affect the times over which the steady state is attained by the system.

Table 4.2 Effect of α_g on heat flow (MJ/m^2)

Time in hours / α_g W/m^2K	Material	½	1	2
15	Metal	10.7	20.3	39.5
	Ceramic	10.7	18.2	32.0
	Clay	13.0	23.3	40.5
45	Metal	17.6	34.7	68.9
	Ceramic	16.5	26.5	45.5
	Clay	23.5	38.9	61.6

Table 4.3 Effect of T_g on heat flow (MJ/m^2)
(α_g = 30 W/m^2K; Material: metal; L = 0.0015 m)

Time in hours / T_g K	½	1	2
700	7.32	14.1	27.6
800	9.67	18.7	36.8
900	12.2	23.7	46.7
1000	14.9	29.0	57.3

Finally, Table 4.4 compares the results of heat flow and wall temperatures with wall thicknesses for a clay wall. The main conclusion is that the heat flow through the inner skin of the stove wall is remarkably insensitive to changes in wall thickness.

Table 4.4 Influence of wall thickness on inner wall heat flow, temperature and outer wall temperatures

Time h / L m	½			1			2		
	Q	T_i	T_o	Q	T_i	T_o	Q	T_i	T_o
0.05	23.1	525	277	36.0	585	356	56.9	603	378
0.10	24.2	505	82	39.3	561	275	60.7	613	275
0.15	23.5	512	T_a	38.6	558	347	61.6	604	172
0.175	23.2	512	T_a	38.1	563	35	61.2	601	130
0.20	22.8	512	T_a	37.6	568	T_a	60.5	601	94

Notes:
Hot gas temperature: 727° C
Environment temperature: 27° C
λ = 1.23 W/mK
κ = $10^{-6} m^2/s$
α_g = 45 W/m^2K
Q is the cumulative heat flow in MJ/m^2
T_i and T_o in °C

It is important to point out here that the results of Tables 4.2 and 4.4 should be interpreted rather carefully. For the configurations we are considering there are three resistances to heat flow - the gas side (as represented by α_g), the wall itself (as represented by its thickness and the material), and the ambient side. The ambient side heat transfer coefficient was allowed to vary parametrically with the temperature of the outer face of the wall and α_a was estimated by using the formulas given in the Appendix. Thid does not greatly influence the relative results, however.

For the metal wall, ignoring the resistance of a metal wall, changing α_g from 15 to 45 W/m^2K changes the total resistance to heat flow only from .1333 to 0.07350. This is the reason why the large change in gas side heat transfer coefficient does not result in a correspondingly large change in heat flow. Since the ambient side heat transfer coefficient is the controlling resistance to heat flow, it was necessary to estimate this quantity more carefully.

For the case of varying thicknesses of the wall such a simple approach is not useful due to the large heat capacity of the wall. More detailed calculations are necessary before it it possible to explain the results of Table 4.3.

4.8 Concluding remarks

It is worthwhile to consider the usefulness of the highly idealized model presented in this work in predicting the heat flow through stove walls. Direct comparisons of the results are not possible since the test results on stove models of the type considered in this study are not available at this moment. Thus we restrict ourselves to some ad hoc comparisons with the results from the metal shielded fire shown in Fig. 4.5. It shows the wall temperature records as a function of time for the points marked on the stove diagram shown as an inset. Before making any strict comparisons, a few points about the experiments are to be mentioned. The periodic nature of the temperature record is a consequence of the mode of stove operation. The latter is done by periodically charging the stove with known quantities of fuel.

Comparison of Figs. 4.5 and 4.4 shows that the response of the actual stove to the fire is much slower than the prediction of the model. Secondly, the maximum temperatures attained in the real system are over 100 degrees lower than the ones predicted by the model. This suggests that the α_g and T_g assumed in the model are too high. For T_g = 700 K and α_g = 15 W/m^2K, the heat loss predicted by the model is over 3 times as large as the one computed from the temperature records measured during the stove operation.

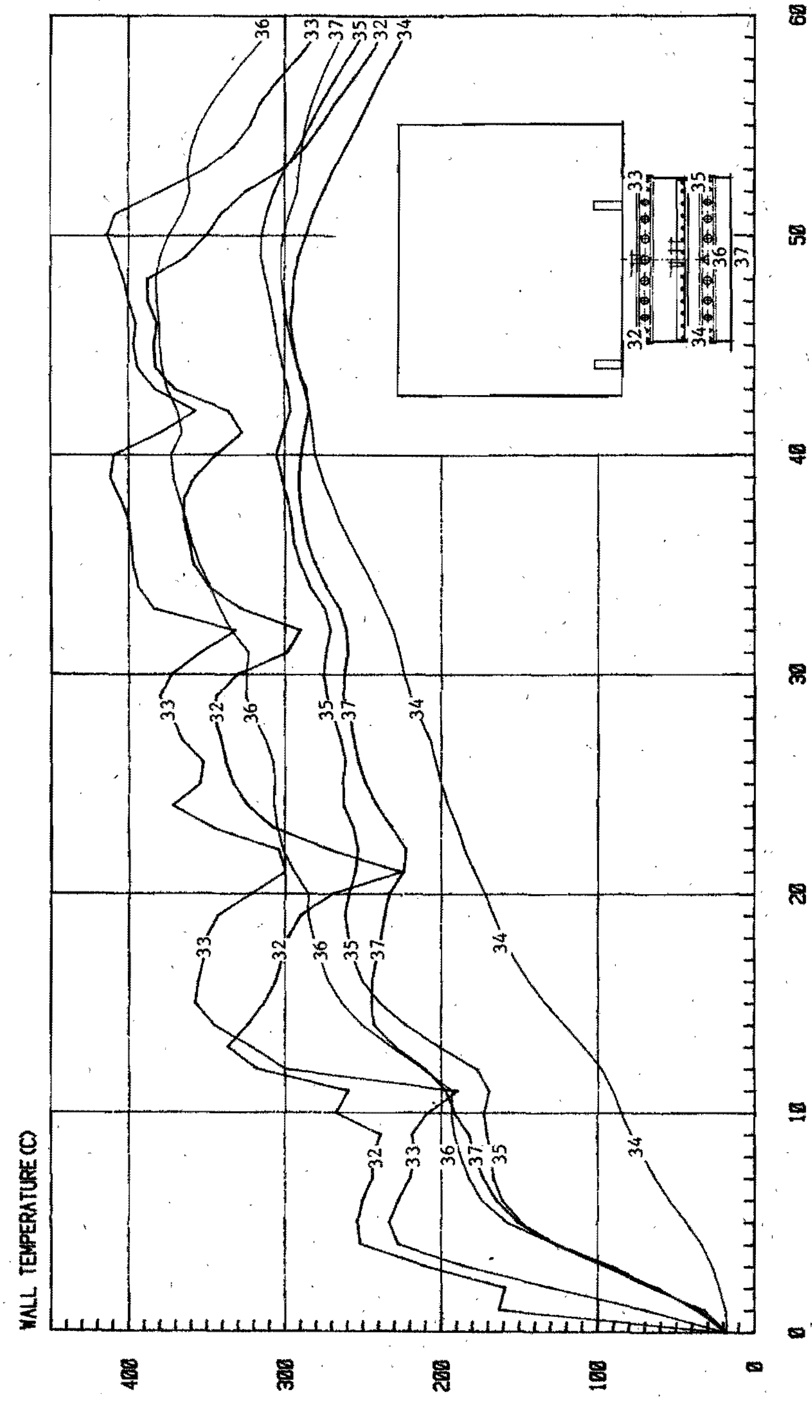


Figure 4.5: Response of the temperature of the stove walls on the fire.

There are a number of factors that probably cause these rather large differences. The main weaknesses of the model lie in treating the problem as a linear one and assuming T_g to be constant rather than periodic as it is in the real situation. It should also be pointed out that the heat loss from the fire box in the laboratory stove is just 1% of the total heat input into the stove. Estimation of such small heat losses by experiment are always subject to great errors. The main heat loss from the laboratory stove is from the shield covering the pan which accounts for over 27% of the total heat input into the stove. The model can be expected to provide better estimates of heat losses in this region since the nonlinear effect of radiation from the fuel bed is absent in this region.

The main use of the model as it exists now is to provide rather quickly the relative changes that could be expected by changing the geometric parameters as well as change of materials of construction.

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APPENDIX

ESTIMATION OF HEAT TRANSFER COEFFICIENTS AND COMBUSTION TEMPERATURES

The combustion temperatures are estimated with the assumption that the constituents of wood in a broad sense are 20% fixed carbon and the rest volatiles. The small amount of ash in the wood has been ignored in the present analysis. The fixed carbon burns on the fuelbed and the volatiles burn in the gaseous phase. The following energy balances have been used for estimating the combustion temperature in the gas phase and fuelbed.

$$m_v B_v = \sum_{j=1}^{4} m_j C_{p,j}(T_g)\{T_g - T_{ref}\} \tag{A.1}$$

$$m_v B_v = m_w B_w - m_c B_c \tag{A.2}$$

$$m_c B_c = m_a \overline{C}_p (T_{o,g} - T_a) + \int_o^{t^*} \sum_{i=1}^{5} \sigma A_o F_{o,i} (T_o^4 - T_i^4)\, dt \tag{A.3}$$

where m_v = mass of volatiles, kg/kg of oven dry wood

B_v = calorific value of volatiles, J/kg

m_j = mass of the constituent j in the combustion products, kg

$c_{p,j}$ = specific heat at constant pressure of constituent j in the combustion products, J/kgK

T_g = combustion temperature, K

T_{ref} = reference temperature taken as 300 K

m_w = mass of oven dry wood, kg

B_w = calorific value of oven dry wood, J/kg

m_c = mass of fixed carbon, kg/kg of oven dry wood

B_c = calorific value of fixed carbon, J/kg

m_a = mass of air required to burn completely 1 kg of carbon into CO_2, kg

$T_{o,g}$ = temperature of the gas leaving the fuel bed, K

σ = Stefan-Boltzmann constant, 5.7×10^{-8} W/m^{2}K^4

A_o = area of the fuel bed surface, m^2

$F_{o,i}$ = configuration factor between the fuel bed and surface i

T_o = the fuel bed temperature, K
T_i = the temperature of surface i, K
t* = time needed to burn 1 kg of wood, s

Equation (A.1) is an energy balance for the gaseous phase. The components j are taken to be CO_2, H_2O, O_2 and N_2. The mass of the components N_2 and O_2 is assumed on the basis of excess air values (which is treated as a parameter). Two important assumptions have been made while writing eq. (A.1). The first one concerns the dissociation effect. At the temperatures prevailing in a wood fire (see later) this assumption is not expected to introduce any significant error in the estimation of T_g. The second assumption is a more serious one and concerns the adiabatic nature of the combustion process. This is not true since the flames lose heat to the stove body and the pan by radiation and convection. This is partially compensated by the fact that we do not include the contribution of radiant heat transfer from the flames in our estimation of heat transfer coefficient to the walls. The left side of eq. (A.1) is determined from eq. (A.2).

Table A.1 presents the results of calculations carried out by Herwijn (1983) using eqs. (A.1) and (A.2) for different excess air factors, λ, m_w, m_w, B_w and B_c have been taken to be 1 kg, 0.2 kg, 18.7 MJ/kg and 33 MJ/kg respectively. The specific heat values as given in standard tables (see for example Handbook of Chemistry and Physics 1978) have been used.

A value of 1000 K has been used in the main text and this corresponds to an excess air factor between 2.0 and 2.5. The available experimental evidence with the group seems to support this assumption.

Table A.1

Estimates of Gas Temperatures

λ (excess air factor)	T_g (K)
1.5	1262
2.0	1074
2.5	950
3.0	861
3.5	794
4.0	742

We now turn to a discussion of Eq. (A.3). It represents an energy balance on the fuel bed. The first term on the right hand side is the heat carried away by the combustion gases of the burning carbon. m_a in that term is taken to be the stoichiometric amount of air to burn carbon to carbondi-oxide. This involves two assumptions. The first is concerned with the formation of CO. Since we have no experimental evidence on the amount of CO produced from the fuel bed combustion, we have taken this to be zero. The second assumption concerns the amount of air passing through the fuel bed. This is taken to be the stoichiometric quantity of air. The argument is that we are interested only in the product $m_a\overline{C}_pT_{o,g}$ and increase in m_a results in a corresponding decrease in $T_{o,g}$. The error introduced by this argument lies in the evaluation of $\overline{C}_p$ which is dependent on temperature.

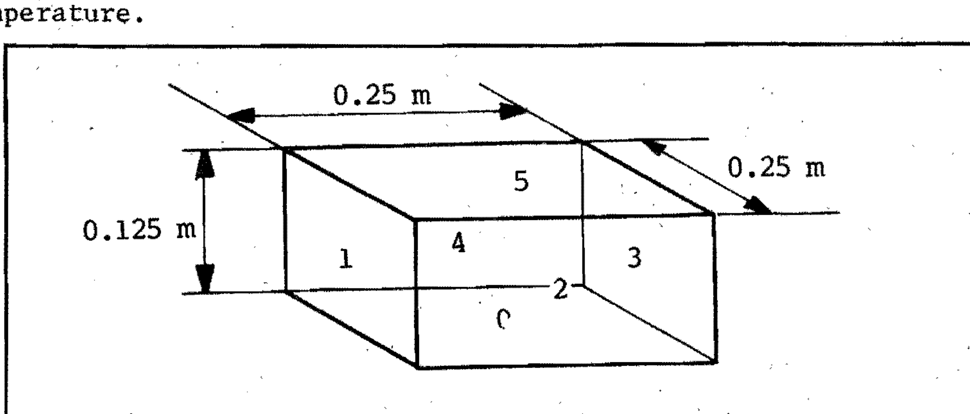

Fig. A.1 The enclosure for the radiant energy balance (surface 0 is the fuel bed, 1, 2, 3 and 4 are the sides of the combustion chamber, and 5 is the pan bottom).

The second term is the radiant energy balance in the enclosure (see Fig. A.1). The $F_{o,i}$ is the configuration factor and can be evaluated from the formulas available in standard heat transfer texts (see for example Eckert & Drake 1972).

The problem of solving for T_o requires the specification of $T_{o,g}$. A simple assumption is that $T_{o,g}$ is equal to the burning temperature of carbon which is said to be 1100 K (Wagner 1978). For the purposes of this work, this value has been assumed. A detailed evaluation of the fuel bed temperature will be taken up in a separate work that is under progress (Herwijn 1983). The pan temperature has been assumed to be 373 K. The walls have been assumed to be at a uniform temperature which is varied parametrically. Table A.2 has been constructed with these assumptions.

Table A.2
Fuel bed temperatures and heat transfer coefficients by radiation

T_w K	$\overline{T}_o$ K	q_r watts	$h_{g,r}$ w/m^2K
300	911	263	12.0
400	914	253	13.5
500	919	231	14.8
600	928	191	15.3
700	943	126	13.4

The measured wall temperatures rarely exceed 700 K and thus the computations have been stopped at this value. Note that the fuel bed temperatures and the heat transfer coefficients are not strong functions of the wall temperature.

The second mode of heat transfer is due to convection and is much more difficult to estimate. The first problem is to determine whether the flow is laminar or turbulent. We tacitly assume that the flow field is laminar because of the relatively low velocities involved. The second problem is to identify the type of solution to be used. There are two choices. The first is the conventional solutions for hydrodynamically and thermally developing flows in a square duct. However, this does not adequately represent the strongly accelerating flow field with varying temperature difference along the flow direction. From Eckert & Drake (1972) the following relations for linearly varying velocity and temperature difference are used.

$$\frac{Nu}{Nu_p} \simeq 1.7 \tag{A.4}$$

This gives the relation between Nusselt numbers for constant free stream velocity (Nu_p) and those for linearly varying velocity fields. Next we have

$$\frac{Nu}{Nu_{is}} \simeq 1.3 \tag{A.5}$$

which represents the relation between Nusselt numbers for constant temperature (Nu_{is}) to that of linearly varying temperature difference for the velocity field given for eq. (A.4).

The local Nusselt number for a flat plate with constant free stream velocity and constant temperature difference is given by

$$Nu_x = 0.332\ Pr^{1/3} Re_x^{1/2} \tag{A.6}$$

where $Nu_x = \frac{\alpha x}{\lambda}$

$Pr = \frac{\nu}{\kappa}$ (Prandtl number)

$Re_x = \frac{Vx}{\nu}$ (Reynolds number)

and ν is the kinematic viscosity.

Thus the local Nusselt number for the present case is given by,

$$Nu_x = 0.734\ Pr^{1/3} Re_x^{1/2} \tag{A.7}$$

This is not the end of the problem. Equation (A.7) provides a local heat transfer coefficient and what we require is an average value for use in the solutions. This requires much more effort than is envisaged in the present work and we use as a first approximation the known result for a flat plate with constant free stream velocity and constant temperature difference. This result simply is to double the coefficient in (A.6) and correspondingly for the present problem, we have

$$\overline{Nu} = 1.468\ Pr^{1/3} Re_L^{1/2} \tag{A.8}$$

The fluid properties are evaluated at the temperature T_g derived earlier.

The characteristic velocity for the Reynolds number is chosen as the velocity at the exit of the fuel bed and is taken to be 0.5 m/s and the characteristic length is 0.125 m. This leads to an average Nusselt number of 30 and the corresponding convective heat transfer coefficient of 16 W/m^2K.

Thus the total heat transfer coefficient is about 30 W/m^2K that includes both convective and radiative transfer. Due to the many uncertainties involved in the estimations provided here, the heat flow results in the main text are given for three values of α_g, namely 15, 30 and 45 W/m^2K. In addition the gas temperatures have also been varied parametrically.

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Part B: Stove designs

5. PERFORMANCE OF THE TUNGKU LOWON WOOD STOVE

by
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5.1 Introduction

In the framework of the research project for improving the efficiency of wood stoves to be used in the third world countries, a test programme has been carried out with an Indonesian mud stove.

This two-pot Tungku Lowon mud stove is based on a traditional Indonesian design. It has been developed by members of Dian Desa, an Indonesian field station.

The drawings and data for the construction materials were delivered by The Intermediate Technology Development Group in Reading, England (ITDG). The stove was made in the workshop of TNO's Division of Technology for Society in Apeldoorn, the Netherlands. In this report a full description will be given of the experiments carried out on the Tungku Lowon stove.

To promote collaboration among research groups working on the subject in different parts of the world. TNO has carried out a number of experiments in accordance with the ITDG-test procedure. On the other hand, some experiments using the TH/TNO-procedure have been carried out by ITDG.
Special attention will be given to efficiency and combustion performance; and a comparison will be made between the Tungku Lowon and the Nouna wood stove, with special reference to the above two items.

5.2 Design of the stove

The Tungku Lowon is a two-pot stackless mud stove. It consists of a combustion chamber without a grate, a flue gas channel and a second heating chamber (Fig. 5.1). The pan hole of the second heating chamber is provided with four apertures for transportation of the flue gases. A gap above the entrance to the combustion chamber prevents cracking of the stove wall when the stove is fired.

The combustion chamber, with a volume of 0.00755 m^3, is provided with two air-inlet holes for secondary air supply, one on either side of the combustion chamber. The stove body material is composed of:

-	clay	20 vol.%	
	sand	40 vol.%	
	ash	10 vol.%	water added 9 l.
	dried cow dung	20 vol.%	
	river sand	20 vol.%	

The total weight of the stove is about 70 kg. Due to shrinkage of the stove after construction, its dimensions change depending on the treatment it is subjected to. The overall dimensions are given in Table 5.1 for three stages of treatment, i.e. after building, after drying and after firing the stove.

5.3 Experimental details

The performance of the stove was determined by boiling water tests (Nievergeld et al, 1981). For these experiments two authentic Indonesian aluminium pans were used. The heights of the pans were 0.115 m and 0.095 m, and the diameters were 0.225 m and 0.20 m, respectively. The height of the pan above the fire plate was 0.18 m. Prior to placing the stove on a table its base plate was provided with a 0.05 m thick glass wool layer. Fig. 5.2 shows the experimental set-up of the Tungku Lowon Stove. Except for the gas sample line and the temperature measuring device, the experimental set-up was similar to that used in the earlier experiments (Claus et al, 1981). Flue gas samples were taken with a special sampling device consisting of a copper ring with an inner diameter of 0.015 m into which four tubes with an inner diameter of 0.003 m had been silver-soldered (Fig. 5.3). The length of the four tubes was such that the tube ends were leveled with the port inlets formed by stove and pan. The flue gas temperature was measured by four chromel/alumel thermo couples located in series at the inlets of the ports.
In order to check the repeatability of these measurements temperatures were measured sequentially in each of the four ports. The temperature spread over the width of the ports was ± 1°C.

Two types of tests were carried out with the Tungku Lowon stove. One serie of tests was made in accordance with the TNO/TH-procedure and one adopting the procedure used at ITDG. For the TNO/TH-tests unsmoothed white fir was used, the size of the wood chips being 0.02 x 0.03 x 0.2 m. The density of the wood was 350 kg/m^3. These experiments were carried out with oven-dry wood and with wood having a moisture content of 6%. A Bunsen burner was used for igniting the wood fuel.
The initial amount of water accommodated on the pans were 4 kg and 2 kg for the first and second pan respectively. The duration of the tests lasted over periods ranging from 1 to 48 hours depending on the type of experiments being carried out. The experiment was considered after the water had been boiling for a certain period of time. The data for these experiments are presented in Table 5.2. The tests carried out at Reading were done with unchaved yelutong wood with an average density of about 355 kg/m^3.

Dimensions of the wood sticks were 0.03 m * 0.03 m * 0.5 m. Moisture contents were 6% and 13% respectively. For igniting the fire, three sticks were dipped into kerosene in such a way that about 10 g of kerosene was absorbed in the wood. The duration of the Reading tests depended on the type of test performed. The start of all the experiments was considered to be the moment at which the wood first ignited. In Table 5.3 the data of these experiments are summarized. The following factors were investigated.

* Heat output (min. and max. heat output)
* Moisture content
  - yelutong wood 7% and 13%
  - white fir - oven dry and 6%
* Firing procedure
* Water content of pans with yelutong wood
  - 2 * 2 l water
  - 4 * 2 l water
* Wood igniting procedure.

5.4 Efficiency of the Tungku Lowon Stove

For the TH/TNO testing procedure, the efficiency is calculated using the formula as given in (Nievergeld et al, 1981).

The calculations involved in the ITDG-testing procedure concerning efficiency and burning rate are described in (Joseph & Loose 1981).

5.4.1 Effect of the heat output of the fire

5.4.1.1 TH/TNO experimental procedure

To investigate the influence of the heat output of the fire the heat load on the stove was varied by changing the charge-time. The results of the experiments are summarized in Table 5.2. In Fig. 5.4 the overall efficiency and the efficiencies of pans 1 and 2 are plotted as a function of heat output of the stove.

Under the conditions investigated the efficiency of the Tungku Lowon stove varies between 18% and 22%. Fig. 5.4 shows that the highest efficiency is obtained with a heat output of about 3.5 kW. As opposite to that of the Nouna wood stove the efficiency of the Tungku Lowon wood stove decreases when the heat output is increased. The change in efficiency can be attributed almost entirely to the first pan, which is about three times higher than that of the second pan. The operating range of the stove is limited. With two pieces of wood per charge the Tungku Lowon stove can be operated with a minimum heat output of 2 kW and a maximum heat output of about 7 kW. The minimum capacity was the result of the fire having gone out. The maximum capacity was limited by a combination of insufficient draught, the moderate size of the combustion chamber and blocking of the flue gas channel.

5.4.1.2 ITDG experimental procedure
(experiments carried out by TH/TNO)

A number of tests have been carried out using the ITDG testing procedure. These tests can be categorised into BP_1, BP_2, BP_2S_{30} tests. BP_1, BP_2 and BP_2S_{30} refer to the time of completion of a test. BP_1 refers to tests that ended when the first pan started boiling, BP_2 to tests that are terminated when the second pan (after being exchanged with pan 1 at BP_1) started boiling. BP_2S_{30} refers to tests that are finished 30 minutes after BP_2. The tests are shown schematically in fig. 5.5. A complete description of the tests is given by Joseph & Loose (1981). The data relating to these tests are presented in Table 5.3. The table shows the results of the experiments with an initial amount of water of 2 * 2 kg and 4 * 2 kg in the first and second pan respectively. The moisture contents of the wood were 13% and 6%.

Due to the different set-up for carrying out the experiments, a direct comparison of the results of the two procedures is not possible. This is certainly true of the efficiency measurements. For the TH/TNO-procedure this efficiency has been deliberately determined for the whole operation range of the stove, whilst for the ITDG measurements the efficiency is an average result of a boiling water test during which the pans were exchanged and firing rate was modified.
Therefore the only comparison that can be made is on the basis of the results obtained for the BP_1 test for a water content of 4 kg in the first pan. The results of this comparison are given in Fig. 5.6. This graph shows that the firing technique used by ITDG results in first-pan efficiencies which are about 6% higher and that the form of the curve showing the relationship between efficiency and heat load is the same.

In chapter 5.7, where the difference in the two testing procedures is discussed in more detail, reasons for the difference in BP_1-results will be dealt with.

5.4.2 Effect of moisture content of the wood

In order to determine the effect of the moisture content of the fuel on the efficiency of the stove, data have been collated in Table 5.4 for white fir and in Table 5.5 for yelutong wood. Comparing the experimental results for white fir with a moisture content of 0% and 6% respectively (see also Fig. 5.4) it is obvious that there is no marked difference in the efficiency of the stove for similar heat loads.

For yelutong wood the efficiency is more affected by the moisture content of the wood. When the data with respect to wood with 6% moisture are compared with those for wood of 13% (see Fig. 5.7), it is obvious that for the yelutong experiments moisture content has a considerable influence on stove efficiency such that the lower the moisture content the lower the efficiency. For the BP_2 tests the average efficiency is about 7% lower than for the wood with 13% moisture.
Visual observations showed that when using wood with the 6% moisture content, during the whole testing period, a lot of wood burns outside of the stove body. The reason for this is that the flame proportion along the dryer sticks is much faster. This effect can only influence the efficiency measurements when long sticks are used as fuel. This does not occur when the much shorter pieces of white fir are used. It can therefore be concluded that the way in which moisture content affects stove efficiency very much depends on the manner in which the stove is fired. When the pieces of wood are so small that all the heat is released inside the stove there seems to be hardly any effect of moisture content on stove efficiency.

5.4.3 Effect of the water content of the first pan

The experiments forming part of the TH/TNO programme have been performed with 4 kg water in the first and 2 kg water in the second pan, whilst the ITDG experiments have been carried out with 2 kg water in the two pans. In order to correlate the TH/TNO- and ITDG measurements a few experiments were performed under similar firing conditions and different water contents of the first pan.

The experimental results are given in Table 5.3.
From this table Fig. 5.8 has been derived, which shows the time BP_1 for the 4 kg pan and 2x BP_1 for the 2 kg pan as a function of the heat load of the stove.
It is obvious that within the scatter of the data obtained no difference in the two experiments can be observed. This means that the average heat losses of the pan per unit time can be considered constant during the heating-up period of the pan. This also means that within the range investigated the water content of the pan does not affect the efficiency, and that for the same pan diameter boiling test values of TH/TNO are comparable with those from ITDG.

5.5 Combustion performance

5.5.1 Effect of the heat output of the fire

5.5.1.1 TH/TNO experimental procedure

The data relating to the combustion performance of the Tungku Lowon stove are given in Table 5.6. This table shows the general flue gas analysis. No soot concentrations were measured because technical problems were experienced in taking reliable samples from the four slits through which the flue gas escapes. However, tests made with the Nouna wood stove showed concentrations to be small, so that they could be neglected in calculating heat losses. As a result of technical problems and limited possibilities of the equipment concerned, C_xH_y could not be measured at every point of time.
In Figure 5.9, CO_2, CO and temperature of the flue gases are plotted as a function of heat output of the stove.
This figure shows that the CO_2-concentration increases when heat output is rising.
This is caused by the fact that at higher heat loads a relatively smaller amount of air is sucked in by the stove (Figure 5.10), which has a favourable effect in flue gas losses. However, the smaller amount of air (low excess air factor) also results in higher CO-concentrations.
Figure 5.11 shows the CO-concentration against the excess air factor.
As Figure 5.4 demonstrating the most working point of the stove is at a heat output of about 3.5 kW. However, the amount of combustion air required for this heat output is more than twice the stoechiometric value, and the CO-concentration is still unacceptably high.
An other serious problem experienced with the Tungku Lowon stove is the smoke emission during the starting and almost entire testing period.
Improvement of the situation with respect to the CO- and smoke production can be achieved by increasing the amount of excess air. The excess air factor strongly influences the CO-concentration (Figure 5.11), but hardly affects the efficiency (Tables 5.2 and 5.6).

The best way to do this is to provide the stove with a chimney. This can also be used to emit the combustion products out from the room or house. Other solutions such as a larger cross-section of the flue gas duct maybe to be considered worthwhile.
In any case it must be concluded that the stove in its present state is not suitable for indoor use.

5.5.1.2 ITDG experimental procedure

The results obtained with the yelutong wood sticks are given in Table 5.7.

5.5.2 Effect of moisture content of the wood

When the data for white fir with 0% and 6% moisture with respect to the combustion performance of the stove are compared, no substantial differences are observed (Tabel 5.4). All that can be said is that there is a small tendency that CO_2 and temperature of the flue gases rise with the increase of the moisture content.

The change in moisture content from 13 to 6% of the yelutong has a marked influence on the combustion performance of the stove. (Table 5.5). First of all there is an increase of about 50% in the combustion rate when the moisture content drops from 13% to 6%. Secondly combustion performance becomes much poorer. The CO-concentration increase from an average of 0,51% to 1,26%, in spite of the fact that the flue gas temperature rise. The reason for this is the much lower amount of excess air when the moisture content is 6%. However, the CO_2- and CO-concentrations and therefore the amount of excess air are of the same order of magnitude as observed for the same combustion rate when the 13% moisture wood is fired.
This confirms the opinion that the heat load is the factor that determines the combustion performance.

5.6 Heat balance

In order to get a clear insight into stove performance it is necessary to draw up heat balances for the stove. The construction of the flue gas outlet of the stove was such that it did not allow available sampling techniques to be used. The development of a special technique for sampling soot would have been far outside the scope of the experiments. However, heat balances drawn up for the Nouna Stove (Claus et al, 1981) had shown that the amount of heat lost because of soot emission could be neglected. It was decided therefore to omit the soot concentration measurements. Because suitable equipment for measuring C_xH_y was not available during all the tests the accurate value of the C_xH_y-concentration in the flue gases could not always be determined. To be able to estimate the amount of heat lost with unburned hydrocarbons a separate experiment was performed in which the ratio CO/C_xH_y was determined. Using this relationship the heat lost as a result of unburned constituents was calculated.

The set-up of the heat balance for the Tungku Lowon is similar to that used for the Nouna Wood Stove. A description of the set-up for establishing the heat balance is given in (Nievergeld et al, 1981; Claus et al, 1981).
The heat balance is presented in Table 5.8.
For the experiments for which no proper calculations could be made the accumulated heat and the part "unaccounted for" have been added together. The percentage "unaccounted for" changes from -6% to +8%. In this figure the heat lost through the wood fuel inlet port has been taken into account.

An other approach to getting an insight into the amount of heat accumulated in the stove body was the use of an excisting computer program for kiln calculations. (see Appendix 1 in Claus et al, 1981). To translate the actual situation into the computer model some simplifications in the construction have been made. Combustion chamber, flue gas channel and second heating chamber have been scheduled in one hole over the length of the model. The properties of the material, such as thermal conductivity, specific heat and density, are based on the properties of the actual materials of which the stove is constructed.

The model represents only half of the stove, because for reasons of symmetry the centra plane can be considered to be an adiabatic surface. The model and its constituent parts are shown in Figure 5.13.

In the model the input can be effected in two different manners. The first is based on the measured inside wall temperatures of the stove. These temperatures are measured in the combustion chamber (3x), flue gas channel (1x) and second heating chamber (2x). The second is based on a knowledge of the total heat input to the stove body.
With this computer model calculations were made for one experiment. This experiment comprised a heating period followed by a cooling down period. Using the program the average surface temperature of the side walls of the stove were calculated. In Figure 5.14 the calculated temperatures and the average measured surface temperatures are plotted for run 7. In this graph three lines can be distinguished. The first line (top line) is the average measured surface temperature, the second one is the calculated line based on the total heat input and the third one is the calculated line based on temperature measurements in the stove (combustion chamber, flue gas channel and second heating chamber).
It is clear that the results based on the heat input as a starting point are in good agreement with the measured temperatures, and that the choice of the temperature as input gives less accurate results.
The reason for this is that the inside temperature of the stove varies considerably along the combustion- and heating chamber, so that an average temperature based on a few temperature measurements is not a sound basis for a good estimate of the average wall temperature.

The "unaccounted for" of the heat balance is acceptable, especially when the difficulties encountered in making reliable flue gas analyses are taken into account. The most striking feature in the heat balance for the Tungku Lowon stove is the low sensible heat loss. Depending on the heat load this loss varies from 10 to 23%.
On the other hand the heat loss due to unburned constituents is high with values ranging from 7 to 17%. The above two figures can be explained by the low excess air factor maintained in this stove. The accumulated heat varies from 37 to 54%.

5.7 Comparison of TH/TNO and ITDG operating and testing procedures

5.7.1 Comparison of operating procedures

The experiments that have been carried out by TH/TNO and ITDG using the same test procedure are summarized in Table 5.9 and 5.10
When looking more closely at the experiments summarized in Table 5.9, 5.9a and Figure 5.15 it appears that the efficiency determined by ITDG lies between 18 and 23% and that determined by TH/TNO between 18 and 22.4%. The efficiency of the first and second pan is roughly in the same order of magnitude just as the boiling times of pan 1. This boiling time varies depending on the heat output between 60 and 30 minutes. These figures are in good agreement. Furthermore the operating range is found to be 2 to 7 kW for the TH/TNO experiments and 2.7 to 7 kW for the ITDG experiments.
It can therefore be concluded that the results of the above two kinds of experiments are nearly the same. However, the highest efficiencies are not observed at the same heat outputs. This is evident from Figure 5.15, where the efficiencies of the respective experiments are plotted against heat output. It appears from this figure that the efficiencies as measured by TH/TNO have the tendency to decrease at the rising of the heat output, whilst the ITDG experiments show an increase of efficiency at a rising heat output.

By comparing the measured flue gas composition figures it is remarkable that these show no similarity at all. This is clearly demonstrated in Figure 5.16, where the CO_2- and CO-concentrations are plotted against heat output. This figure shows that the CO_2-concentrations measured at Apeldoorn are about two to three times higher and the CO-concentrations even more then four times higher than those measured at Reading.

What is even more important is that the ITDG-measurements show a greater scatter and have certainly not the same measure of accuracy as the TNO-data. Only the value for a heat load of ±6 kW is close to the TNO-data, and for this experiment the measured efficiency is nearly the same as that found by TNO.

When the possibility of a failure in the sampling and analysing system is excluded, it is reasonable to assume that the excess air factor has been much higher during most of the ITDG experiments. Unfortunately no temperature measurements and heat balance data are available of these experiments. This would have provided the opportunity to check the sensible heat losses calculated with the measured CO_2- and CO-concentrations in the heat balance of the stove.
Based on the available data it must be concluded that the performance of the stove at ITDG has not been the same as that at TNO during most of the experiments.
A probable reason for this difference could be a difference in dimension of the flue gas channel either resulting from different manufacturing tolerances or from blocking due to soot, tar or the loading procedure applied.

Table 5.10 gives a summary of the results of the experiments carried out in accordance with the ITDG testing procedure (Joseph & Loose, 1981). For making an overall comparison possible between the ITDG an TNO/TH test results the average values for efficiency, boiling point and burning rate have been collated in Table 5.11.
From this table it is clear that when the same test procedure is used entirely different results are obtained with respect to boiling times. The efficiencies are in good agreement, but there is a great difference in burning rates especially in the BP_1 part of a complete test. For this period the burning rate during the ITDG-test is almost twice than that found during the tests conducted at TNO.
The amount of fuel necessary to make the water boil in the first pan is smaller for the tests carried out by TNO. The fact that the overall efficiency is smaller as well, is caused by the poorer heat transfer to the second pan.
At BP_1 the temperature of pan 2 for the TNO-test is ±40°C and for the ITDG-test ±60°C, which means that at a starting temperature of about 20°C approximately twice as much heat is absorbed by the 2nd pan during the ITDG experiments. Because the duration of the test is about 1.6 times shorter, the heat transfer to the second pan during the ITDG experiments has been about 3.2 times higher. This example illustrates the difference in performance in spite of the fact that a similar test procedure was needed.

The fact that the heat transfer also to the second pan is so much higher and that the burning rate especially in the BP_1-period is higher too, indicates that in the ITDG experiments the wood has burned with much larger flames.

In Table 5.3 some measurements carried out by TNO with wood of 6% moisture content are given. For these measurements the average BP_1 is 14.5 minutes, BP_2 is 21.2 minutes and the average burning rate is 20.2 minutes. These values are much closer to the ITDG-values. An explanation for the discrepancy between the TNO- and ITDG-measurements possibly lies in the fact that different techniques were used to measure moisture content.

5.7.2 Comparison of test procedures

With the experience obtained with the two test procedures it is possible to make a comparison. This will be done for the experiments performed at TNO, for a number of results.

Boiling time BP_1

This time, necessary to make the contents of the first pan boil, is measured in each test procedure.

Figure 5.17 clearly indicates that for the same water contents in pan 1 the boiling time for the ITDG firing procedure is shorter, i.e. by 20-35%, depending on the heat load.

One explanation of this difference in boiling time is that 10 grams of kerosene are used in the ITDG experiments to light the fire.

For an average BP_1-time of 35 minutes this means that the heat load has been 0.23 kW higher than that determined from the decrease in wood weight. In relation to the 4 kW heat load this means a 6% higher heat load. This does not fully cover the difference in boiling time, however. An other possible explanation is the difference in loading procedure resulting in a different excess air factor. Figure 5.12a shows, however, that this difference (if any) is very small and that it is certainly not large enough to clarify the difference in BP_1.

A more probable explanation is that during the first part of the experiment more volatiles are burning. This results in a higher heat production per kg weight loss than during a long trial or a trial such as that performed at TH/TNO in which smaller wood charges are used.

An other factor worth considering in this respect is the uncertainty involved in weighing the wood. Errors may be introduced by rubbing the charcoal from the sticks, because of ash still present in the sticks. An other important aspect is that the different loading procedures may result in a different distribution of the flames in the combustion chamber, with the ultimate effect of a higher efficiency for the ITDG-test procedure.
From the factors quoted in the foregoing the extra heating value of the kerosene and the higher heat production due to combustion of the volatiles are the principal errors introduced in the ITDG procedure. In the ITDG experiments it is clearly noticeable that the BP_1 (and therefore BP_2 as well) is strongly affected by the temperature of the stove wall. There remains however, a good relationship between BP_1 and the total heat output of the fire. It is advisable therefore to quote BP-values always in relation to the heat load at which they have been determined.

Boiling time BP_2

In the ITDG-test procedure to determine BP_2 the position of pan 2 is changed after BP_1 has been reached. This procedure is completely different from the one used by TH/TNO, and it is not possible therefore to make a comparison between the two.

Efficiency

The same agreements as used for the BP_1 and BP_2 tests apply to the efficiency of the stove. Only the efficiencies of the first pan are directly comparable during the BP_1 test. The efficiencies are shown in Figure 5.6.
With respect to the BP_2-tests only overall efficiencies can be compared, whilst it is hardly possible to make a comparison for the BP_2S_{30}-tests because the firing conditions were altered during operation.
In Figure 5.18, BP_2 efficiencies are given for the TH/TNO- and ITDG-experiments. Because of the stable combustion rate reached with the TH/TNO-testing procedure during the measurements (Claus et al, 1981), the efficiency determined for the whole test period has been used.

Differences in efficiency are not noticeable. Therefore it can be concluded that when the duration of the test is long enough any differences in the measured efficiency become negligible. This strengthens the arguments used in explaining the difference in BP_1 formed for the ITDG- and TH/TNO procedures because during longer testing periods the effect of kerosene and volatiles combustion becomes of minor importance.

Heat balance

One of the important tools to find ways for improving stove performance is the heat balance of the stove.
To be able to draw up such a balance with an acceptable degree of accuracy the duration of the test should be long enough and the burning rate must be kept constant.
This aim can only be reached by using the TH/TNO testing procedure.

Firing practice

During the experiments it was experienced that in case of the ITDG-testing procedure it is easier to maintain a constant fuel bed. The batch-wise loading procedure practised in the TH/TNO procedure can cause blocking of the flue gas channel especially at high heat loads.

In conclusion it can be said that the ITDG-firing procedure has some practical advantages but does not give the test stability that is necessary to draw reliable conclusions for further development of wood stoves.
In view of the latter aspects the TH/TNO procedure is more suitable.

5.8 Comparison between the Nouna Wood Stove and the Tungku Lowon Wood Stove

5.8.1 Efficiency

When the overall efficiencies of the Nouna wood stove (between 17 and 23%) and the Tungku Lowon wood stove (between 18 and 22%) are compared, it can be concluded that there are no significant differences in performance of the stoves under cooking conditions as prevail in the field, although the efficiency of the second pan is generally higher for the Tungku Lowon stove than for the Nouna stove.
It is important to notice that the Nouna stove is provided with panholes of 0.25 and 0.2 m and that the pan diameters used for the Tungku Lowon experiments are 0.225 and 0.2 m respectively. The efficiency for the first pan of the Tungku Lowon stove must therefore be corrected by 25% to make figures comparable. When this correction is taken into account efficiency range of the Tungku Lowon stove becomes 21-26%, which is considerably higher than the efficiencies of the Nouna stove. This is in spite of the fact that the distance above the fire plate is 0.18 m for the Tungku Lowon stove and 0.15 m for the Nouna stove so that a lower efficiency could be expected.
An explanation for the higher efficiency is no doubt the lower excess air factor with the Tungku Lowon stove, which results in higher flame temperatures.

5.8.2 Combustion performance

As stated already in Chapter 5.5 the combustion performance of the Tungku Lowon stove is very poor. This is caused by the lack of combustion air. As compared with the Nouna stove the cross-sections of the flue gas channels and inlet- and outlet ports are of the same order of magnitude. The main and crucial difference is the stack with which the Nouna stove is provided.
The smallest cross-section in the Tungku Lowon stove has the flue gas outlet with an area of 0.0032 m^2. In case of the Nouna stove the smallest cross-section is that of the damper aperture, which for a 37.5% opened damper amounts to 0.0033 m^2.

For these more or less comparable situations the excess air factor for a heat load of 5 kW is 2.06 for the Tungku Lowon stove and 5.2 for the Nouna stove. For a heat output of 7 kW these figures are 1.6 and 4 respectively. This means that the amount of combustion air in the Nouna stove is 2.5 times larger than in the Tungku Lowon stove.

5.9 Conclusions

With respect to the different aims of the trials the following conclusions can be drawn.

Performance of the Tungku Lowon

- The main characteristic of the Tungku Lowon is that it can be operated at heat loads ranging from 2 to 7 kW, with efficiencies from 18 to 22% and BP_1 times from 15 to 18 minutes. This applies to an arrangement with two pans on top of the stove.

- The efficiency of the Tungku Lowon is not very sensitive to burning rates.

- The efficiency of the first pan is about 3 times that of the second pan.

- Exploratory measurements to investigate the effect of moisture content indicate that when the wood fuel pieces are so small that they burn completely inside the combustion chamber no marked differences in efficiency and combustion performance can be noticed for wood with 0% and 6% moisture content when the same heat load is used as a reference point.

- The efficiency measurements give similar results when the water contents of the first pan are 2 kg and 4 kg respectively provided the heat load is used as a reference.

- The combustion performance of the Tungku Lowon is poor. Almost during the entire duration of the test smoke escape from the stove. The CO-content of the combustion gases is unacceptably high. This makes the Tungku Lowon in its present state unsuitable for use in badly ventilated kitchens.

- Research to improve the combustion performance of the stove should concentrate in the size of the flue gas channels, the application of a chimmey and/or revising the back.

- It is possible to draw up heat balances with a good degree of accuracy ("unaccounted for" between -6% and +8%). Accumulated heat ranges from 37 to 54%, sensible heat losses from 10 to 23% and losses due to unburned constitnents from 7 to 17%.

- Wall temperatures predicted with the aid of a computer model are in good agreement with the measured values when the heat flow to the stove wall is used as input for the computer program.

Comparison between the ITDG and TH/TNO experiments

- When the TH/TNO testing procedure is tried out at ITDG and TNO, the efficiency measurements give similar results. This also applies to determination of the operating range and boiling time BP_1. The differences in measured gas concentrations are so large, however, that, irrespective of failures in the sampling and analysing system, it must be concluded that there has been a different in performance of the stoves during the greater part of the experiments.

- When the ITDG testing procedure is carried out at ITDG and TNO, the efficiencies are in good agreement. However completely different results are found for the boiling times BP_1, BP_2 and BP_2S_{30} because of difference in the burning rate.

- Measurements conducted at TNO show that the time at which the first pan starts boiling (BP_1) is different for the TH/TNO and ITDG testing procedure. The main reasons for this difference are:
  - the method of ignition adopted
  - the difference in fuel wood length and
  - the amount of wood in the fire box.

- Measurements carried out at TNO show that overall efficiencies during the BP_2-period are in good agreement for the TH/TNO- and ITDG-procedures.

- A proper heat balance can only be drawn up when the TH/TNO-testing procedure is used. Therefore this procedure provides very good physical insight into the working of a wood stove and thus is very suitable for development work in wood stoves.

- For the Tungku Lowon the ITDG-testing procedure has partical advantages because batchwise loading as applied in the TH/TNO-testing procedure can cause blocking of the flue gas channel especially at high heat loads.

Comparison of the Tungku Lowon and the Nouna Stove

- For the same pan diameters the efficiency of the Tungku Lowon Stove is 3 to 4% (absolute) higher than that of the Nouna Stove. The explanation is the low excess air factor at which the Tungku Lowon Stove operates.

- The combustion performance of the Tungku Lowon Stove is very much poorer than that of the Nouna Stove. This is caused by the chimney of the Nouna Stove. This chimney introduces so much draft that the amount of combustion air in the Nouna Stove is about 2.5 time larger than in the Tungku Lowon Stove.

5.10 References

P. Nievergeld, W. Sulilatu, J. Meyvis (1981)
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S. Joseph, J.C. Loose (1981)
Testing of a two-hole Indonesian mud stove.
Interim Technical Report
Intermediate Technology Development Group.

Table 5.1

	Overall dimensions in (m.)		
	After building	After drying	After firing
Height of the stove	0.22	0.22	0.22
Height of the combustion chamber	0.19	0.19	0.19
Diameter of the combustion chamber	0.18	0.185	0.185
height second - heating chamber	0.105	0.095	0.095
Diameter of the heating chamber	0.14	0.135	0.135
Wood entrance			
height -	0.11	0.10	0.10
width -	0.15	0.165	0.165
Split above combustion chamber	0.005	0.02	0.02
Cut aways heating chamber			
width -	0.04	0.04	0.04
depth -	0.02	0.018	0.018
cross-section of the flue gas channel -			
height -	0.07	n 0.07	0.07
width -	0.10	n 0.095	0.095
Total length of the stove	0.70	0.70	0.70
Air holes	0.02	n 0.017	0.017

Table 5.2 Efficiency of the Tungku Lowon wood stove as a function of the heat output of the fire

kind of wood : white fir.
size of the wood pieces : 0.02x0.03x0.2 m
depth of pans in stove : –
initial amount of water : first pan 4.0 kg
second pan 2.0 kg

Symbols:

m.c. – moisture content of wood [%]
Δm_f – mass of charge [kg]
Δ_t – time between the two charges [min]
$\dot{Q}$ – heat output of the fire [kW]
n_i – number of charges [1]
m_f – total amount of wood used [kg]
t_t – total burning time [min]
T_i – initial temperature of the water [^oC]

t_h – time to boiling : t_{b1} – of pan 1 [min]
: t_{b2} – of pan 2 [min]
m_s – amount of water evaporated : m_{s1} – of pan 1 [kg]
m_{s2} – of pan 2 [kg]
η – efficiency : η_1 – of pan 1 [%]
η_2 – of pan 2 [%]
η_t – total [%]

TNO/TH – experiments

run no.	m.c. [%]	Δm_f [kg]	Λ_t [min]	$\dot{Q}$ [kW]	n_i [1]	m_f [kg]	t_t [min]	T_i [^oC]	t_h [min]		m_s [kg]		η [%]		
									t_{b1}	t_{b2}	m_{s1}	m_{s2}	η_1	η_2	η_3
charge: 2 pieces															
5	oven dry	0.110	10	3.43	11.	1.2098	117	17	53.5	96.5	0.68267	0.2455	15.64	5.93	21.60
7	"	0.106	10	3.40	12	1.2728	120	20	50.5	106	1.040	0.2368	16.90	5.50	22.40
6	"	0.107	9	3.71	9	0.9625	81	19	49	68	0.4852	0.1007	15.0	5.5	20.50
1	"	0.097	8	4.20	11	1.1837	88	18	46.5	64	0.7757	0.1909	15.1	5.5	20.60
4	"	0.104	6.5	5.02	10	1.0448	65	21.5	40.5	65	0.4715	0.1181	13.9	5.6	19.50
2	"	0.110	5	6.88	13	1.4323	65	17	35	65	0.7214	0.1436	13.4	4.5	17.9
9	"	0.113	15	2.20	6	0.6776	91	19.5	63	–*	0.2547	0.0323	15.85	4.83	20.7
charge: 1 piece of wood															
3	oven dry	0.054	8	2.10	16	0.8283	128	16.5	76	–*	0.4917	0.0682	15.8	5	20.8
8	"	0.062	8	2.10	16	0.9271	136	20.5	?	–*	0.6514	0.0715	16.1	4	20.1
charge: 2 pieces															
10	6	0.120	10	3.16	10	0.2003	100	20.5	48	82	0.7354	0.1916	15.65	5.82	21.5
11	"	0.123	6.5	4.07	10	1.2301	65	20.	42	50	0.4626	0.1413	15.04	6.31	21.35
12	"	0.062	8	2.13	16	0.993	128	20.5	61	–*	0.6496	0.0598	17.1	4.2	21.3

*) not cooked

Table 5.3 TH/TNO Experiments - according the ITDG Procedure.

kind of wood : yelutong.

TNO/TH - experiments

water content of pans 2 x 2 kg.

Run no.	type of test	efficiency pan 1 (%)	efficiency pan 2 (%)	boiling time pan 1 (min)	boiling time pan 2 (min)	temp. pan 2 (^{o}C)	total test time (min)	$\dot{Q}$ (kW)	burning rate (g/min)	stove condition	MC (%)
18	Bp 1	18.2	2.56	31	-	34	31	2.48	8.45	cold	13
19	"	17.3	4.4	22	-	37	22	3.58	12.17	hot	"
20	"	15.3	4.9	21.5	-	41	21.5	4.09	13.92	"	"
21	"	15.4	5.2	23.5	-	45	23.5	3.93	13.37	"	"
26	"	19.1	2.9	38	-	30	38	1.77	6.03	cold	"
28	"	15.8	6.1	22	-	50	22	4.03	13.71	hot	"
22	Bp 2	11.61	9.05	27	41.5	35	41.5	3.38	11.51	cold	"
23	"	11.34	9.82	20	30	49	30	4.53	15.41	hot	"
24	"	11.55	9.91	17	28	46	28	4.75	16.17	"	"
15	Bp2S30	9.75	11.28	21.5	36.5	38	66.5	3.14	10.69	cold	"
16	"	10.24	13.81	20.5	30	52	60.0	3.27	11.13	hot	"
25	"	9.53	14.12	17.0	28	45	58	3.50	11.89	"	"
27	"	9.10	13.35	22.0	36	34	66	2.98	10.13	"	"
29	"	10.02	14.8	17.0	27	46	57	3.3	11.24	"	"

water content of the pans 4 x 2 kg.

Run no.	type of test	efficiency pan 1 (%)	efficiency pan 2 (%)	boiling time pan 1 (min)	boiling time pan 2 (min)	temp. pan 2 (^{o}C)	total test time (min)	$\dot{Q}$ (kW)	burning rate (g/min)	stove condition	MC (%)
1	Bp 1	19.6	3.62	32	-	50	32	4.19	14.24	cold	13
2	"	19.26	-	46	-	43	46	2.92	9.94	cold	"
3	"	16.41	4.5	30	-	67	30	5.56	18.91	hot	"
4	"	17.97	5.3	27	-	70	27	5.51	18.74	hot	"
5	"	16.97	5.7	26.5	-	77	26.5	6.06	20.60	hot	"
6	Bp 2	14.85	7.28	37	45	54	45	4	13.62	cold	"
7	"	15.18	7.15	27	36	62	36	4.91	16.70	hot	"
8	"	15.67	7.80	25.5	31	67	31	5.39	18.33	"	"
9	"	10.64	5.41	27.0	31	79	31	5.44	18.50	"	"
10	Bp2S30	13.68	10.7	37.5	47	57	77	2.93	9.97	cold	"
11	"	14.04	12.55	36	45	57	75	2.95	10.03	hot	"
12	"	15.91	17.40	36	41	80	71	3.03	10.30	hot	"

water content of the pans 2 x 2 kg.

Run no.	type of test	efficiency pan 1 (%)	efficiency pan 2 (%)	boiling time pan 1 (min)	boiling time pan 2 (min)	temp. pan 2 (^{o}C)	total test time (min)	$\dot{Q}$ (kW)	burning rate (g/min)	stove condition	MC (%)
30	Bp2	6.41	6.39	17	25.5	49	25.5	5.86	18.72	cold	6
31	"	8.63	7.18	14.5	21	56	21.0	5.47	17.49	hot	6
32	"	7.84	6.86	12	17	60	17	7.61	24.32	hot	6

Table 5.4 The effect of moisture content of the wood fuel

White Fir

Run no	wood charge [-]	Δ_t [min]	$\dot{Q}$ [kW]	efficiency [%]	CO_2 [%]	CO [%]	O_2 [%]	T. [°C]	Excess air factor [-]	moisture content [%]
7.	2	10	3.4	22.4	8.16	0.78	12.01	205	2.42	oven dry
4.	2	6.5	5.02	19.5	9.23	0.87	10.19	210	2.06	oven dry
3.	1	8	2.1	20.8	4.3	0.28	16.7	157	4.62	oven dry
10	2	10	3.16	21.47	9.03	0.5	11.45	228	2.23	6
11	2	6.5	4.07	21.35	10.49	1.21	9.41	227	1.85	"
12	1	8	2.13	21.30	4.5	0.33	16.69	160	4.7	"

Table 5.5 TNO/TH - experiments

Yelutong

Run no	type of test	$\dot{Q}$ [kW]	efficiency [%]	CO_2 [%]	CO [%]	O_2 [%]	T. [°C]	Excess air factor [-]	moisture content [%]
22	Bp 2	3.28	20.66	5.45	0.36	15.11	107	3.74	13
23	"	4.53	21.16	8.07	0.48	12.04	190	2.53	"
24	"	4.75	21.46	10.23	0.71	9.95	202	1.98	"
30	Bp 2	5.86	12.80	8.57	1.3	12.29	214	2.45	6
31	"	5.47	15.81	11.56	1.22	9.13	268	1.81	"
32	"	7.61	14.70	9.18	1.28	11.92	274	2.3	"

Table 5.6 Combustion performance of the Tungku Lowon stove as a function of the heat output of the fire, the wood pieces per charge.

Symbols :

m.c.	-moisture content of wood	[%]
n_i	-number of the wood pieces	[1]
$\dot{Q}$	-heat output of the fire	[kW]
T_g	-flue gas temp.	[^oC]

TNO/TH - experiments

kind of wood: white fir

initial amount of water
pan 1: 4 kg; pan 2: 2 kg.

run no.	m.c. [%]	n_i [1]	$\dot{Q}$ [kW]	Flue gas composition						excess air factor
				CO_2 [%]	CO [%]	O_2 [%]	C_xH_y [ppm]	Soot [$\frac{mg}{nm^3}$]	T_g [^oC]	
3	oven dry	1	2.1	4.3	0.28	16.7	650	-	157.	4.62
8	"	1	2.1	4.35	0.28	16.86	650	-	157.76	4.60
7	"	2	3.30	8.16	0.78	12.01	2400	-	205.	2.42
6	"	2	3.71	8.02	0.79	11.58	2400	-	196	2.36
1	"	2	4.20	10.62	0.83	9.85	2600	-	250.	1.90
4	"	2	5.02	9.23	0.87	10.19	2700	-	210.	2.06
2	"	2	6.88	12	1.81	7.34	7000	-	213.	1.60
9	"	2	2.2	5.74	0.43	15.06	950	-	149	4.42
10	6	2	3.16	9.03	0.5	11.45	1300	-	228	2.23
11	"	2	4.07	10.49	1.21	9.41	4500	-	227	1.85
12	"	1	2.13	4.5	0.33	16.69	700	-	160	4.70

Table 5.7 Combustion performance of the Tungku Lowon wood stove according the ITDG procedure as a function of the heat output of the fire.

TH/TNO - experiments

Symbols:
m.c. -moisture content of the wood [%]
n_i -number of the wood sticks [1]
$\dot{Q}$ -heat output of the fire [kW]
T_g -flue gas temp. [^oC]

kind of wood : yelutong
initial amount of water
pan 1: 2 kg; pan 2: 2 kg.

run no.	m.c. [%]	n_i [1]	$\dot{Q}$ [kW]	Flue gas composition						excess air factor
				CO_2 [%]	CO [%]	O_2 [%]	C_xH_y [ppm]	Soot [mg/nm^3]	T_g [^oC]	
26	13	5	1.78	2.94	0.16	18.12	380.	-	61	6.97
28	"	"	4.04	5.96	0.32	14.8	775.	-	161	3.3
22	"	"	3.39	5.45	0.36	15.11	820.	-	107	3.74
23	"	"	4.54	8.07	0.48	12.04	1200.	-	190	2.53
24	"	"	4.77	10.23	0.71	9.95	2150.	-	202	1.98
15	"	"	3.15	-	-	-	-	-	192	-
16	"	"	3.28	-	-	-	-	-	196	-
25	"	"	3.51	6.79	0.49	13.39	1250.	-	195	2.99
27	"	"	2.99	6.13	0.37	14.78	975.	-	158	3.33
29	"	"	3.31	7.11	0.43	13.51	1025.	-	213	2.87
30	6	5	5.86	8.57	1.3	12.29	-	-	214	2.45
31	"	"	5.47	11.56	1.22	9.13	-	-	268	1.81
32	"	"	7.61	9.18	1.28	11.92	-	-	274	2.3

Table 5.8 Heat balance of the Tungku Lowon wood stove for the test period as a function of the heat output of the fire and the wood pieces per charge.

TNO/TH-experiments

kind of wood: white Fir

initial amount of water;

pan 1: 4 kg,pan 2: 2 kg.

run no	moisture content (%)	wood charge (pieces)	heat output (kW)	first pan (%)	second pan (%)	sensible heat in flue gas (%)	unburnt constituents (%)		Rad. en conv. to surroundings (%)	accumulated heat (%)	unaccounted for (%)	accumulated heat + unaccounted for (%)	remarks
							CO	C_xH_y					
3	0	1	2.1	15.8	5	20.8	4	2.4	13.1	-	-	38.9	
8	"	1	2.1	16.1	4.	20.1	3.9	2.5	12.6	41.9	-1.1.		
5	"	2	3.43	15.64	5.9	33.5	4.		10.3	49.			not reliable CO_2 measurement
7	"	2	3.4	16.9	5.5	14.4	5.6	5.	10.3	35.9	6.4		
6	"	2	3.71	15	5.5	13.6	5.7	4.8	8			47.4	
1	"	2	4.2	15.1	5.5	15	4.6	4.1	7.7	54.	-6.		
4	"	2	5.02	13.9	5.6	13	5.5	4.8	6.6	39.6	11.		
2	"	2	6.88	13.4	4.5	10.4	8.4	9	4.4	44.1	5.8		
9	"	2	2.2	15.85	4.83	14.8	4.5	2.7	7.8			49.52	
10	6	2	3.16	15.65	5.82	18.73	4.1	2.4	8.2	37.1	8.		
11	"	2	4.07	15.04	6.31	18.83	9.5	6.8	4.4			39.12	
12	"	1	2.13	17.1	4.20	23.71	5.1	2.5	12.2	-		35.2	

Table 5.9 Comparison of the TH/TNO and ITDG experiments according the TH/TNO procedure.

water content of the pans: 4 x 2 kg.

kind of wood : White Fir - oven dry.

ITDG Experiments (Reading)

$\dot{Q}$ [kW]	Δ_t [min]	η [%]	Flue gas composition[%]			temp. [°C]	excess air factor
			CO_2	O_2	CO		
2.7	14	18.9	2.4	18.1	0.11	-	8.3
3.42	12	19.6	5.7	16.9	0.17	-	3.6
4.41	10	20.4	4.7	18.3	0.19	-	4.3
4.13	8	21.5	-	-	-		
4.92	8	21.9	3.2	18.5	0.21	-	6.1
5.62	7	22.6	5.8	16.5	0.28	-	3.5
6.03	6	18	13.8	7.5	1.66	-	1

TH/TNO Experiments (Apeldoorn)

$\dot{Q}$ [kW]	Δ_t [min]	η [%]	Flue gas analysis [%]			temp [°C]	excess air factor
			CO_2	O_2	CO		
2.1	8	20.8	4.3	16.7	0.28	157	4.62
2.1	8	20.1	4.35	16.86	0.28	157	4.60
2.2	15	20.7	5.74	15.06	0.43	149	4.42
3.4	10	22.4	8.16	12.01	0.78	205	2.42
3.7	9	20.5	8.02	11.58	0.79	196	2.36
4.2	8	20.6	10.62	9.85	0.83	250	1.90
5.02	6.5	19.5	9.23	10.19	0.87	210	2.06
6.88	5	17.9	12.0	7.34	1.81	213	1.6

Table 5.9a ITDG experiments according the TH/TNO test procedure.

Water contents of the pans : 4 x 2 kg.
kind of wood : white fir - oven dry.

ITDG Experiments (Reading)

$\dot{Q}$ [KW]	Bp 1 [min]	Efficiency [%]		
		pan 1	pan 2	total
2.5	54	14.1	4.8	18.9
3.14	54	15.25	5.55	21.1
3.42	45	13.4	5.7	19.6
4.13	30	15.7	5.8	21.5
4.14	40	14.7	5.7	20.4
4.76	34	13.8	5.7	19.5
4.92	34	15.9	6.0	21.9
5.62	32	16.5	6.1	22.6
6.03	31	11.8	6.3	18.1

Table 5.10 Comparison of the ITDG and TH/TNO Experiments according the ITDG procedure.

Water content of the pans: 2 x 2 kg. Kind of wood: yelutong

moisture content : ∿ 13 %

ITDG Experiments (Reading)

Test no	Type of test	Efficiency pHU 1 [%]	Efficiency pHU 2 [%]	Boiling time pan 1 [min]	Boiling time pan 2 [min]	Temp. pan 2 Bp 1 [°C]	Total test time [min]	Burning rate [g/min]	Stove condition [-]
1	Bp 1		18.1	17	-	61	17	19	-
2	"		17.9	12.1	-	66	12.1	29.4	-
3	"		17.8	12.5	-	59	12.5	26.2	-
4	"		16.2	15.5	-	60	15.5	22.6	-
5	"		20.2	16.8	-	55	16.8	17.2	-
6	"		19.7	19.0	-	62	19.0	18.3	-
7	Bp 2		22.0	15	23	55	23.0	18.5	-
8	"		21.8	16	25	51	25.0	17.4	-
9	"		21.9	15	21	61	21.0	18.2	-
10	Bp2S30		20.8	14.8	22.8	49.	52.8	12.8	-

TH/TNO Experiments (Apeldoorn)

18	Bp 1		18.17	31	-	34	31	8.45	cold
19	"		17.26	22	-	37	22	12.17	hot
20	"		15.35	21.5	-	41	21.5	13.92	"
21	"		15.36	23.5	-	45	23.5	13.37	"
26	"		19.15	38.0	-	30	38	6.03	cold
28	"		15.85	22.0	-	50	22	13.71	hot
22	Bp 2		20.66	27	41.5	35	41.5	11.51	cold
23	"		21.16	20	30	49	30	15.41	hot
24	"		21.46	17	28	46	28	16.17	hot
15	Bp2S30		21.03	21.5	36.5	38	66.5	10.69	cold
16			24.05	20.5	30.	52	60	11.13	hot
25			23.65	17	28.	45	58	11.89	"
27			22.45	22	36.	34	66	10.13	"
29			24.82	17	27.	46	57	11.24	"

Table 5.11 Average values of the ITDG and TH/TNO

Experiments according the ITDG procedure

(with yelutong wood).

	ITDG Experiments				TNO/TH Experiments			
Type of test	Efficiency (%)	Test time (min)	Burning rate (g/min)	Fuel used gram	Efficiency (%)	Test time (min)	Burning rate (g/min)	Fuel used gram
Bp 1	18.23	15.5	22	341	17	26	11.3	294
Bp 2	22	23	18	414	21	32	14.4	461
Bp2S30*	21	53	13	689	23	62	11	682

* one measurement

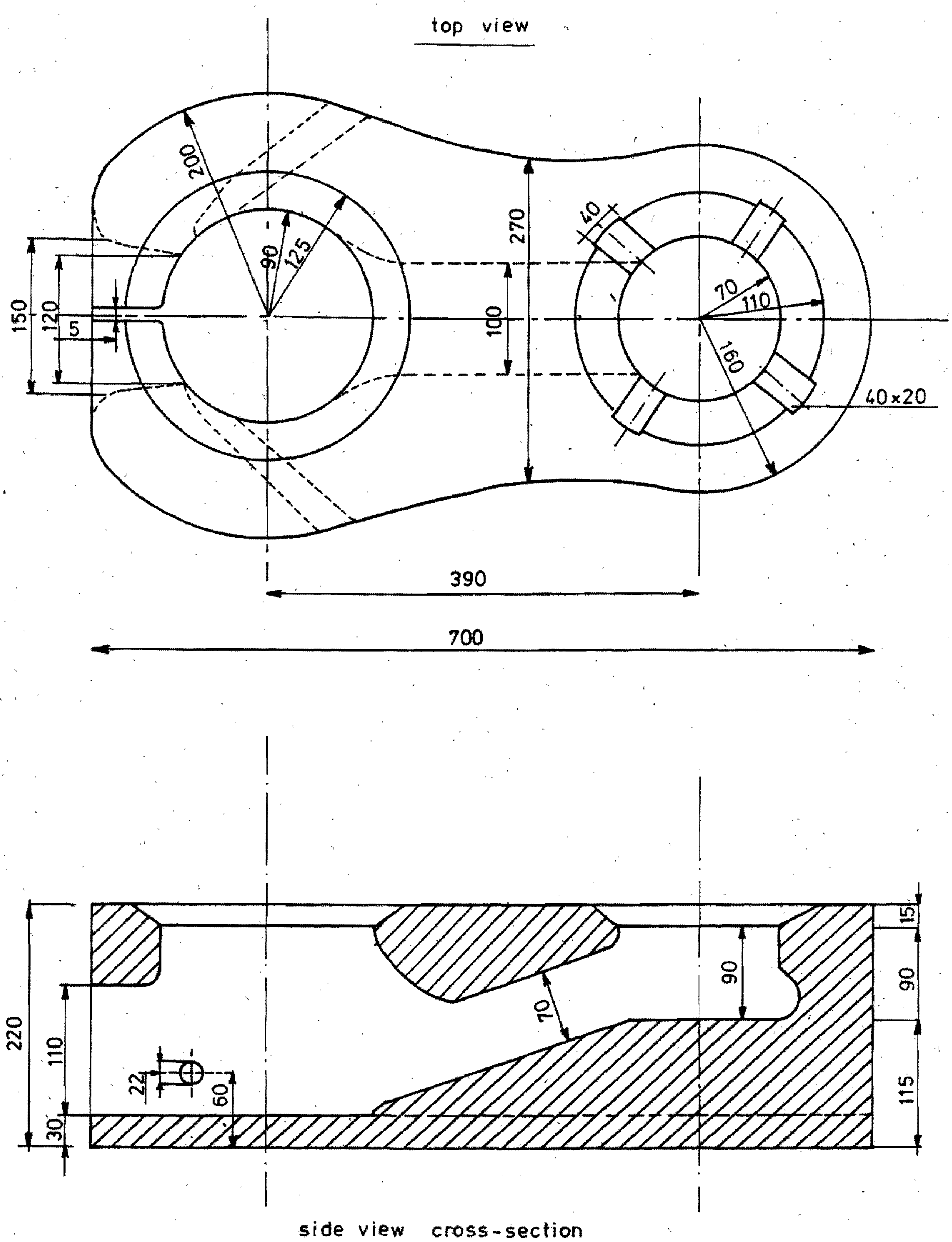
Table 5.12 Combustion performance of the Tungku Lowon stove as a function of the Burning rate.
Kind of wood: yelutong (13% m.c.).

TH/TNO Experiments (Apeldoorn)

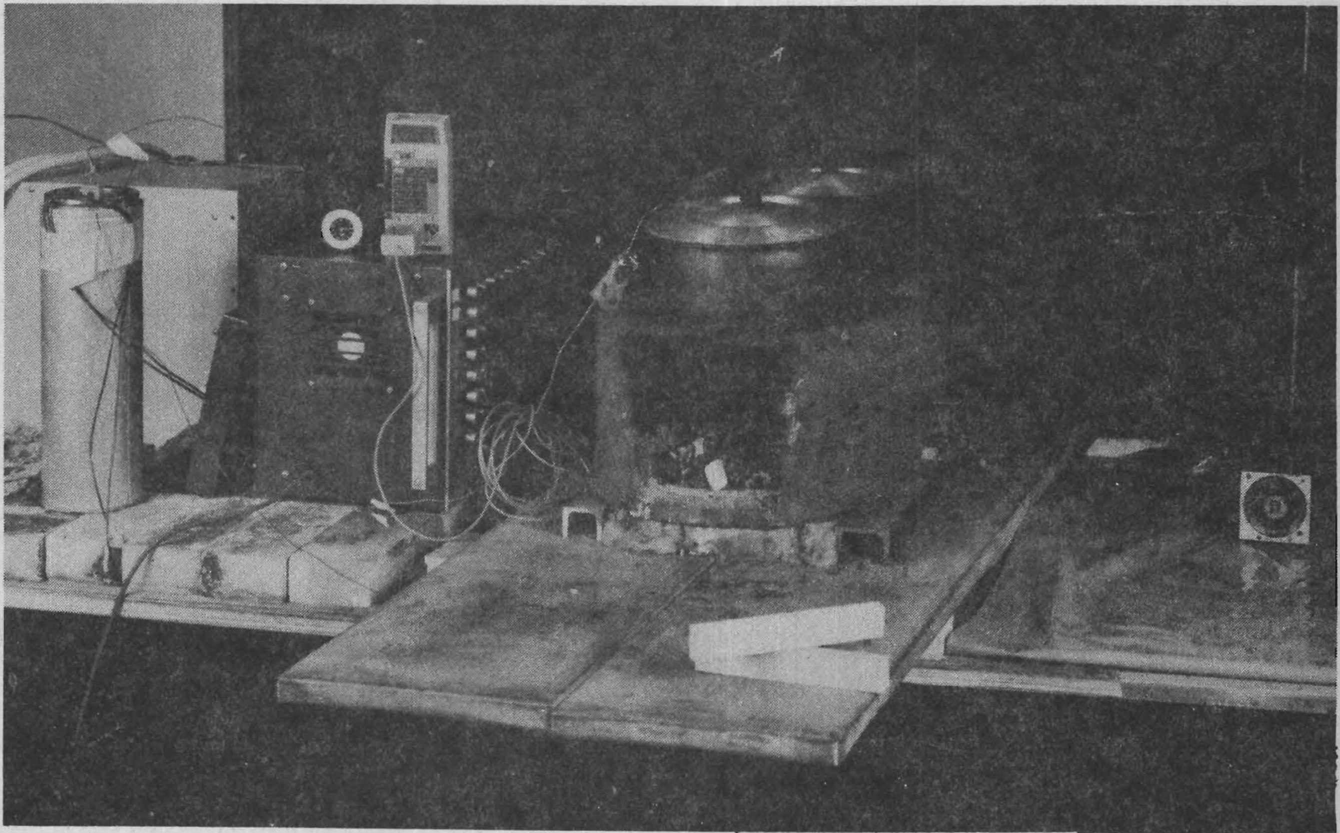
Run no	Type of test	Burning rate [g/min]	Flue gas composition CO_2 [%]	CO [%]	O_2 [%]	Stack temp [°C]	stove condition [-]	excess air factor [-]
26	Bp 1	6.035	2.94	0.16	18.12	61	cold	6.97
28	"	13.71	5.96	0.32	14.8	161	hot	3.3
22	Bp 2	11.51	5.45	0.36	15.11	107	cold	3.74
23	"	15.41	8.07	0.48	12.04	190	hot	2.53
24	"	16.17	10.23	0.71	9.95	202	hot	1.98
15	Bp2S30	10.69				192	cold	
16		11.13				196	hot	
25		11.89	6.79	0.49	13.39	195	"	2.99
27		10.13	6.13	0.37	14.78	158	"	3.33
29		11.24	7.11	0.43	13.51	213	"	2.87

ITDG Experiments (Reading)

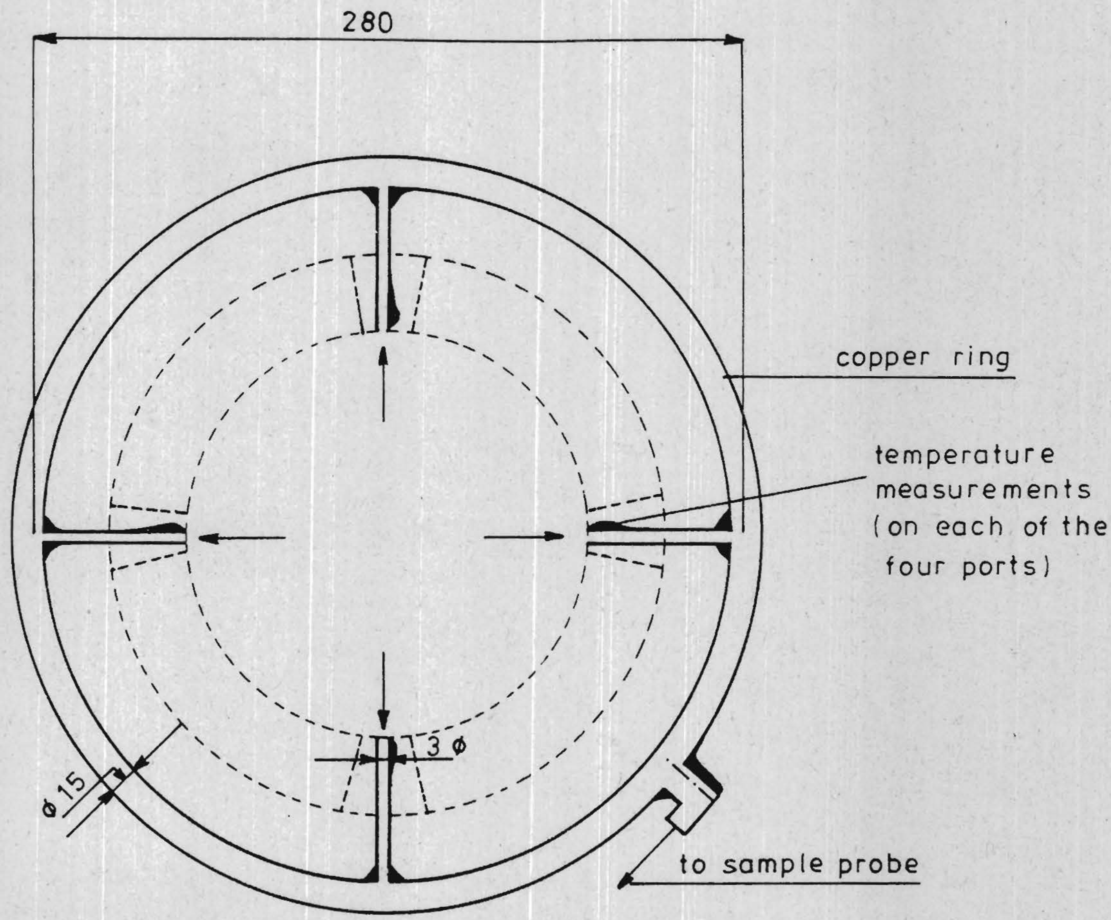
Run no	Type of test	Burning rate [g/min]	CO_2 [%]	CO [%]	O_2 [%]	Stack temp [°C]	stove condition [-]	excess air factor [-]
1	Bp 1	19	9.6	1	12.1	169	-	2.1
2	"	29.4	11.5	1.2	17.2	248	-	1.73
3	"	26.2	9.5	1.1	16.7	187	-	2.44
4	"	22.6	5.4	1.3	15.8	174	-	4.22
7	Bp 2	18.5	6	1.2	20.1	230	-	3.17
8	"	17.4	5.9	0.8	19	172	-	4.06
9	"	18.2	6.2	1.2	16.6	282	-	3.08
10	Bp2S30	12.8	6.1	0.1	16.6	185.	-	3.43



The Tungku Lowon Stove scale 1:5	MT—TNO 84940 Fig. 5·1



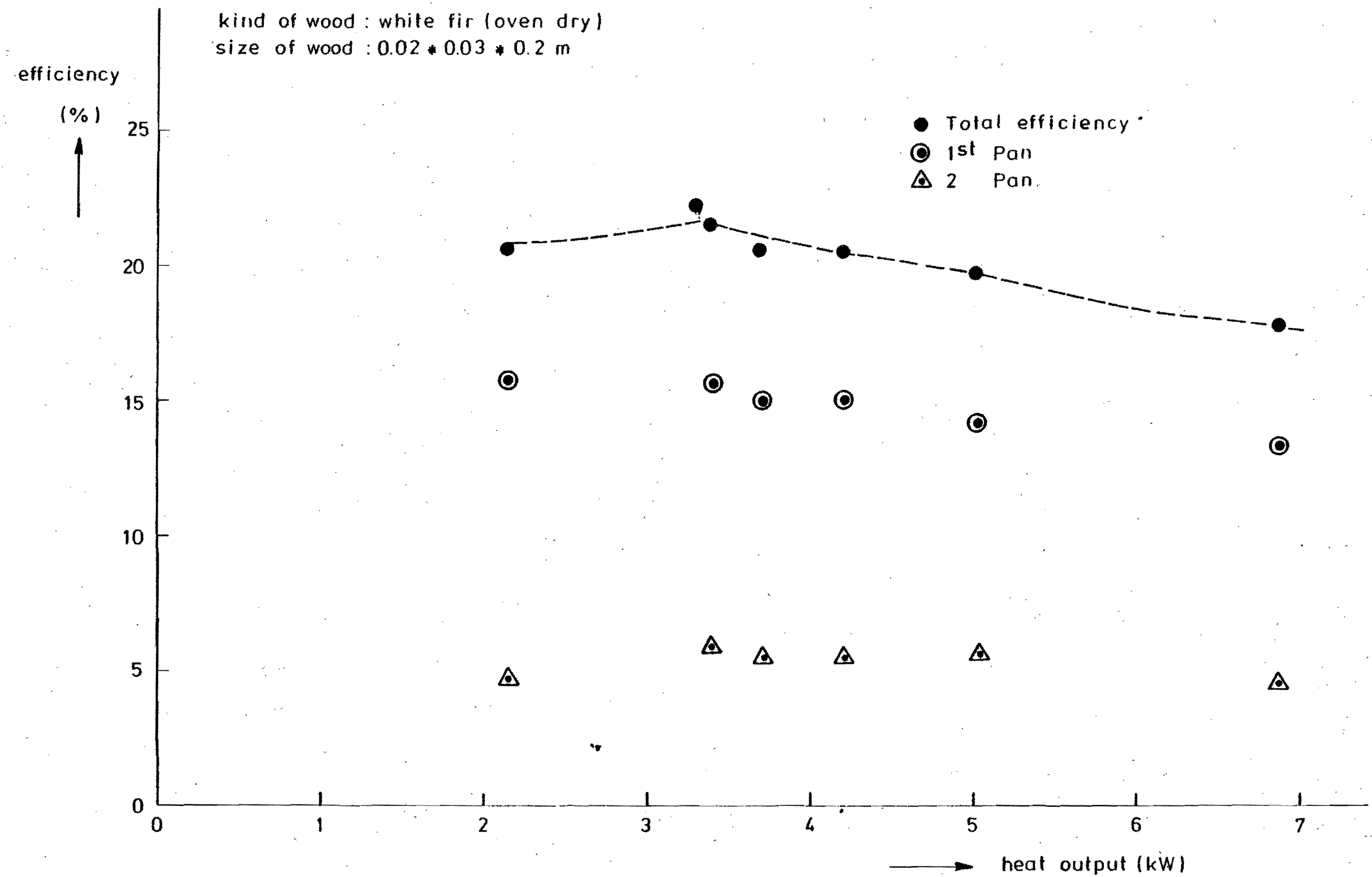
Experimental set-up of the Tungku Lowon wood Stove	MT-TNO 84940 Fig. 5.2



Gas analysis arrangement

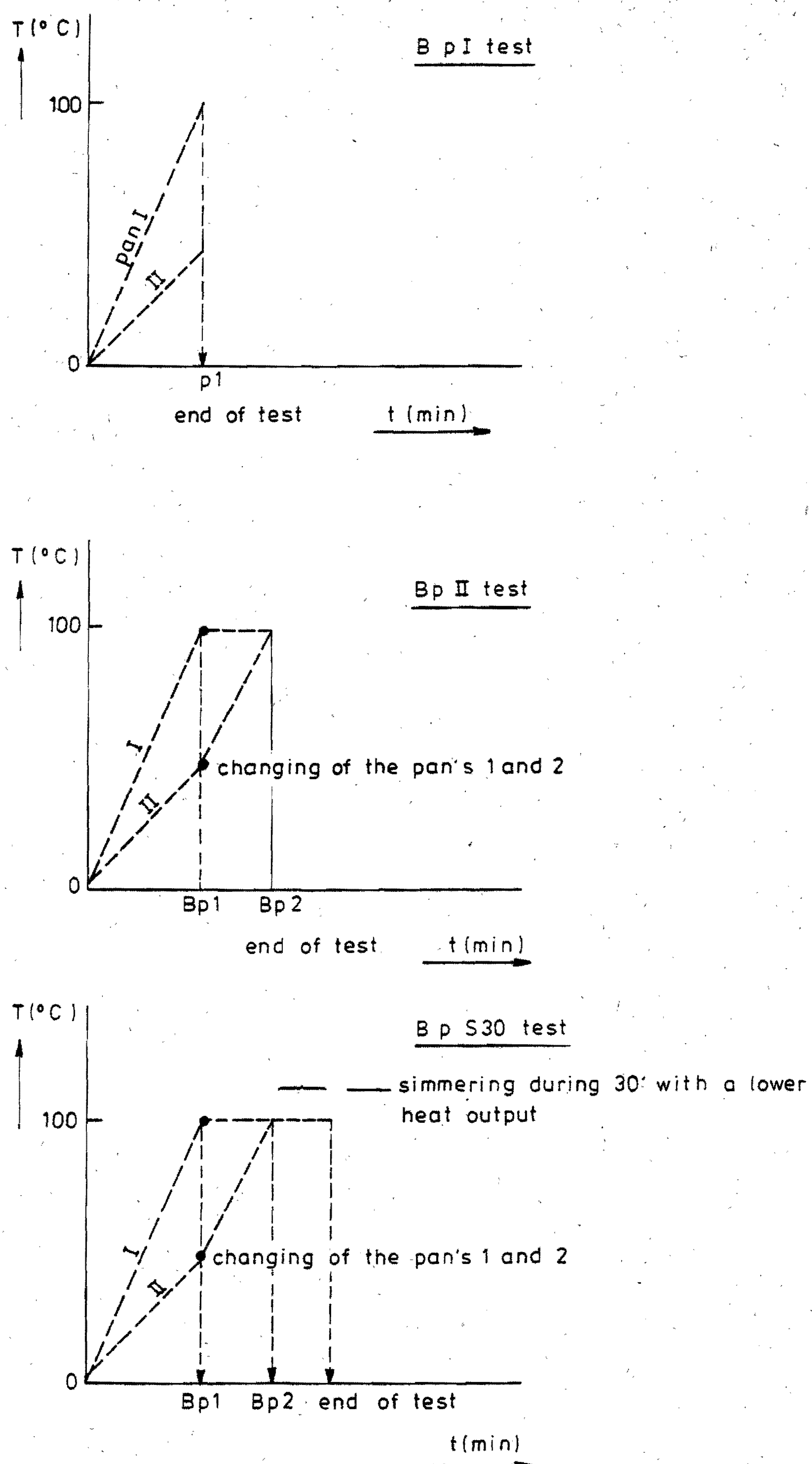
scale 1 : 3

MT-TNO
84940
Fig. 5.3

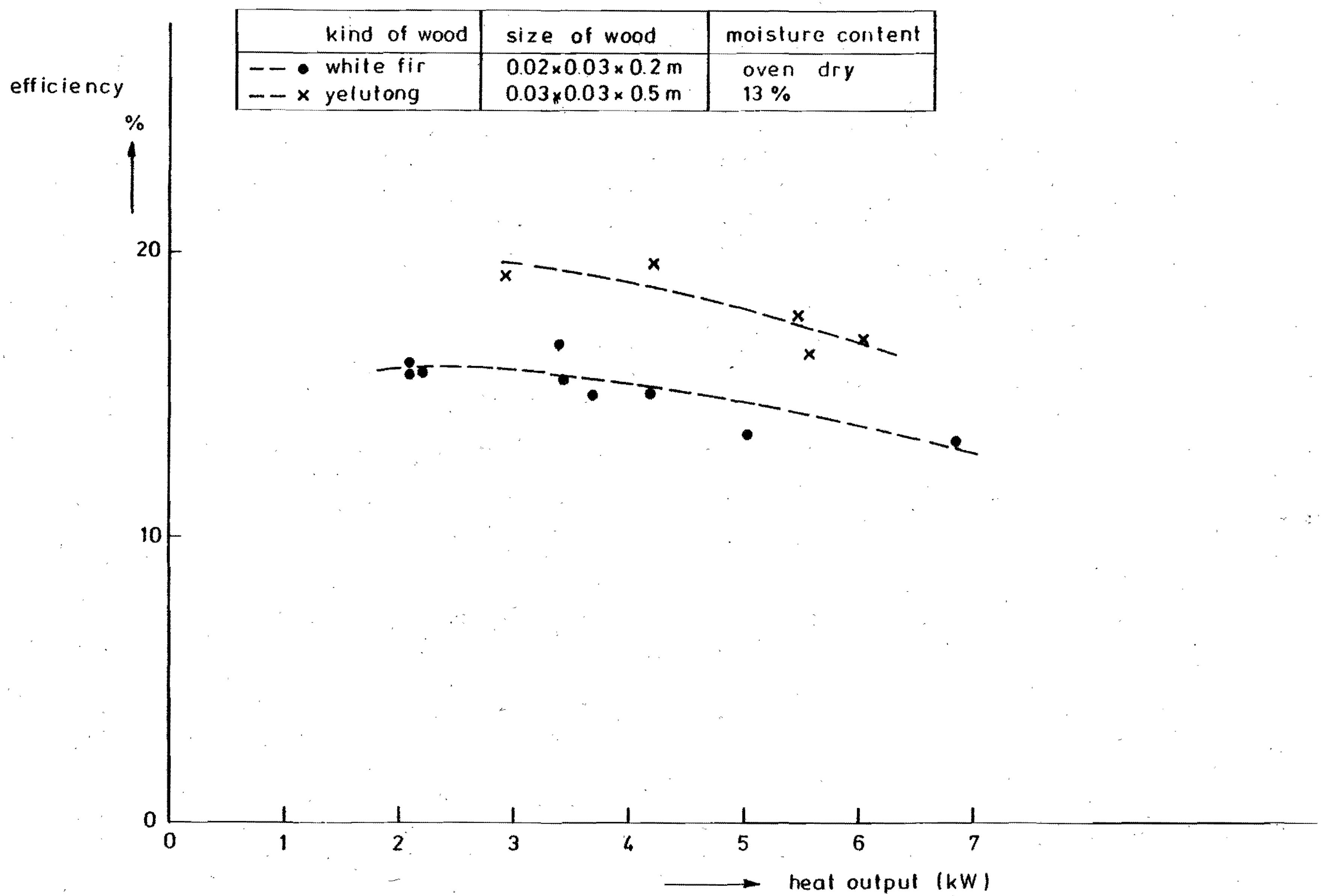


Efficiency of the Tungku Lowon as a function of the heat output

MT-TNO
84940
Fig. 5.4



Schematic time temperature diagram of ITDG tests	MT-TNO 84940 Fig. 5.5

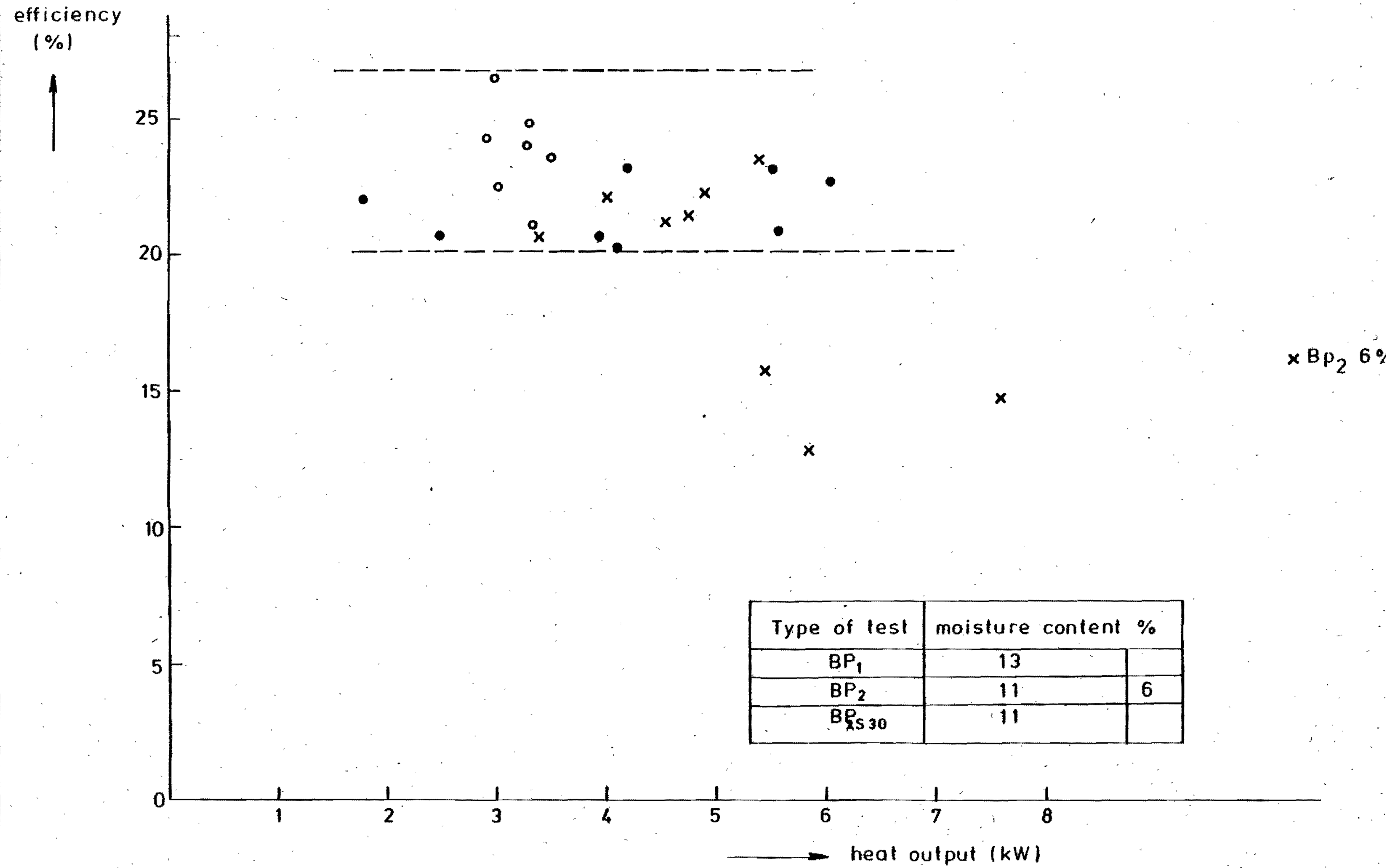


Efficiency of first pan for different firing procedures with 4 kg of water in first pan

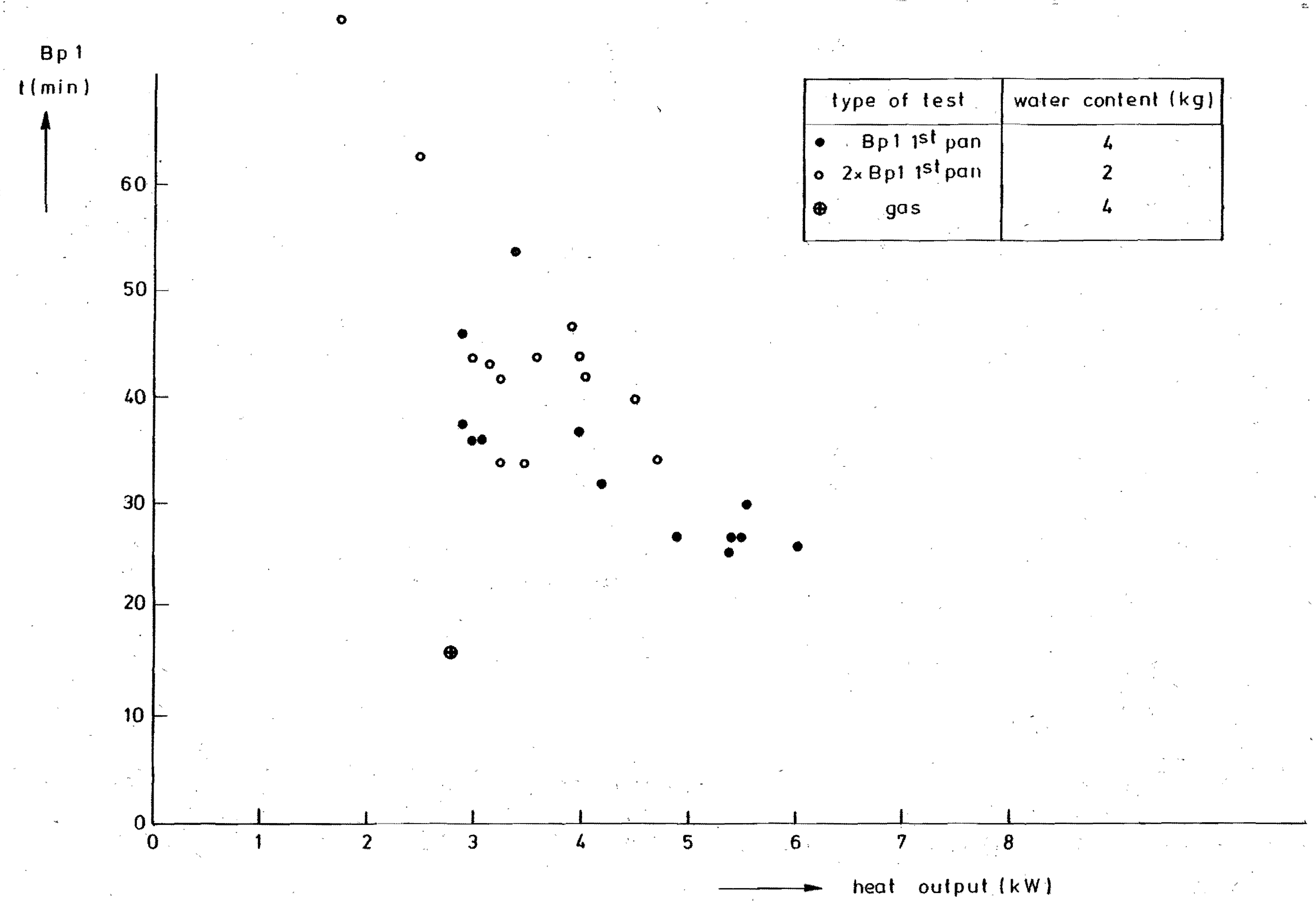
MT-TNO 84940
Fig. 5.6

Average efficiency of the Tungku Lowon Stove when fired with yelutong as a function of the heat load

MT-TNO
84.940
Fig. 5.7

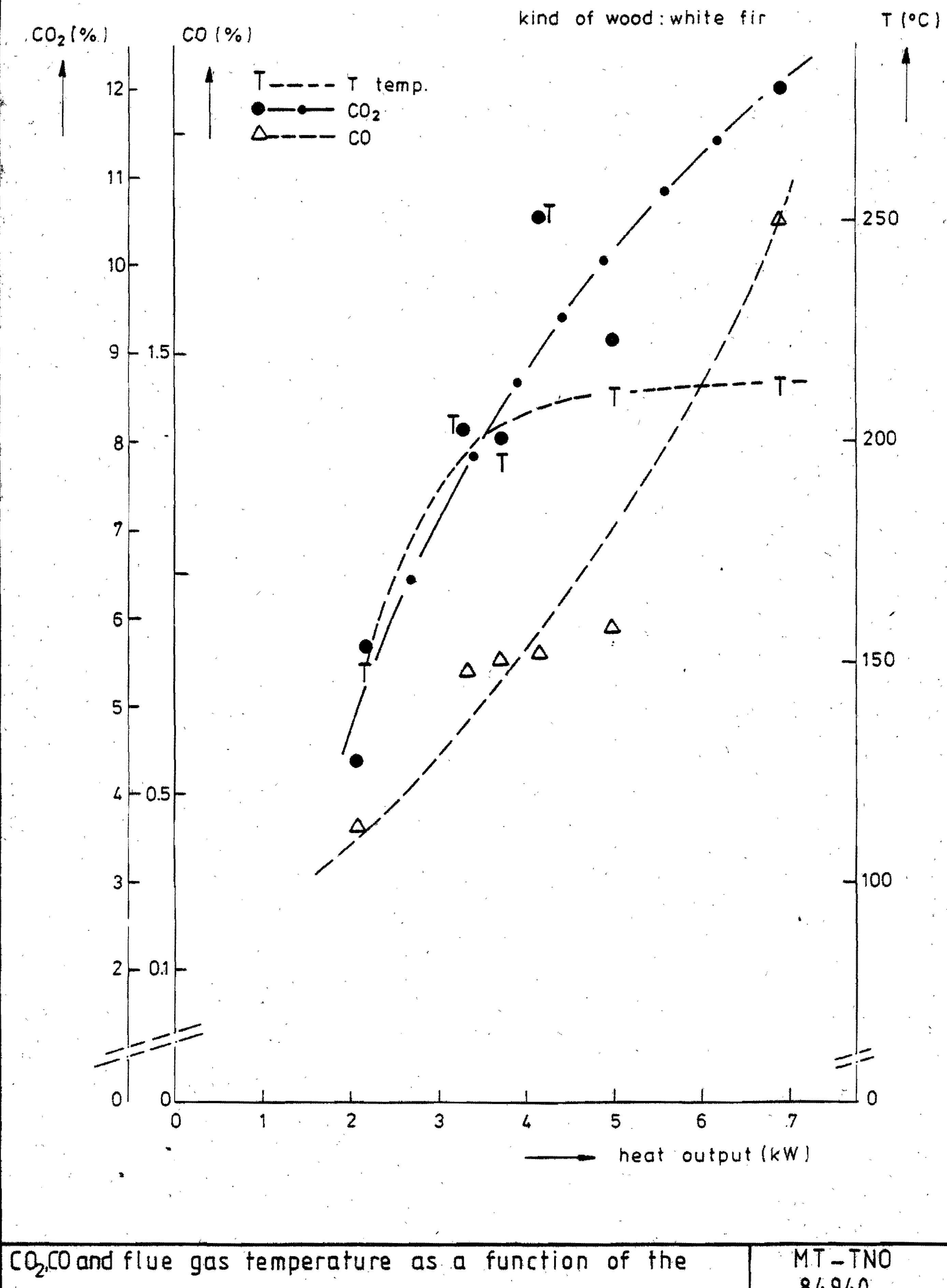


Type of test	moisture content %	
BP_1	13	
BP_2	11	6
BP_{AS30}	11	

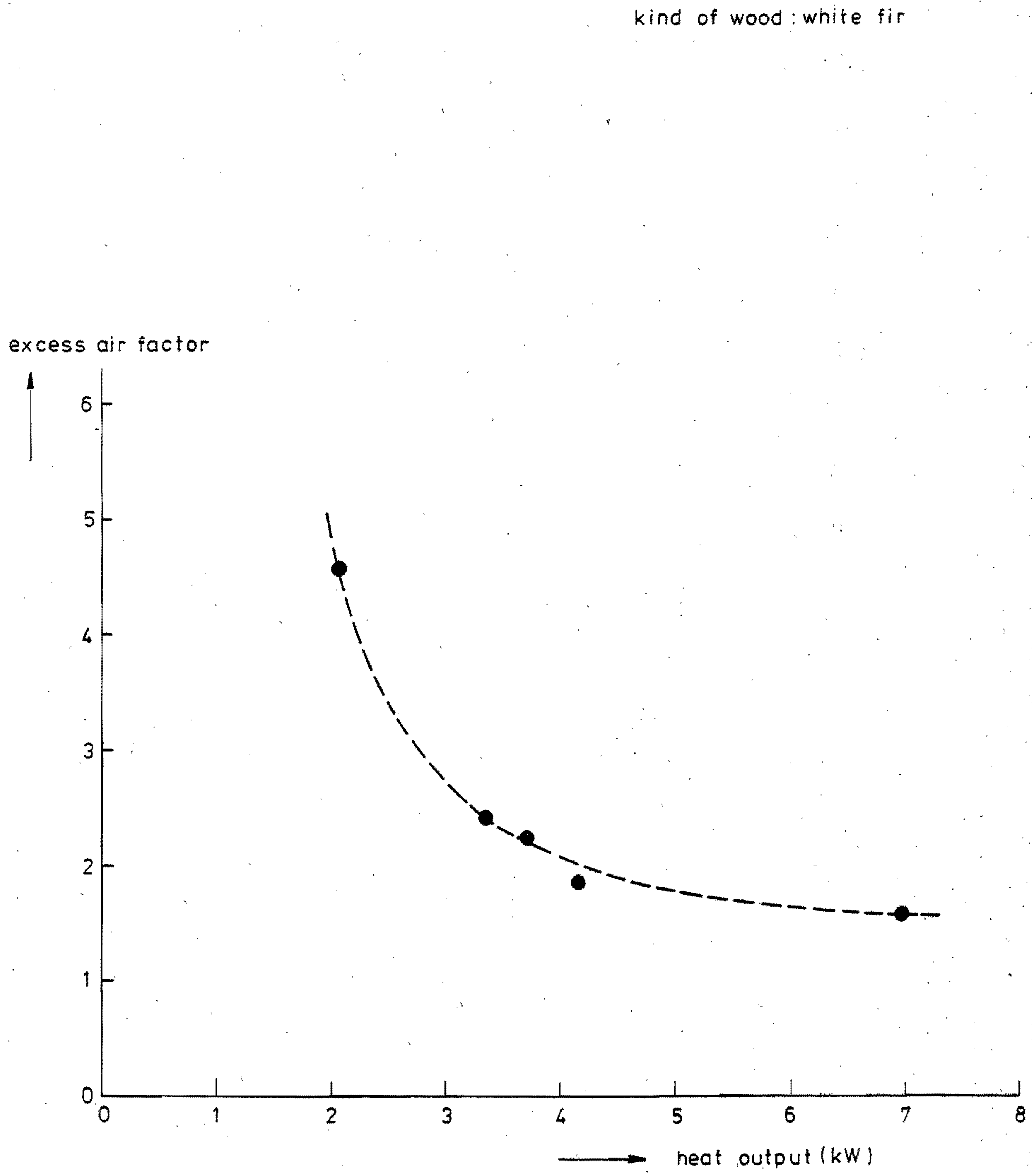


BP1 comparison for different water contents of the first pan

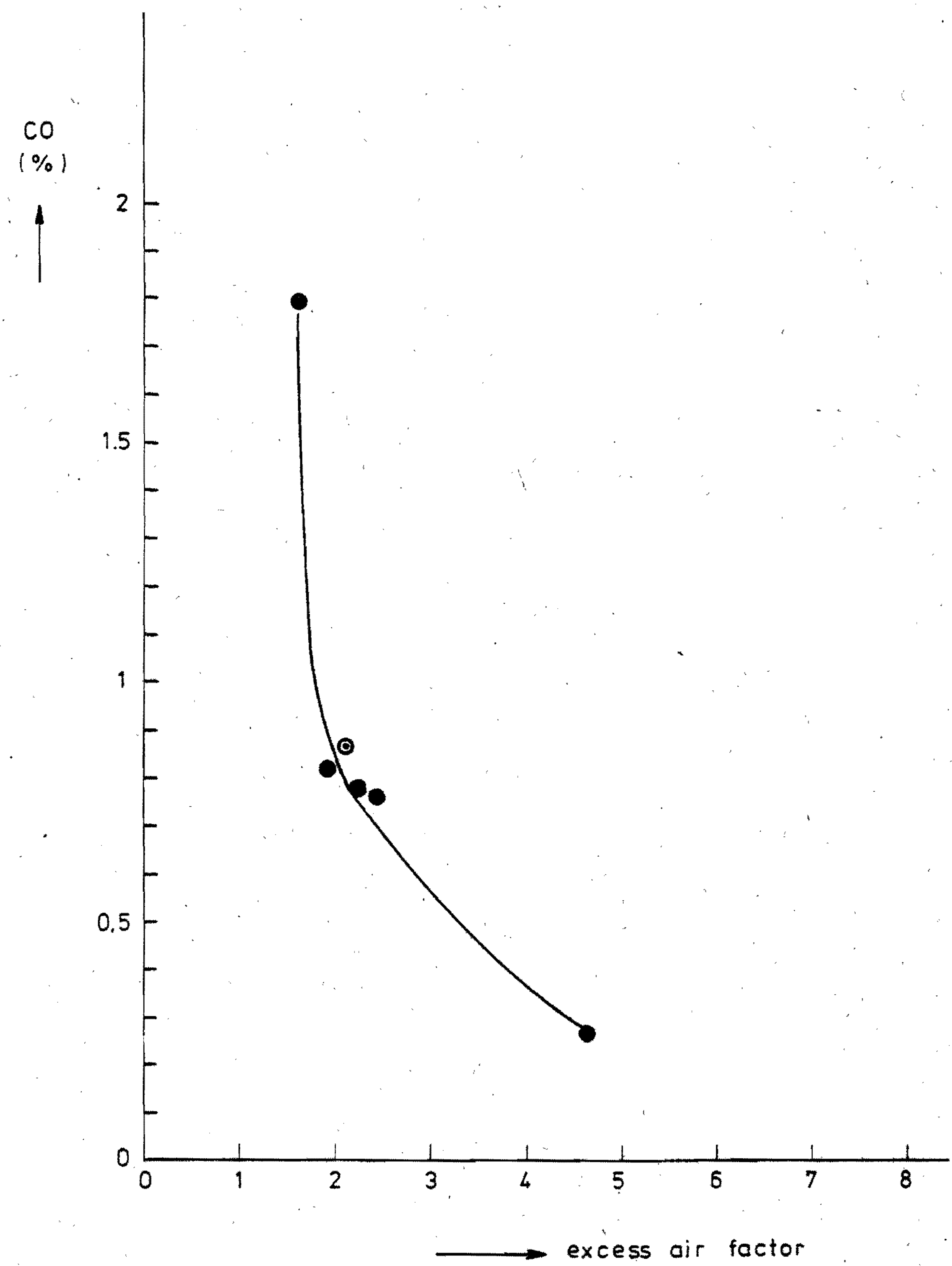
MT-TNO
84940
Fig. 5.8



CO_2, CO and flue gas temperature as a function of the heat output	MT-TNO 84940 Fig. 5.9

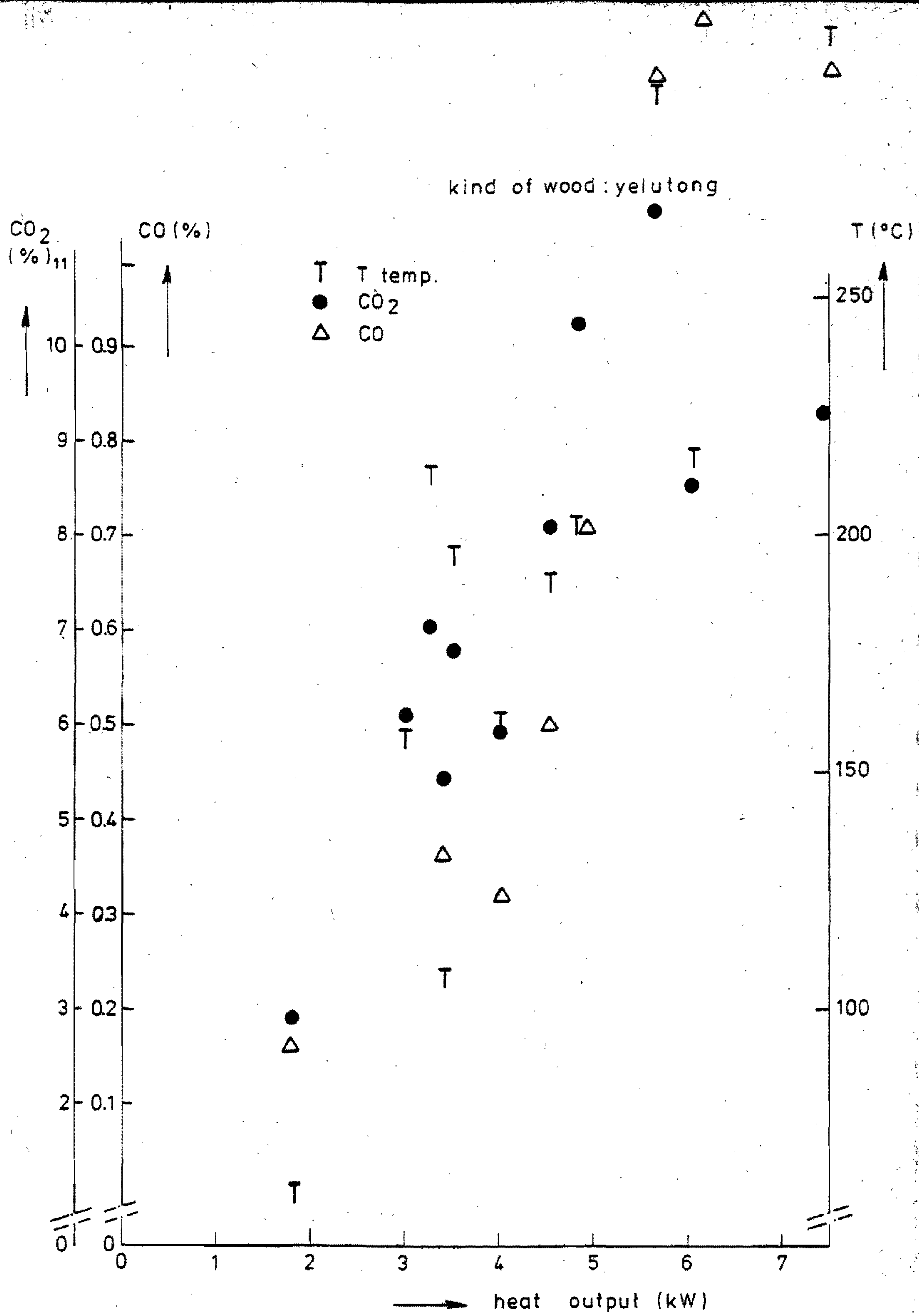


Excess air factor as a function of the heat output	MT-TNO 84940 Fig. 5-10

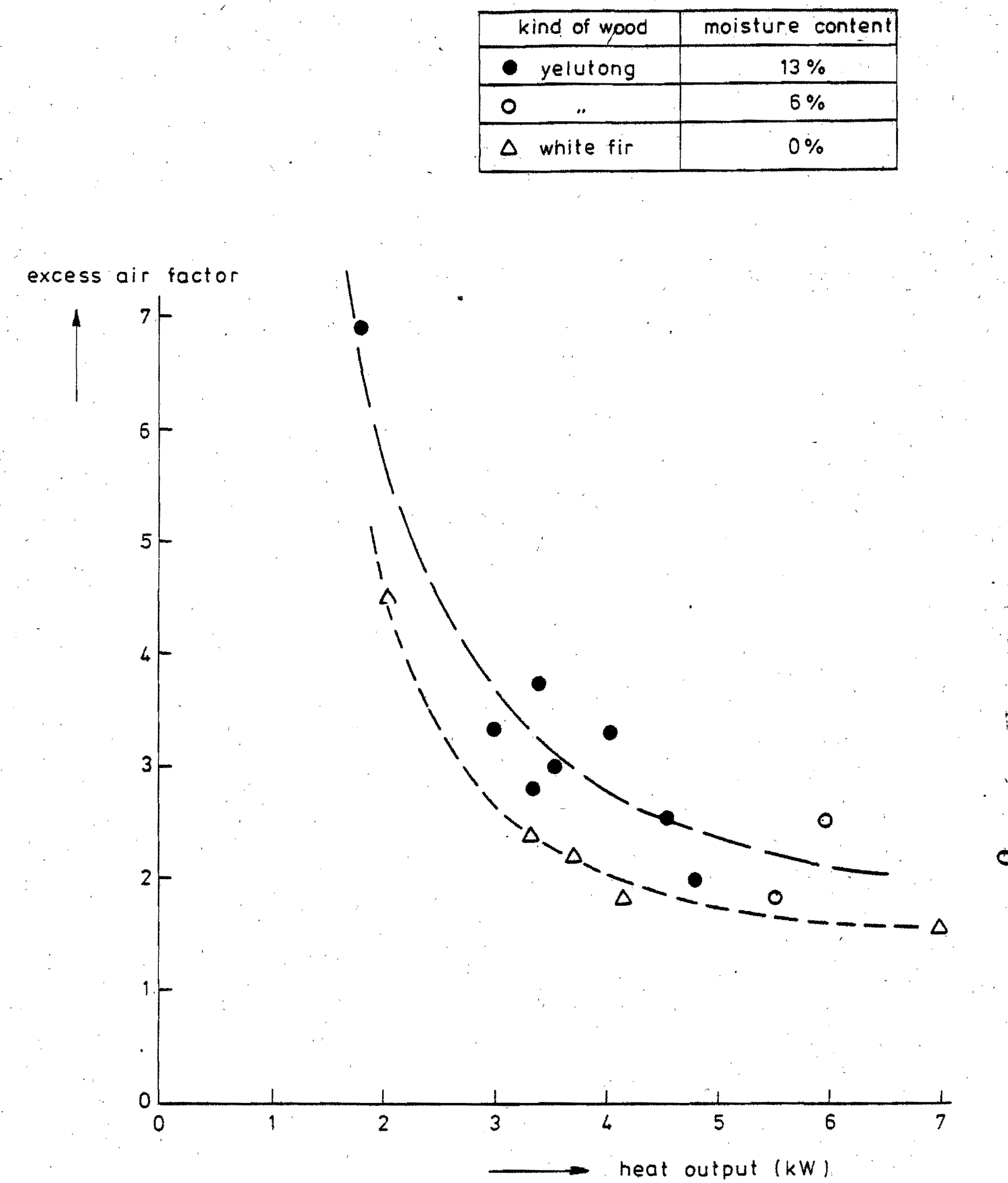


CO as a function of the excess air factor

MT-TNO
84940
Fig. 5-11

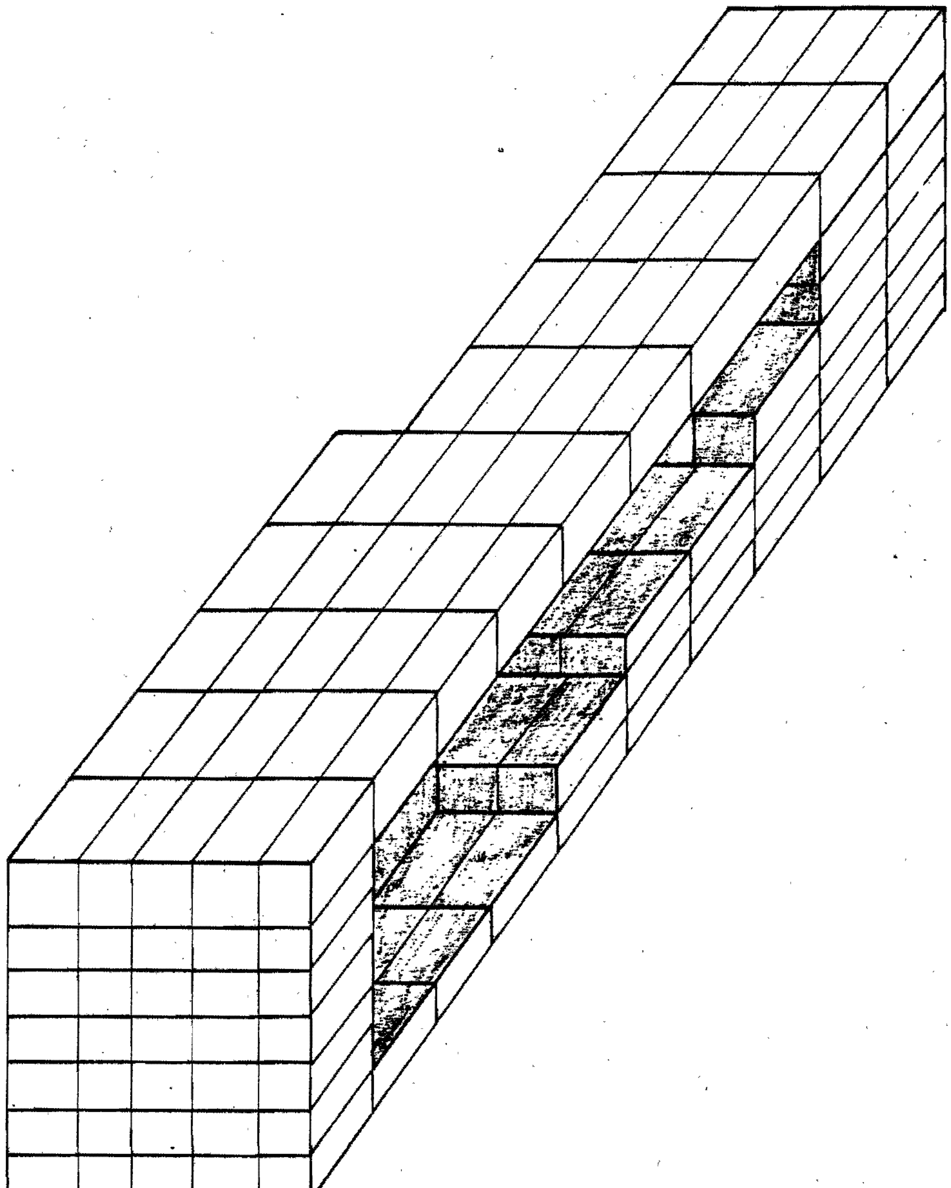


CO_2 CO, and flue gas temperature as a function of the heat output	MT—TNO 84940 Fig. 5·12

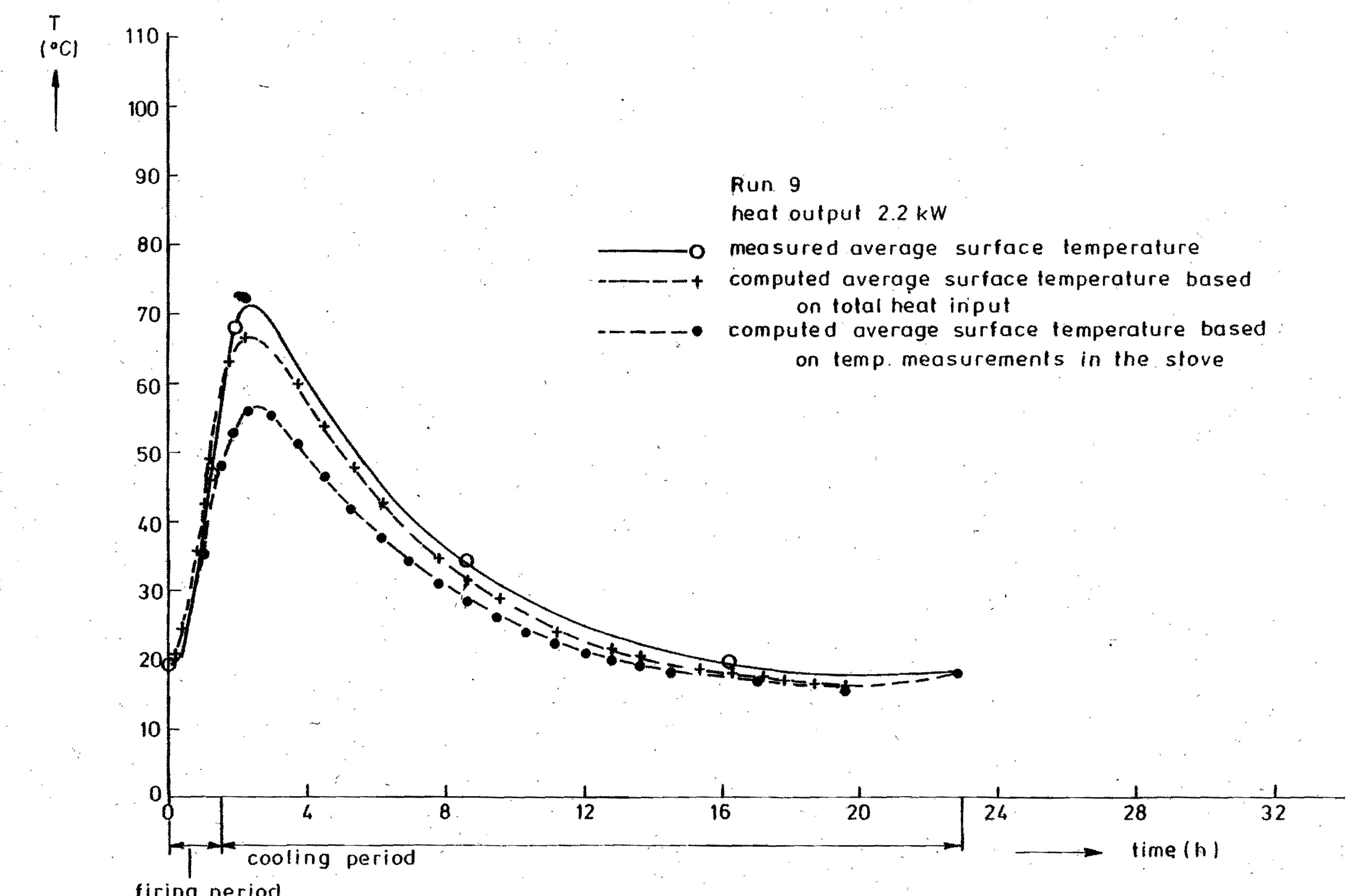


Excess air factor as a function of the heat output

MT-TNO
84940
Fig. 5·12a

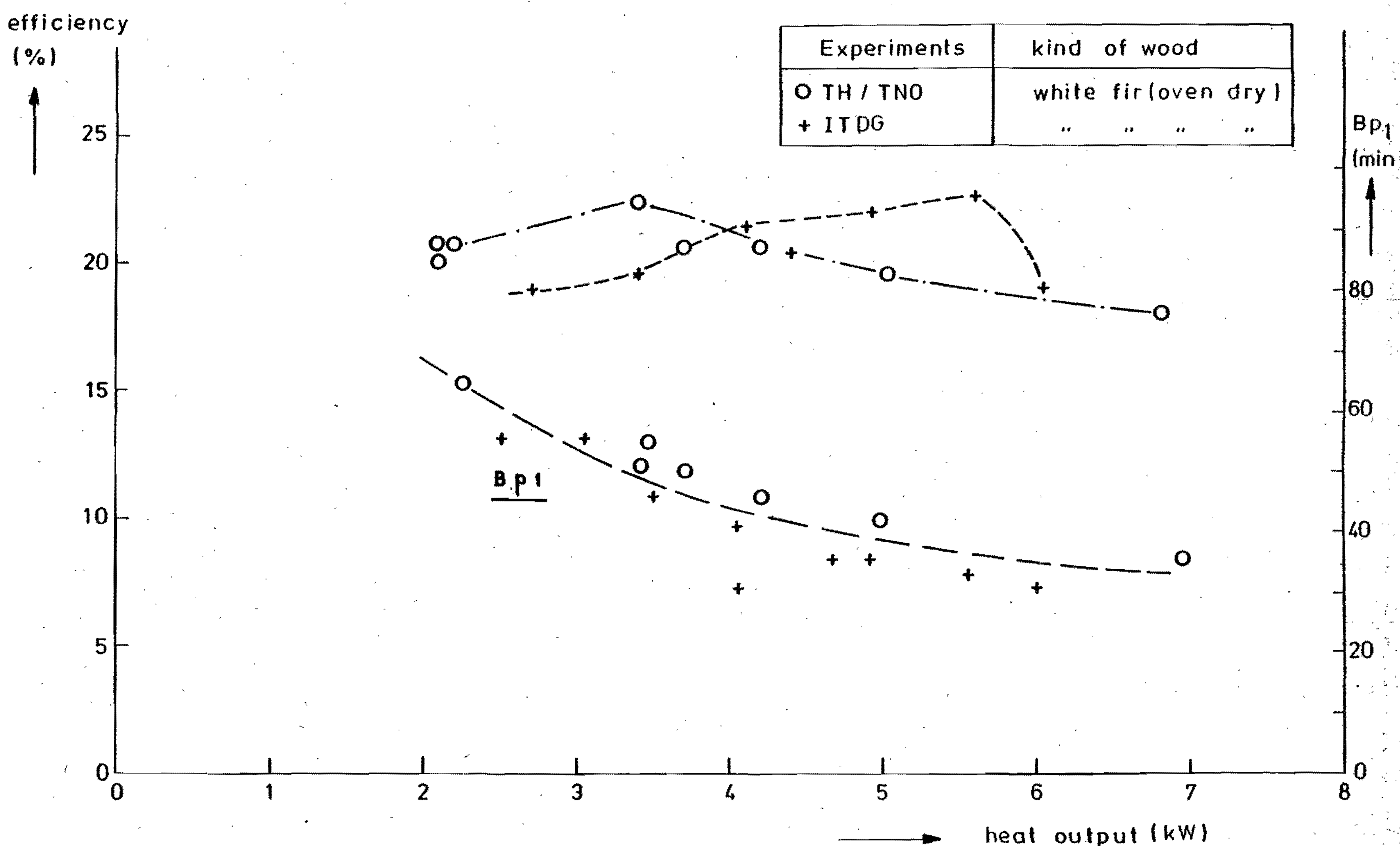


Model of stove with element division	MT–TNO 84940 Fig. 5·13



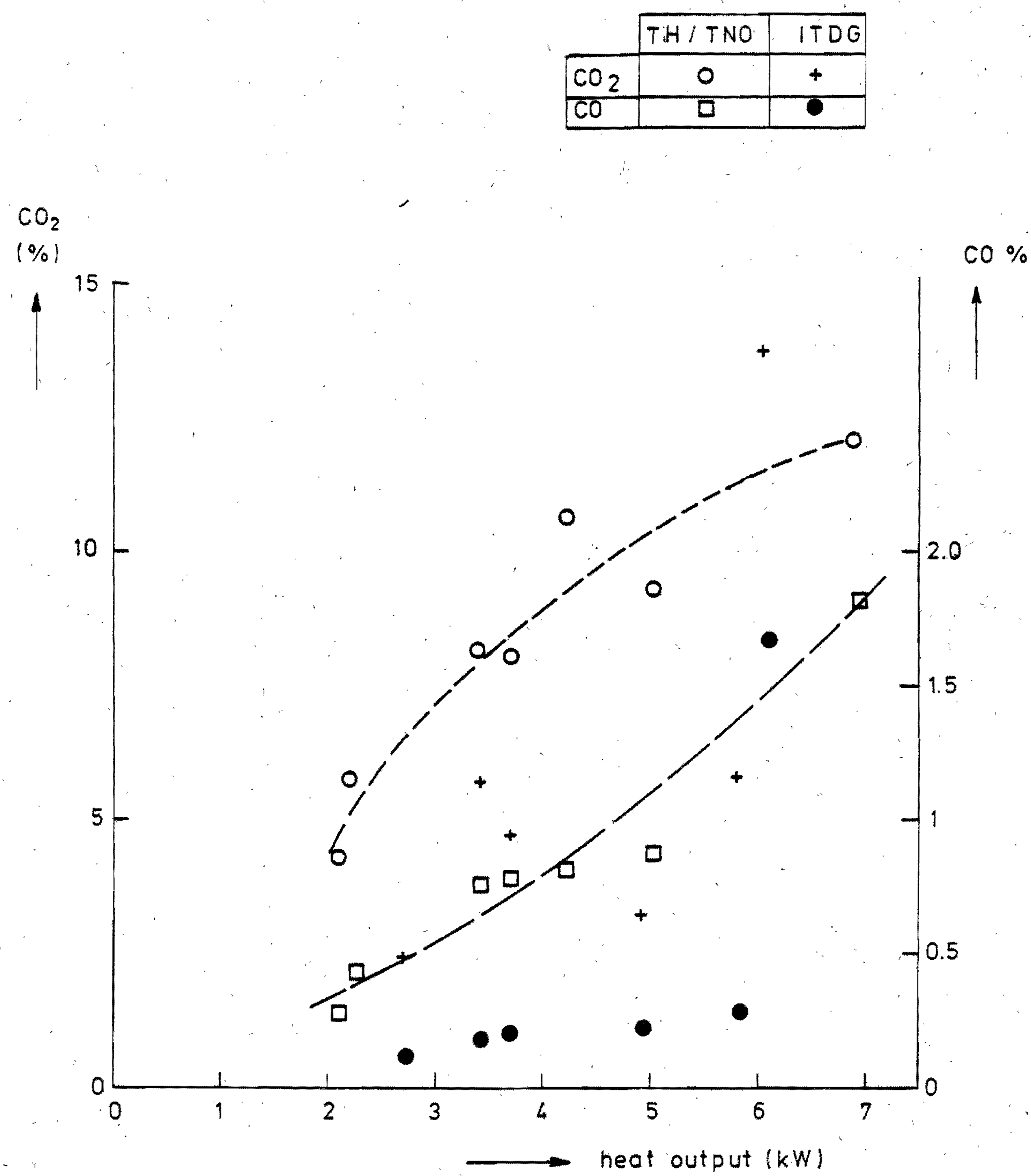
Measured and calculated temperatures of the side walls

MT-TNO
84-940
Fig. 5.14

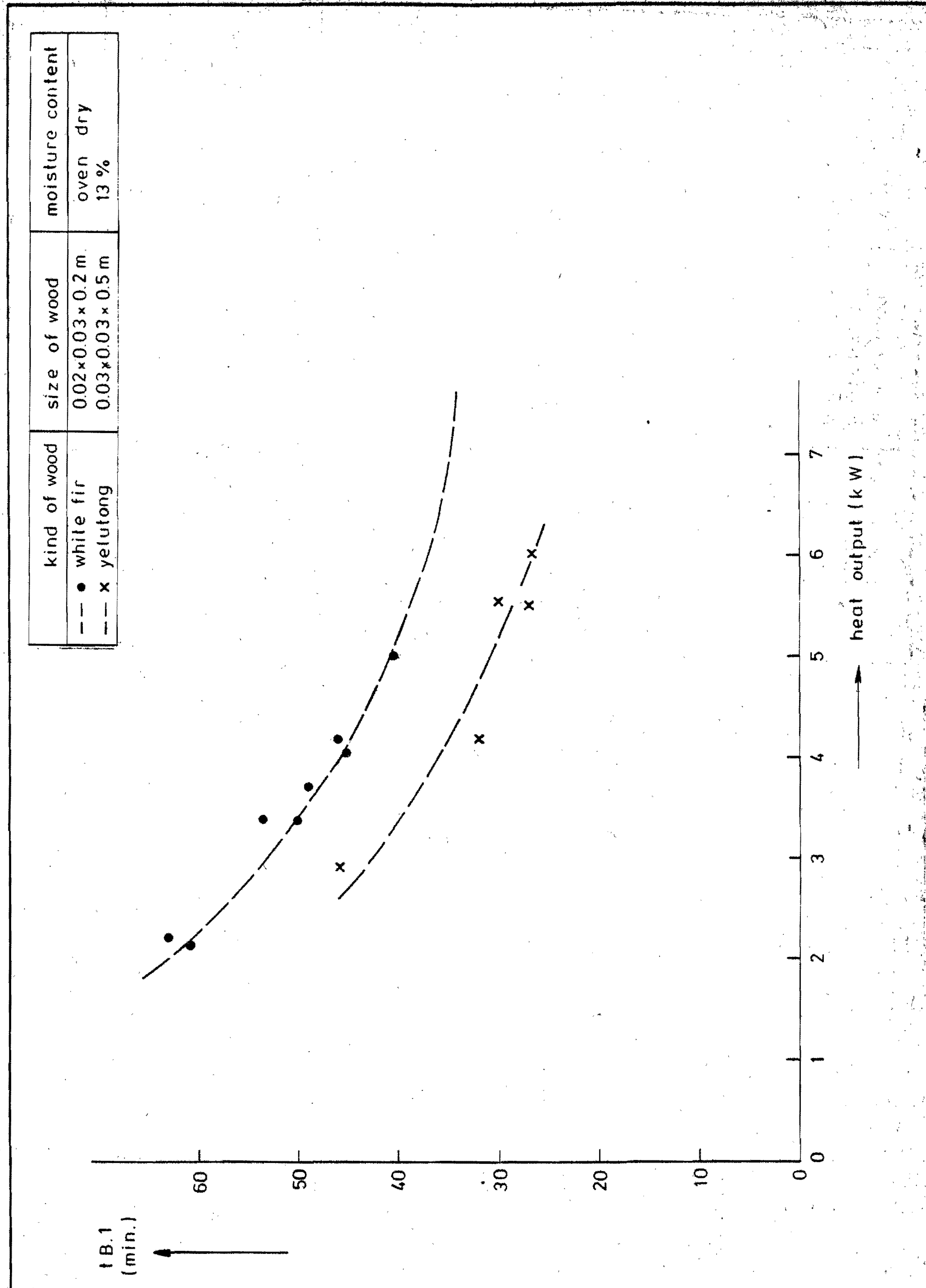


Efficiency and Bp_1 as a function of the heat output for the TH/TNO and ITDG experiments

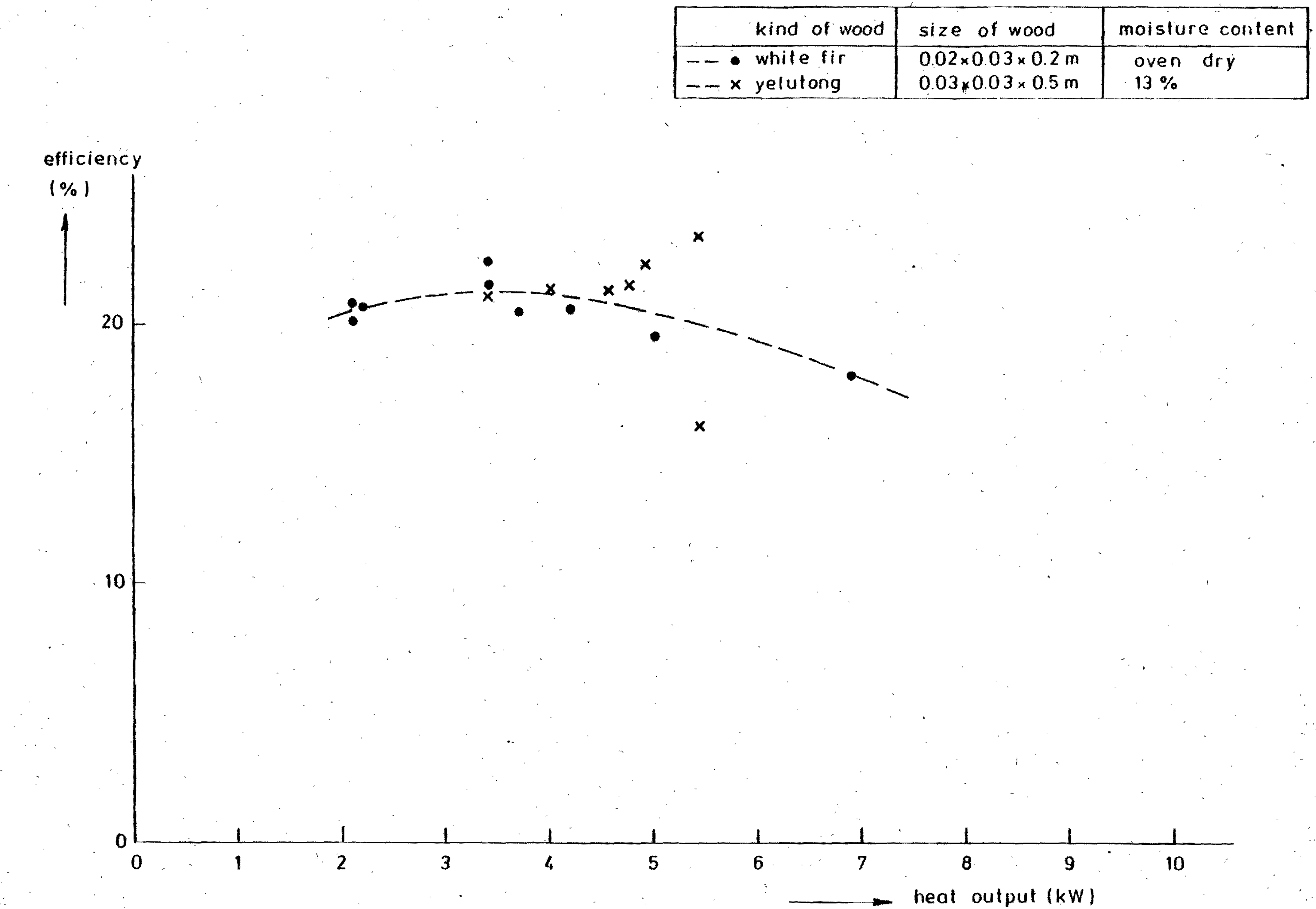
MT-TNO
84.940
Fig. 5.15



CO_2 and CO concentrations as a function of the heat output for TH/TNO and ITDG experiments with white fir	MT–TNO 84940 Fig. 5·16



Time for boiling of water in first pan with 4 kg. water for two ways of firing.	MT—TNO 84940 Fig. 5·17



Comparison of efficiencies during BP_2 for ITDG and TH/TNO test procedure

MT-TNO
84940
Fig. 5.18

Part C: Fuels, testing procedure and performance analyses.

6. AN EXPERIMENTAL METAL STOVE

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6.1 Introduction

The reasons for designing and constructing an experimental metal stove were two-fold. Firstly, the Woodburning Stove Group wanted to have a device where it had easy access to changes in the configuration of the stove. Secondly, we wanted to implement some of the design ideas the group has gathered over the last two to three years while testing currently popular stoves.

The following changes are possible with the design:

(a) the height between the grate and the pan bottom;
(b) adjustable air inlets and chimney dampers;
(c) conversion of the stove into a simple shielded fire by easily removable top collar and chimney assembly; and
(d) a simple collar at the top of the stove to permit variable depth insertions of the pan into the stove body.

The design ideas that can be incorporated into the operation at will are:

(a) provision of primary and secondary air holes with independent adjustments of openings (from 0 to 100% openings);
(b) preheating of combustion air with a double wall construction with the outer wall carrying two sets of primary air holes - one set at the top and the other at the bottom - again with provision of independent controls;

(c) incorporation of a removable eccentric collar on top of the stove to provide for a more uniform flow of combustion gases around the pan on their way to the chimney; and

(d) easily removable grate to study the possible influence of the grate shape on the stove performance.

In addition the stove has been designed as a closed fire chamber with a fuel door.

A geometric view (Fig. 6.1) shows the principal features of the design.

We would like to stress that this stove, although it burns wood and it cooks food, is anything but the final goal we are aiming for: a woodburning cooking stove to be used in the field in vast numbers. The present design is nothing more than a device put together from an investigative point of view, and not from considerations of marketing.

6.2 Technical data

The stove is constructed from 1,5 mm thick stainless steel plate except for the grate, which is a welded mild steel construction. The pan and the chimney are of aluminium. It weighs 28 kg.

The stove is designed to operate at a power of 6 kW. As a point of reference we have taken some crude velocity measurements from the past experience. These velocities ran from 0,5 to 1 m/s of the flue gas in a 0,11 m wide chimney.

Holding to the lowest value and taking into account a flue gas temperature of 150° C a flow rate of $3 . 10^{-3}$ m^3 per second at 0° C results. The corresponding in-flow of combustion air is $2,8 . 10^{-3}$ m^3/s. For a 6 kW fire this leads to an excess air factor of 2. Higher velocities will lead to more excess air.

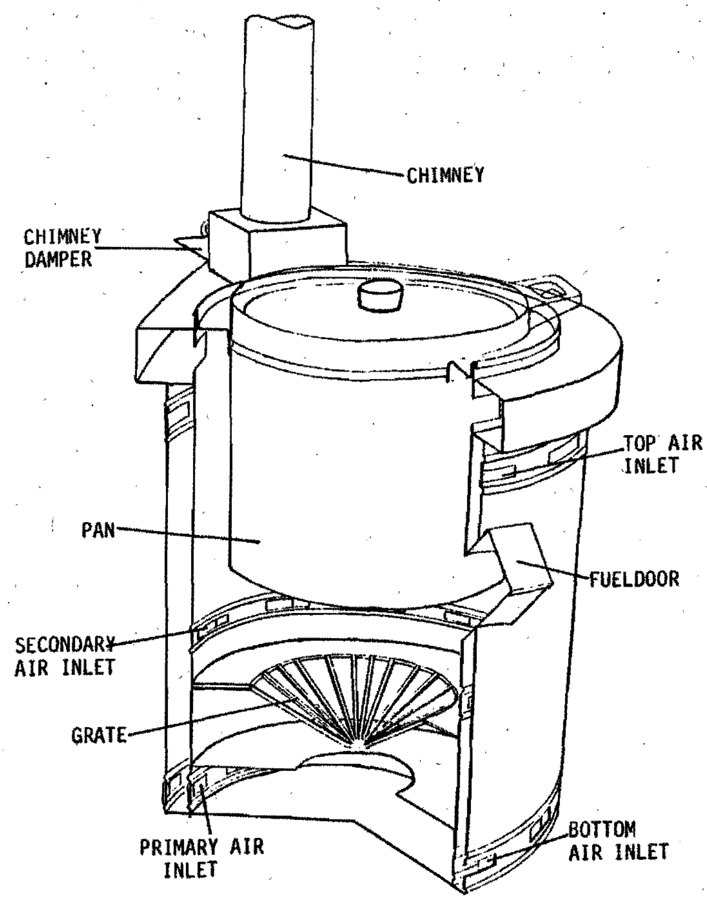

Fig. 6.1. Geometric view of the experimental stove.

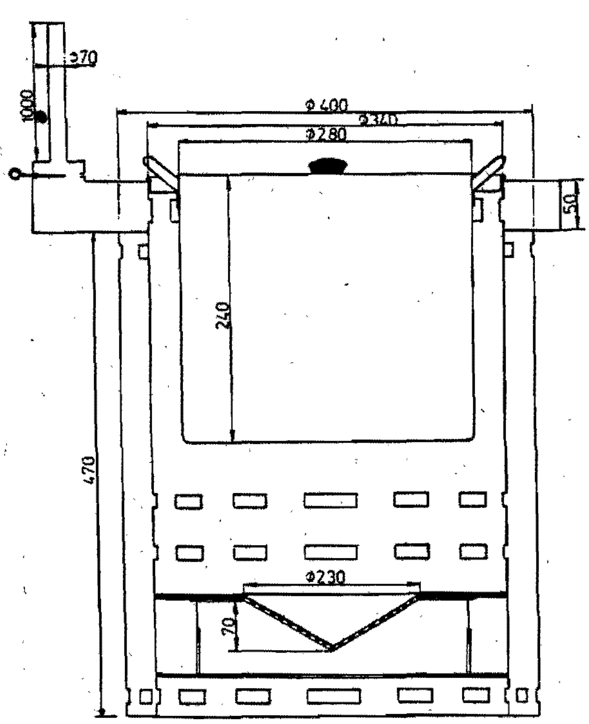

Fig. 6.2. Main dimensions of the stove (dimensions in mm)

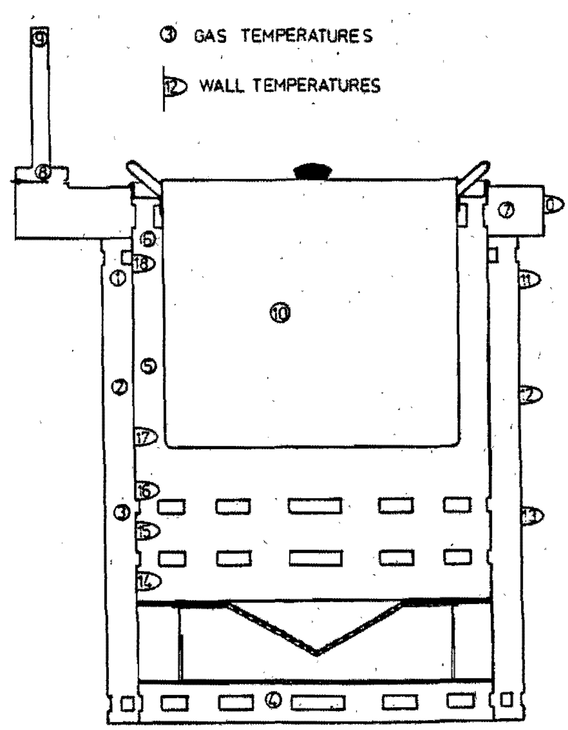

Fig. 6.3. Scheme of positions of the thermocouples.

To us it seemed sufficient to have 60% extra combustion air (λ = 1,6). It was thus decided to have a smaller chimney diameter of 70 mm available in lengths of 0,25 and 1 m. The cross-section of this pipe is 3850 mm^2. To be on the safe side the other passages in the stove were taken about two times bigger than the chimney cross-section. However, there was still the option to reduce these passage areas. So the inlet areas at their 100% openings are as follows.

top inlet:	12 squares of	$50 \times 10\ mm^2$	= 6000 mm^2
bottom inlet:	12 " "	$50 \times 15\ mm^2$	= 9000 mm^2
primary air:	10 " "	$50 \times 15\ mm^2$	= 7500 mm^2
secondary air:	10 " "	$50 \times 10\ mm^2$	= 5000 mm^2

All the inlets are equipped with sliding collars to vary the openings. The chimney has a slide valve at the bottom. The overall size of the stove is dictated by the pan size with a diameter of 280 mm and a 240 mm height and the distance between grate and pan bottom enabling a maximum distance of 200 mm.

The main dimensions are presented in figure 6.2.

6.3 Experimental details

Tests on the performance of the stove described in this chapter concern a limited number of variables. Other parameters and their effect on the performance of the stove will be studied in due course.

The parameters under investigation are:

- the nominal power range;
- the distance between grate and pan bottom;
- the distance between grate and pan bottom in relation to the inlet area of the secondary air holes;
- the pan area exposed to the hot gases (in other words the depth of the pan sunk into the stove body).

In addition there were a few tests to assess the influence of the eccentric gap around the pan, the effect of preheating combustion air and the use of a larger pan.

The parameters that have been set to fixed values are:

- The amount of fuel burnt was always 1 kg of oven-dry white fir.
- The dimensions of the wood blocks were 32 x 32 x 111 mm^3.
- The size of the fuel charges and their feeding rate to the stove is kept at 0,25 kg every 13 minutes. This means a nominal power of 6 kW. Of course this does not hold for the case of varying the power range.
- The pan is filled to one third of its capacity with 5 kg of water.
- The height of the chimney was 1 m and the damper at its bottom was always fully open.

All the tests were boiling water tests with a lid covering the pan. Of every first charge one block was split into four smaller pieces and placed on the grate, the remaining blocks were stacked on top of it and then the wood was lit by a propane burner. With most of the tests temperatures at nineteen spots on the stove (see fig. 6.3) were recorded at 1 minute intervals throughout the experiment. Carbonmonoxide, carbondioxide and oxygen content of the flue gases were recorded at intervals of 10 seconds.

6.4 Instrumentation

The stove temperatures are measured with chromel-alumel thermocouples connected to a datalogger (Hewlett & Packard, type 3497 A), which is monitored by a Hewlett Packard HP 85 computer. CO and CO_2 are measured by infrared gas analyzers (Maihak, type UNOR) and the O_2-measurement is based upon the paramagnetic principle (Siemens, type OXYMAT). The output of the three gas analyzers is also fed to the datalogger mentioned above. The data is stored in a cassette to be transferred later on to the Burroughs 7700 computer for further processing like drawing graphs and making heat balances.

6.5 Stove performance

The performance of the stove is measured in terms of efficiency given by the formula in K. Krishna Prasad (1981). The results of all the tests are summarized in table 6.1 (see Appendix 6.1).

6.5.1 The power range

Determination of the power range was established by varying the time interval between the charges. As mentioned before the charges were 0,25 kg except for the first experiment (see table 6.1) with a power of 4,7 kW which had three charges 0,31 kg. With these experiments (no.'s 1 to 9) both primary and secondary air inlets were set at 50% of the total inlet area. Secondary air was admitted at 50 mm above the grate. The combustion air entered the stove at the top and was therefore preheated. In figure 6.4 the plotted points of efficiency versus power show marginal differences. At the lowest power the fire showed a tendency to die down and at the highest power the fuel bed was building up and when charging another batch of wood there were still a lot of flames from the previous charge. The minimum - maximum power ratio is therefore 1 : 2,2 and the efficiencies range between 37,6% and 42,3%. The time to get the water to boil - t_{boil} - versus the power shows a more pronounced difference

6.5.2 Distance between grate and pan bottom

For this set of experiments (no.'s 10 to 16) secondary air was closed down completely and the primary air inlet fully

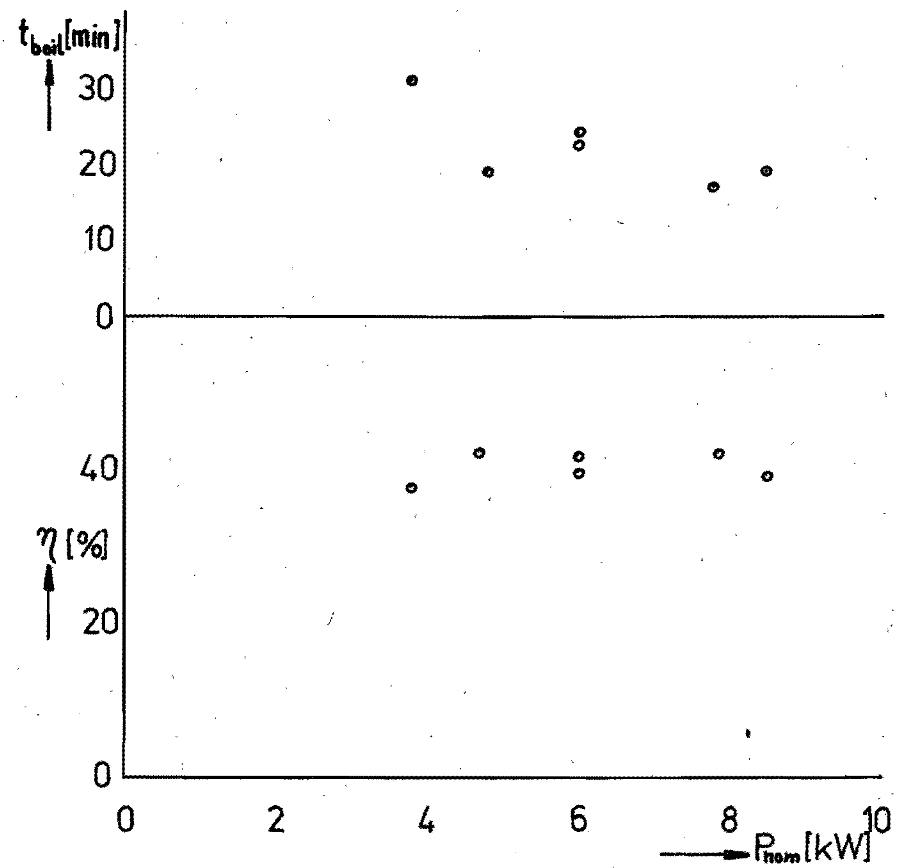

Fig. 6.4. Efficiency and time to boiling as a function of the power output of the fire.

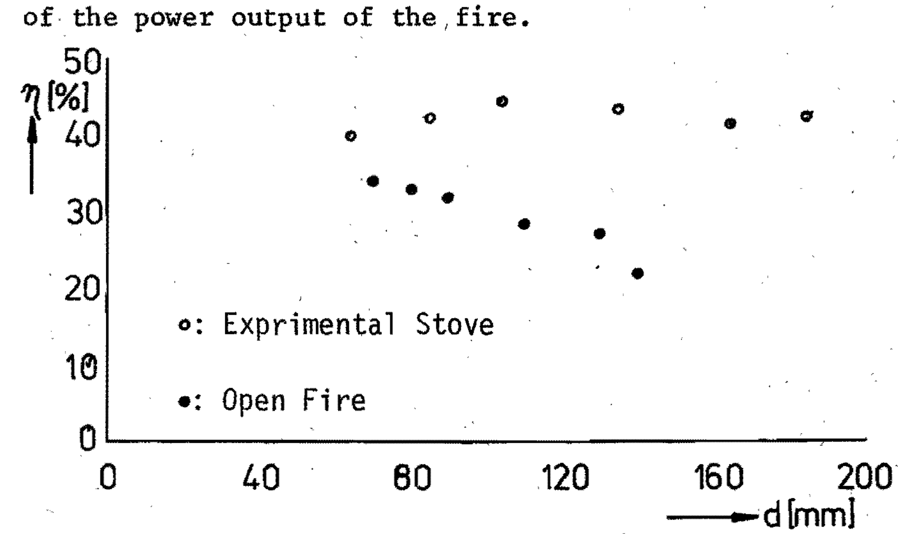

Fig. 6.5. Efficiency as a function of the distance between grate and pan bottom.

open (72 cm^2). The reason for this was that when lowering or raising the grate the distance between the grate and secondary air inlet changes as well. The results are plotted in figure 6.5 and the efficiency ranges between 40,0 - 44,7%. It is interesting to note that this stove shows a very little variation in efficiency with height between the grate and the pan bottom. This result is in contrast with open fires (see P. Bussmann et al., 1983) and the tubular stove (J. Delsing, 1981), both of which show drastic reductions in efficiency with increasing distance between grate and pan bottom.

However, there is strong evidence for poorer combustion with decreasing distance between grate and pan bottom. Observing the average CO contents (see table 6.2) per charge interval of 12 minutes every charge number shows an increasing CO content with smaller heights.

Table 6.2

Exp. no.	D (mm)	Average CO content per charge interval of 12 minutes in percentages			
		1	2	3	4
10	185	0,05	0,19	0,24	0,26
11	165	0,12	0,19	0,21	0,28
13	135	0,10	0,20	0,36	0,39
14	105	0,23	0,28	0,45	0,48
15	85	0,18	0,35	0,37	0,52
16	65	0,35	0,43	0,39	0,40

6.5.3 Distance between grate and pan bottom in combination with secondary air

In this set of experiments (no.'s 18 to 30) secondary air is supplied at 50 mm above the grate for four different distances

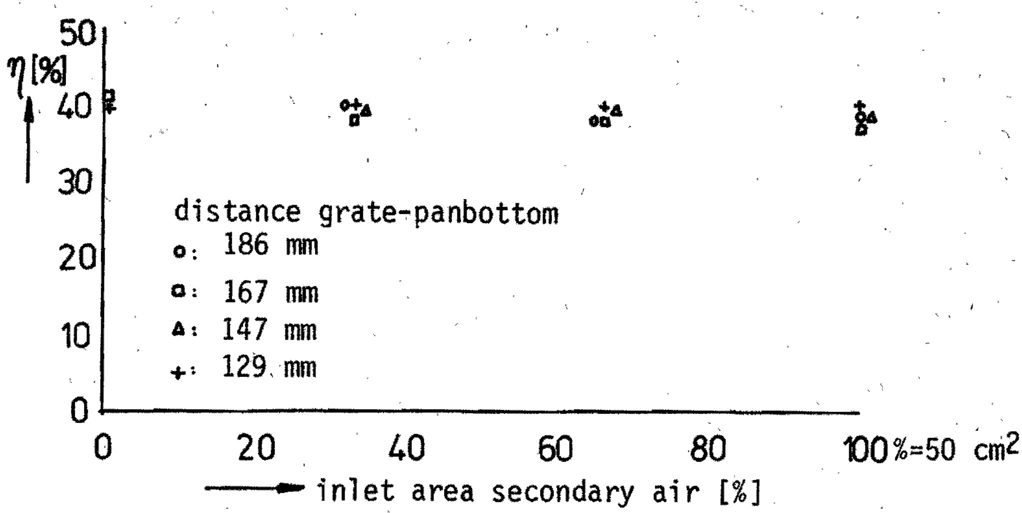

Fig. 6.6. Efficiency as a function of secondary air and the distance between grate and pan bottom.

(186 mm, 167 mm, 147 mm and 129 mm) and for each distance three different inlet areas are chosen (33%, 66% and 100%) and for two distances also 0% secondary air is included. The results are presented in figure 6.6. And the straight forward conclusion of these tests is that secondary air in this stove has no influence. Due to this it was decided to do subsequent tests without admitting secondary air.

6.5.4 Effect of the primary air inlet

Experiments no. 17 and 30 revealed that a decrease of the primary air inlet area by one third did not show a significant change in the efficiency. Therefore some tests (no.'s 37-41) were carried out with which the air inlet was reduced furthermore plus a check test (no. 41) in which the inlet for primary air was set fully open. With all the tests no secondary was admitted

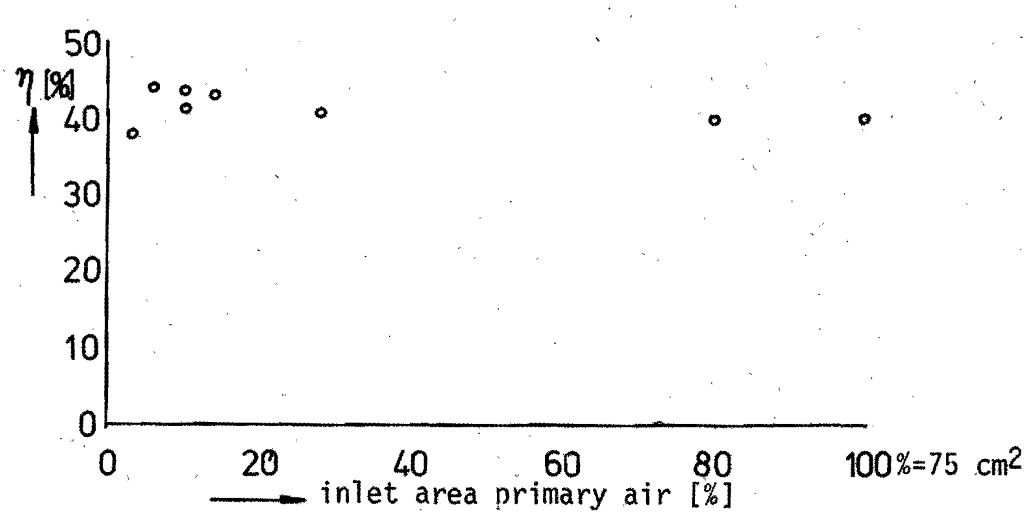

Fig. 6.7. Efficiency as a function of the primary air inlet area.

and the combustion air was not preheated. Figure 6.7 shows an increase in the performance ($\eta = 40,6 - 43,8\%$) only at relatively small openings (0 - 10%; 0 - 15 cm^2). This is probably due to the fact that with small openings less excess air is going through, resulting in higher gas temperatures and hence a better heat transfer. (Compare the three experiments discussed in Sec. 2.2, representing respectively no. 40, 41 and 42.)

Note: Experiments no. 39 is not representative because the stove was tested to determine the extent to which there was air leakage. The test showed no air leaks since the fire extinguished within five minutes with the air inlets closed. To proceed the air inlet was opened by little steps to determine the minimum opening at which the fire could sustain itself. This appeared to be at an opening of 2 cm^2.

6.5.5 Pan wall exposed to the hot gases

By shifting the ring around the pan downwards, the pan depth into the stove can be decreased, which means that the pan wall area exposed to the hot gases and the hot stove wall also decreases. With the experiments no.'s 33-36 (see table 6.1) the distance between grate and pan bottom is kept at 150 mm. This, however, limited the number of experiments, because lifting the pan furthermore and keeping the same pan-grate distance resulted in blocking the fuel loading door. To evade this one could take the pan off and then add the wood. This was considered inconvenient and only four tests are carried out with the normal refuelling procedure. They show a decreasing efficiency (42,3% - 37,3%) with a decreasing pan depth (205 mm - 102 mm) (see fig. 6.8). Extending the number of experiments for smaller pan depths seems worthwhile, because the smallest pan depth of 10 cm is just above the water level in the pan. Going beyond this level might influence the efficiency even more, but it also calls for repeating the other four tests at the new refuelling procedure for good comparison.

It should be noted that shifting the pan upwards, more of it is exposed to the environment and due to this a higher cooling rate of the pan occurs especially in draughty conditions.

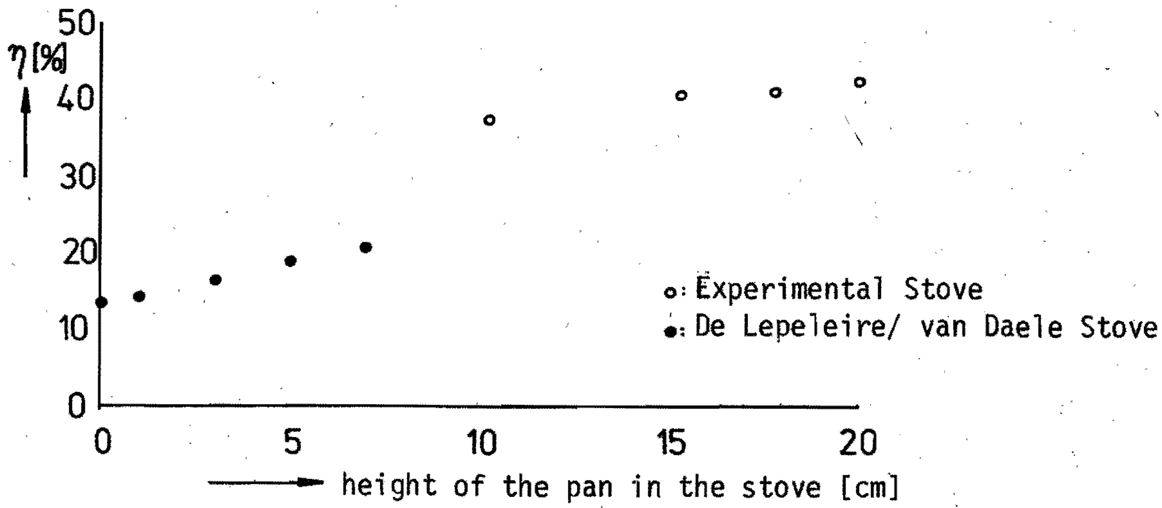

Fig. 6.8. Efficiency as a function of various pan depths.

6.5.6 The eccentric gap around the pan

This design idea was suggested by W. Micuta in "Modern Stoves for all" (1981). When sinking the pan into the stove body, and having a chimney, one has to prevent the hot gases being short circuited directly to the chimney entrance. Therefore, a ring-like baffle was installed which narrowed the passage near the chimney and provided for a wider gap at the other side of the pan.

Comparing the experiments 4-5 and 12-13 it shows a significant difference of 4,6 to 5,6 percentage points with and without the use of an eccentric ring.

6.5.7 Preheating the combustion air

Also an idea of Micuta was to equip a stove with a double wall and lead the combustion air from the top to the bottom underneath the grate in order to preheat the combustion air. This procedure also reduces the temperature at the outside of the stove. Preheating in experiment 13 resulted in an efficiency of 43,4% whereas experiment no. 17 without preheating resulted in 40,3% efficiency. There is a difference but not very large. One of the reasons for the meagre result is probably that there is too large a gap (30 mm) between the two walls. This is due to the fact that all the energy taken up is by convection. To promote this the gap between the walls should be small (see De Lepeleire, 1983).

6.5.8 The use of a bigger pan

Out of 1 mm sheet aluminium a pan was constructed with a diameter of 320 mm and a height of 240 mm. Introducing this pan into the stove brought about two changes in the configuration of the stove. Firstly, the heat exchange surface was enlarged from 0,21 m^2 to 0,32 m^2. Secondly, the gap between the stove wall and the pan reduced from 30 mm to 10 mm. In the first test (no. 43) the circumstances were the same as for the previous test with the normal pan (no. 42; η = 40,6%), and an efficiency of 46,8% was obtained. In the second test (no. 44) the pan was filled with 12 kg of water instead of 5 kg and the air inlet was reduced from 75 cm^2 to 6 cm^2 and an efficiency of 55,3% was recorded.

6.6 Heat balance calculations

Heat liberated by burning wood in a stove ends up as follows:

(a) The pan,

(i) heating up the water and

(ii) producting steam during the boiling period

(iii) heat stored in the pan

(iv) heat lost due to convection and radiation from the pan walls;

(b) The stove body, accounting for

(i) radiative and

(ii) convective heat losses of the exposed parts to the environment

(iii) and the accumulated heat in the stove body;

(c) Stack losses, consisting of

(i) sensible heat carried away by the hot gases

(ii) unburnt CO and

(iii) unburnt C_xH_y

The latter two are a result of incomplete combustion and called the latent heat.

(d) Soot and tar deposits partly stuck to the pan walls, and the inside of the stove body and partly carried away by the flue gases.

(e) Unburnt charcoal in the ash at the end of an experiment.

Efficiencies in our studies always account for a (i) and (ii).

Quantitive measurements of C_xH_y, soot and tar by T.N.O. (J. Claus et all, 1981) and weight loss measurements of the fuelbed in the Nouna Stove and visual observations of the reminder of the fuelbed at the end of an experiment with the Experimental Metal Stove as well as the De Lepeleire Stove revealed that c (iii), d and e are insignificant.

6.6.1 The T.N.O. method for calculating heat balance.

The procedure for calculating the heat balance by T.N.O. for the De Lepeleire Stove, the Nouna stove and the Tungku Lowon is as follows. There are three periods during an experiment, which play a role in determining the various heat losses.

(1) The firing period. At fixed time intervals a fixed quantity of wood is fed to the stove. The firing period ends when a new charge of wood should be added but is omitted. So the firing period is simply determined by the number of charges times the charging interval. The duration of the firing period ranges between 80 and 180 minutes. During this period gas and surface temperatures and CO-, CO_2-, and O_2-concentrations of the flue gas are recorded at 20 second intervals. It is in this period only that the excess air factor and the stack gas losses are determined.

(2) The burning out period. This accounts for a relatively small time lapse (6 to 28 minutes) in which the remaining wood and charcoal is burned. This phase reaches its end when the CO_2 content drops below 1%, this coincides with the moment that the water in the first pan ceases to boil. Then the pans are taken out and weighed to determine the water loss due to boiling. Then the overall efficiency over the entire experiment is calculated.

For calculating the radiative and convective heat losses the stove is divided in about 20 surface elements. Of every element the temperature is recorded at 20 second intervals. In this way every 20 seconds the radiative heat losses of all the elements are calculated and adding them together over both firing and burning out period will give the total heat loss by radiation. The same applies for the convective heat loss.

(3) The cooling period. After the pans are taken of the stove all the opening in the stove body like the fuel loading entrance, pan holes and chimney entrance are sealed. During this phase the stove is left for 24 to 48 hours to cool down to room temperature to determine the accumulated heat in the stove body. The cooling rate of the stove body is recorded by means of temperature measurements at 1 to 5 minute intervals. The thermocouples for measuring these temperatures are placed at representative spots in the stove body. The spots were selected by means of an infrared meter. When the stove body reaches its initial state at room temperature, the total amount of radiative and convective heat losses account for the accumulated heat in the stove body. This type of calculating the accumulated heat applies mostly for high mass stoves, because it is very difficult to assess the temperature profile in a complex geometry and therefore the heat distribution in the stove. Schematically the experiment looks as follows:

firing period	burning out	cooling period
samples at 20 sec. intervals duration of 2 - 3 hours	samples at 20 sec. intervals duration of 20 min.	samples at 1-5 min. duration of 24-48 hours

→ calculation of Q_{sens}, Q_{lat}, λ (firing period)

→ calculation of Q_{acc} (cooling period)

→ calculation of Q_{rad}, Q_{conv}, η (firing period and burning out)

The argument for neglecting the burning out period concerning the calculation of the stack losses and excess air is that during this period the gas flow is small and the CO and CO_2 concentrations are low. In this period the burning rate drops because available fuel is diminished, so a smaller amount of CO_2 is produced. Because of the lower burning rate the temperature in the chimney drops. Hence less air is drawn in.

6.6.2 The THE method.

The method for calculating the overall heat balance by THE differs from the TNO method for several reasons:

(a) If we restrict ourselves to the Experimental Metal Stove, the total experimental time is significantly shorter, being of the order of 80 to 100 minutes, than the time taken with e.g. the Nouna Stove tests which last for 120 to 180 minutes.

(b) Also the firing period takes much lesser time. About 50 minutes as against 80 to 150 minutes with the TNO-tests.

(c) However, the burning out period is longer (40 to 60 minutes) due to a severe build up of the fuel bed. It therefore is also difficult to make a fair distinction between the firing and burning out period.

(d) The stoves, tested by THE, were mainly low mass metal stoves. So keeping a cooling period seemed redundant because at the end of an experiment the temperature of various elements were accurately known.

So the heat balance calculations adopted by THE look as follows. Calculation of the excess air factor, the sensible heat loss and the latent heat loss is based upon the carbon balance. Therefore the CO and the CO_2 content is averaged over the entire experiment and related to the total amount of wood burned.

For the sensible heat the temperature at the bottom of the chimney is taken and also averaged over the experiment. For determining the radiative and convective heat loss the same procedure is followed as with TNO with the only difference that the time interval at which the temperature is measured is 1 minute instead of 20 seconds. The accumulated heat in the stove body is determined by the initial and final temperatures of the elements. The initial temperature is assumed to be room temperature which holds for the stove as a whole. And every element is assumed to have a uniform final temperature.

6.6.3 Instantaneous heat balance.

There were some motives which made it desirable to have a closer look at the heat distribution in a stove during operation. The considerations for taking up this task were twofold. One is based upon practical reasons on designing a stove and another is from an investigative point of view.

When designing a stove one should aim for a piece of equipment in such a way that it matches the cooking task. For instance cooking rice involves bringing rice and water to the boil in about 20 minutes and let it simmer for about ½ an hour. The idea is to charge a sufficient amount of wood with which it is possible to have a period of a relatively high power during the flame phase. This phase also enables one to build up a fuelbed which contains enough charcoal to fulfil the simmering task at a low power. Ideal would be if, one charge will meet both ends. There are two sets of limitations that hinder the realization of this ideal. In the heating phase these factors are like size of the combustion chamber, production of unburnt constituents (smoke), CO emission, a too fast escape of the volatiles, etc.. In the simmering phase these are too high a power or cooling of the combustion gases due to excess air.

All these short comings will be uncovered by the instantaneous heat balance and appropriate measures can be taken to improve the design. For instance when during the flame phase high absolute values of CO are found, combined with excess air factors, the combustion chamber might need a closer look. Or when low efficiencies are found in combinations with high powers. Baffles in the stove need attention. In case of high excess air factors during simmering a reduction of the air entries is called for.

Returning to the tests and their results of the Experimental Metal Stove, a few phenomena caught the eye. One is the difference between the charged or nominal power output and the average power output. As mentioned before the burning out period is about as long as the firing period. Therefore the mean power is half the nominal power (see table 6.1.).

From visual observations and previous studies of the open fire (Bussmann and Visser, 1981) it appeared that the heat liberated by the fire was far from stationary. Looking at the gas analysis chart (fig. 6.9.) there is quite a variation in the concentrations of the various gas components. E.g. O_2 varied between 8% and 20%, CO_2 between 13% and 0,5% and CO between 0,5% and 0,05%. These changes coincide with the flame and non-flame period. Whereas with the flame period relatively high CO_2 percentages were obtained as against low percentages for CO_2 in the non-flame phase. Also the temperatures showed a similar tendency of increasing and decreasing as the gas concentrations.

Whereas all these phenomena are relative measures an attempt is made at quantifying them by drawing up an instantaneous heat balance.

6.6.4 Instantaneous volume flow.

The main problems encountered in doing this was to establish how much wood and which part of it was burnt per minute, because the

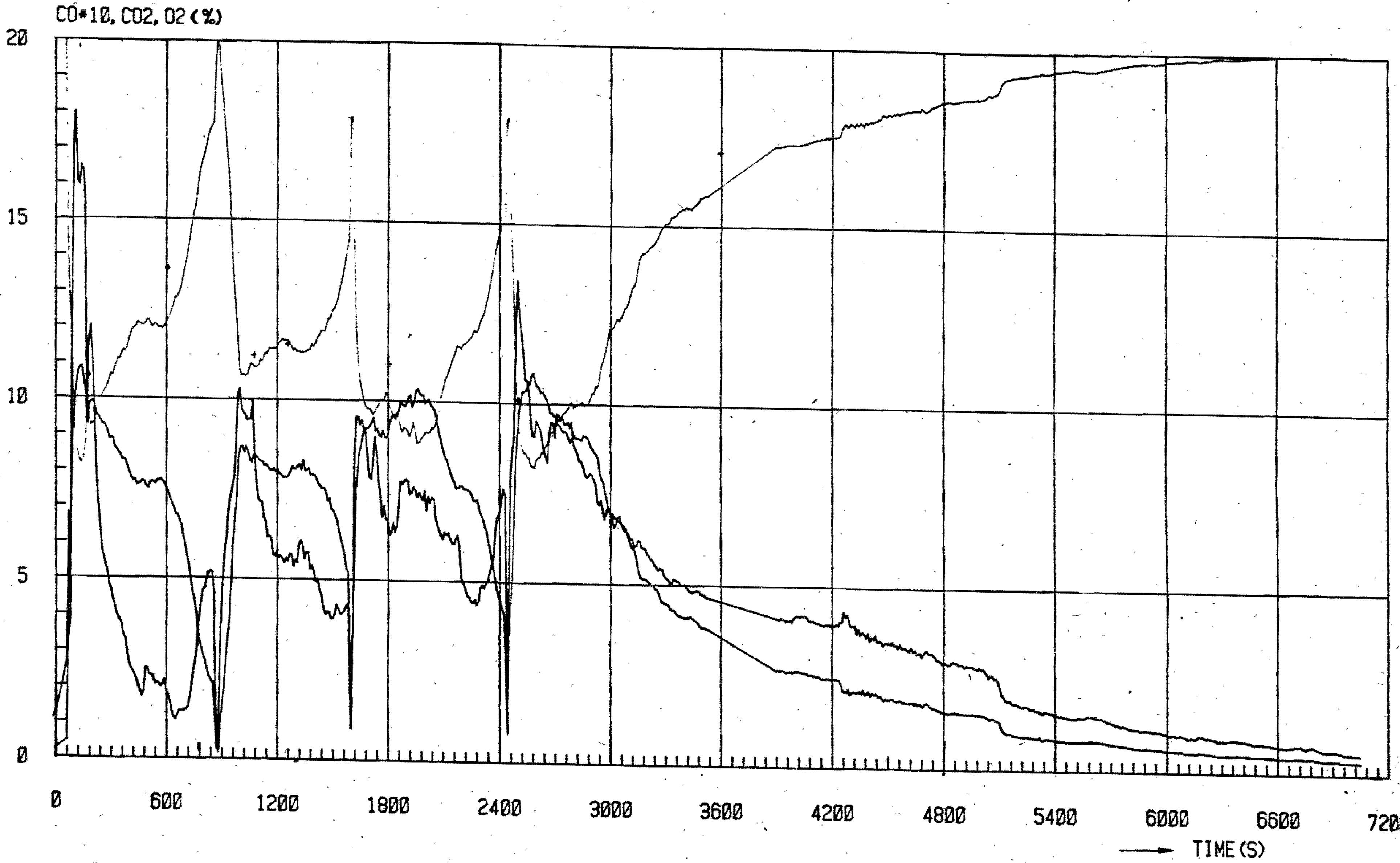

Fig. 6.9. Gas analysis chart of experiment no. 40.

volatiles produce less heat per unit mass (9,7 mJ/kg) than charcoal (33 mJ/kg).
While the stove was too heavy to put on the balance, used for the open fire, there had to be another way of assessing the weight loss of the fuelbed.
Therefore we tried to relate the volumeflow through the stove to the draught in the chimney, which is determined by the temperature in the chimney and the temperature of the environment.

An average flue gas flow $\overline{\phi}_H$ is calculated, based upon mean values of CO and CO_2 averaged over the entire experiment.
The driving force for this flow is the mean chimney temperature $\overline{T}_{chim}$ defined as:

$$\overline{T}_{chim} = \frac{1}{n} \sum_{1}^{n} \tfrac{1}{2} (T_{top} + T_{bot}) - T_e$$

T_{top} = temperature at the end of the chimney [^oC]
T_{bot} = temperature at the bottom of the chimney [^oC]
T_e = temperature of the environment [^oC]
n = total number of temperature measurements during an experiment

The quotient of the average flue gas flow and the mean chimney temperature acts as a constant for calculating the instantaneous rate of volume flow in the chimney

$$\phi_{chim} = \frac{\overline{\phi}_H}{\overline{T}_{chim}} \cdot (\tfrac{1}{2}(T_{top} + T_{bot}) - T_e)$$

According to this model the volume flow varied between 1 and 6 litres per second during an experiment.

Armed with the instantaneous volume flow and the recorded values for CO and CO_2, the amount of carbon burned can be calculated. Also the latent and sensible heatlosses per minute can be calculated.

Now the next problem is whether the carbon originates from the volatiles or the charcoal. That determines the heat liberated by the fire. Relating the instantaneous heatlosses to the momentary heat input causes very unsatisfactory results. Leaving an unaccounted for of - 110% to + 60%. At the time of this writing this problem had not yet been solved. The instantaneous heat balance could tell us more about what happens in the stove when it is in operation. But after dealing with all kinds of phenomena on heat transfer we have to return to where we commenced, at the motor of the system - the fire -.

6.7 Heat balance results.

Over the past year a computer program for heat balance calculations has been developed and is near completion. At this stage some results could be presented. Some problems with the data processing kept us from presenting the results of all the tests.
The formulas applied in the computer program are listed in Appendix 6.2.

6.7.1 Overall heat balance.

In table 6.3 the results of five tests are tabulated. The experiment numbers correspond with the numbers in table 6.1. The three tests with the numbers 40,41 and 42 are the same as discussed in chapter 2. The four lines at the top of table 6.3 indicate some of the initial conditions at which the tests are run and the parameters that are changed. The experiments 40 and 42 are conducted under the same conditions so one gets an impression about the deviation of the calculated values.

In experiment no. 42 the primary air is set fully open which leads to significant changes, compared with the previous two tests. The mean power increases as well as the excess air.

Therefore the amount of flue gas going through the stove per unit time has increased and a higher sensible heat loss results of 35,5% compared to 24% of the previous tests.

Experiment no.		40	41	42	13	16
d (grate-pan bottom)	mm	80	80	80	135	65
prim. air inlet	%	16	16	100	100	100
preheated comb. air		no	no	no	yes	yes
nominal power	kW	6,0	6,0	6,0	6,0	6,0
mean power	kW	2,6	3,0	3,8	2,9	2,9
time to boil	min	19	21	18	19	19
total burning time	min	118	105	83	98	103
$\overline{CO}$	%	0,42	0,41	0,25	0,26	0,31
$\overline{CO_2}$	%	4,35	4,68	3,72	4,75	3,84
$\overline{O_2}$	%	15,46	15,36	16,75	15,25	16,13
excess air - λ -	-	4,6	4,3	5,5	4,3	5,3
useful heat efficiency	%	43,5	41,6	41,3	44,0	40,1
stack losses						
sensible heat	%	24,2	23,3	35,3	22,8	27,1
latent heat	%	6,9	6,2	4,5	3,9	5,5
stove body losses						
radiation	%	5,2	4,75	3,1	6,0	4,1
convection	%	16,7	15,0	10,9	16,9	13,1
accumulation	%	4,4	4,5	5,7	6,2	5,3
unaccounted for	%	- 0,9	+ 4,9	- 0,8	+ 0,2	+ 4,8

table 6.3 Overall heat balance results and other calculated values.

Comparing the heat losses by radiation and convection the three experiments show a decrease in heat loss with time.

At the tests no. 13 and no. 16 the distance between grate and pan bottom has been halved from 135 mm to 65 mm. It can be seen that the differences are not as great as one should expect. Most striking is the decrease in efficiency whereas one should expect the opposite from the open fire experiments. Another point which can be clearly stated is that in the case of experiment no. 16 with the small distance of 65 mm combustion quality is far lower compared to experiment no. 13. This can be concluded from the higher CO-content and the higher CO-content and the higher excess air factor. As a consequence absolute values of CO production and latent heat loss will be higher in experiment no. 16 compared to no. 13. As an illustration some of these values are presented in table 6.4.

In general it can be stated that 40% of the heat is delivered to the pan, 30% is leaving the chimney as stack losses. Whereas the CO production in terms of combustion quality is not much to worry about. As a health hazard it should always be looked at. About 20% of the heat generated is lost to the surroundings by radiation and convection. The heat stored away in the stove body is 5% and the unaccounted for is about 5%.

6.7.2 Instantaneous heat balance.

The instantaneous heat balance is an attempt to determine the heat losses at 1 minute intervals. While weight loss experiments of the fuel bed were beyond the reach in the Metal Stove as well as water weight losses of the pan only during the period of bringing water to the boil it was possible to determine the useful heat going into the pan on the bases of water temperature measurements.

time	CO-emission $\frac{mg}{s}$		latent heat kW		CO_2-content %	
min.	13	16	13	16	13	16
1	0,2	0,1	0,00	0,00	1,19	1,0
3	2,7	26,2	0,02	1,24	6,8	8,9
5	6,8	22,2	0,06	0,20	9,1	8,1
7	3,4	18,2	0,03	0,17	7,3	6,1
9	3,2	16,2	0,03	0,15	5,9	5,6
11	4,1	15,7	0,04	0,14	5,8	3,5
13	8,8	12,6	0,08	0,12	4,2	2,0
15	7,9	23,2	0,07	0,21	8,4	6,1
17	8,6	32,8	0,08	0,30	7,0	9,0
19	7,5	25,7	0,07	0,24	6,7	7,9
21	10,8	27,6	0,10	0,25	6,5	7,4
23	10,6	24,8	0,10	0,23	5,6	6,0
25	21,0	18,4	0,19	0,17	8,2	3,0
27	38,6	19,4	0,35	0,18	14,0	3,5
29	18,6	24,5	0,17	0,22	11,0	7,7
31	12,7	23,1	0,12	0,21	9,5	7,6
33	11,1	26,2	0,10	0,24	8,6	7,5
35	20,5	33,9	0,19	0,31	5,6	7,4
37	25,1	30,6	0,23	0,28	6,6	5,8
39	42,5	33,5	0,39	0,31	13,0	4,5
41	19,6	37,4	0,18	0,34	11,0	7,1
43	13,8	28,1	0,13	0,26	10,5	6,3
45	12,7	23,4	0,12	0,21	9,4	5,8

table 6.4 Instantaneous values of CO-emission, latent heat and CO_2-content of experiments no. 13 and no. 16.

Taking experiment no. 40 as an example the following table can be produced (see table 6.5). The numbers in the first column are the minutes in which the heat losses are determined. The second column represents the useful heat that went into the pan to heat up the water. The third column represents the total heat losses and the fourth is the sum of the values in the second and third column; in other words, the heat that we have been able to account for. The fifth column lists calculated values of the accounted heat divided by 40%. This will be discussed later. The sixth represents the heat liberated by the fire calculated with the model applied in the computer program. The seventh column gives the difference between the accounted heat and the heat produced by the fuel bed. The unaccounted for is expressed in percentages. In the last column the gas flow through the stove is given merely from an illustrative point of view. At the bottom of the table the average values are calculated of the foregoing numbers.

As a comparisson we calculate the heat that went into the pan during the heating up period,

$$P_{\text{to boil}} = \frac{5{,}0.4{,}186.(100-24)}{19.60} = 1{,}40 \text{ kW}$$

which exactly conforms with the mean value mentioned at the bottom of column 2. If we assume an efficiency of 40% during the heating up period the total amount of heat generated will be:

$$\frac{1{,}40}{0{,}4} = 3{,}50 \text{ kW}$$

which is the same as the average value of the accounted heat. Then same procedure is followed for the instantaneous values and the results are shown in column 5. Ruling out the initial two values and the last one, the resemblance with the accounted heat values in column 4 is satisfying. The reason why the calculated heat in the first two minutes turns out to be too

time	heat into the pan	total heat losses	total heat losses + pan	calcul. divided by 40%	fuel bed	unacc. for	gas flow through the stove
[1]	[2]	[3]	[4]	[5]	[6]	[7]	[8]
min	kW	kW	kW	kW	kW	%	L/S
1	0,38	0,94	1,32	0,95	2,65	+ 50	1,7
2	0,94	2,26	3,20	2,35	6,06	+ 47	2,7
3	1,40	2,50	3,90	3,50	6,28	+ 38	3,1
4	1,95	2,67	4,62	4,88	5,70	+ 19	3,2
5	2,02	2,75	4,77	5,05	5,43	+ 12	3,3
6	1,64	2,66	4,30	4,10	5,04	+ 15	3,3
7	2,13	2,44	4,57	5,33	4,52	- 1	3,2
8	1,78	2,58	4,36	4,45	4,70	+ 7	3,3
9	2,16	2,59	4,75	5,40	4,80	+ 1	3,4
10	1,78	2,63	4,41	4,45	4,45	+ 1	3,4
11	1,85	2,42	4,27	4,63	3,86	- 11	3,3
12	1,47	1,98	3,45	3,68	2,69	- 28	3,1
13	0,98	1,37	2,35	2,45	1,76	- 34	2,8
14	0,70	0,90	1,60	1,75	0,78	-105	2,5
15	0,56	0,58	1,14	1,40	1,56	+ 27	2,2
16	0,63	1,33	1,96	1,58	4,72	+ 58	2,9
17	1,60	2,34	3,94	4,00	5,99	+ 34	3,4
18	1,67	2,64	4,31	4,18	5,94	+ 27	3,5
19	0,87	2,72	3,59	2,18	5,61	+ 36	3,5
	1,40		3,52	3,49	4,34	mean values average over 19 minutes	

table 6.5 Instantaneous heat losses and their derivative values during the heating up period of experiment no. 40.

low is due to the fact that relatively more heat is required to heat the stove body and therefore the assumed efficiency of 40% is too high. Calculating the heat absorbed by the pans by taking only the temperature rise of the water is bound to give unrealistic values when boiling temperature is reached. That is why in the 19th. minute the estimated heat that went into the pan, is too low, because it is likely that already heat is taken up to evaporate water.

The main conclusion that can be drawn is that determination of the different heat losses of the various elements is rather good. Even assessing the stack losses which are based upon the flue gas flow make sense. Whereas the model for determining the flow seemed a weak link in the calculation, this doubt can be taken away.

From values listed in column 6 and the average value it can be said that the heat liberated by the fuel bed is estimated too high. Along with this they show a very inconsistent course which is stressed once more by the percentages unaccounted for in column 7.

There are various effects that obstruct a clear establishment of the heat flow from the fuelbed.
One is the energy that went into the wood to heat to liberate the volatiles. Although this energy is incorporated into the heating value of the wood. This heat is taken up in a rather short time lapse and has to be compensated for in a later stage.
E.g. when 0,25 kg. of white fir with a specific heat of 0,27 kJ/kgK has to be heated to 420^{o} C and when this takes one minute then a heat flow of

$$\frac{0{,}27 \times 400 \times 0{,}25}{60} = 4{,}5 \text{ kW is required}$$

Thus when a new charge is added less heat can be delivered to the pan in the first minute than the successive minutes.
A second problem is to assess the contribution of the volatiles to the heat flow. We assume that, during the firing period, (which consists of 4 batches of wood each weighing 0,25 kg. charged every 13 minutes) only volatiles are burning. Then the average power will become

$$P = \frac{0{,}8.15{,}2.10^3}{52.60} = 3{,}9 \text{ kW}$$

1 kg. of white fir consists of 80% volatiles with a heating value of 15,2 MJ/kg. The value of 3,9 kW is still higher than the average value of the accounted heat (3,52 kW). That suggests that not all the volatiles were burned during the firing period, which was more or less confirmed by the fact that there were still flames visible after 52 minutes. But the whole reasoning gets clouded if one realizes that during the firing period also charcoal is burning.

It becomes clear that we are not get able to determine the instantaneous heat liberated by the fuelbed and therefore needs closer investigation.
Solving this problem will provide a better grip on how to control the fire, one of the most promising factors of saving fuel.

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APPENDIX 6.1 Table 6.1: Summary of the experimental results

Exp. no.	Mass wood (kg)	Power nominal (kW)	Power average (kW)	Time total (min)	Time to boil (min)	Distance gr.-pan B (mm)	Preheat. of comb. air	Mass evap. (kg)	A prim. (cm^2)	A sec. (cm^2)	η (%)	Height pan wall (mm)
1	0,93	4,7	-	-	19	170	yes	2,51	37	25	42,2	205
2	1,02	6,0	3,4	94	23	170	"	2,60	37	25	39,7	
3	1,00	7,8	4,7	69	17	170	"	2,70	37	25	42,3	
4	0,99	6,0	3,3	94	23	170	"	2,26	37	25	36,4*	
5	0,98	6,0	3,1	98	24	170	"	2,60	37	25	42,0	
6	-	-	-	-	-	-	"	-	-	-	-	
7	0,96	3,8	2,5	120	31	170	"	2,23	37	25	37,6	
8	0,98	8,5	3,6	85	19	170	"	2,40	37	25	39,5	
9	0,94	6,0	-	-	22	170	"	2,18	37	25	37,3	
10	0,94	6,0	3,4	86	21	185	"	2,57	60	0	42,2	
11	0,94	6,0	3,2	92	19	165	"	2,50	60	0	41,4	
12	0,94	6,0	2,9	100	22	135	"	2,34	60	0	39,3*	
13	0,90	6,0	2,9	98	19	135	"	2,55	60	0	43,9	
14	0,92	6,0	2,9	100	19	105	"	2,69	60	0	44,7	
15	0,96	6,0	3,0	100	18	85	"	2,63	60	0	42,4	
16	0,96	6,0	2,9	103	19	65	"	2,44	60	0	40,0	
17	0,96	6,0	3,4	87	19	135	no	2,49	60	0	40,3	
18	1,00	6,0	3,9	80	21	167	"	2,37	21	50	37,4	
19	1,00	6,0	3,7	85	19	167	"	2,41	21	34	38,0	
20	0,90	6,0	3,2	89	21	167	"	2,42	21	17	38,4	

Table 6.1 (continued)

Exp. no.	Mass wood (kg)	Power nominal (kW)	Power average (kW)	Time total (min)	Time to boil (min)	Distance gr.-pan B (mm)	Preheat. of comb. air	Mass evap. (kg)	A prim. (cm^2)	A sec. (cm^2)	η (%)	Height pan wall (mm)
21	1,00	6,0	3,8	83	20	129	no	2,59	21	50	40,2	205
22	0,98	6,0	3,6	84	20	129	"	2,47	21	34	39,8	
23	1,00	6,0	3,3	94	19	129	"	2,61	21	17	40,4	
24	0,99	6,0	3,5	89	19	186	"	2,54	21	17	40,4	
25	1,00	6,0	3,4	91	20	186	"	2,48	21	34	38,8	
26	0,99	6,0	3,3	93	19	186	"	2,43	21	50	38,6	
27	1,00	6,0	3,3	94	20	147	"	2,52	21	50	39,3	
28	1,00	6,0	3,3	94	22	147	"	2,53	21	34	39,5	
29	1,01	6,0	3,4	92	19	147	"	2,53	21	17	39,1	
30	1,01	6,0	3,3	97	18	129	"	2,68	21	0	40,8	
31	0,99	6,0	3,3	95	17	102	"	2,87	21	0	43,9	
32	1,00	6,0	3,4	92	18	80	"	2,98	21	0	44,8	
33	1,00	6,0	3,3	94	20	150	"	2,77	21	0	42,3	205
34	1,02	6,0	3,4	95	18	150	"	2,74	21	0	41,2	178
35	1,00	6,0	3,4	92	18	150	"	2,64	21	0	40,8	153
36	0,99	6,0	3,8	82	18	150	"	2,32	21	0	37,3	102
37	1,00	6,0	3,6	88	18	80	"	2,82	10,5	0	42,9	205
38	0,97	6,0	3,0	102	17	80	"	2,81	24	0	43,8	
39	0,99	6,0	2,6	120	34	80	"	2,37	0	0	37,7	
40	1,00	6,0	2,9	106	19	80	"	2,90	7,5	0	43,3	

Table 6.1 (continued)

Exp. no.	Mass wood (kg)	Power nominal (kW)	Power average (kW)	Time total (min)	Time to boil (min)	Distance gr.-pan B (mm)	Preheat. of comb. air	Mass evap. (kg)	A prim. (cm^2)	A sec. (cm^2)	η (%)	Height pan wall (mm)
41	1,00	6,0	3,3	94	21	80	no	2,71	7,5	0	41,6	205
42	1,00	6,0	3,8	82	18	80	"	2,64	75	0	40,6	
43	1,03	6,0	4,0	80	16	80	"	3,26	75	0	46,8**	
44	1,00	6,0	3,5	90	32	80	"	2,85	6	0	55,3**	205

Remarks:

* no eccentric ring

** larger pan size ϕ: 320 mm
H: 240 mm

APPENDIX 6.2.

To determine the heat losses due to radiation, convection and accumulation, the stove is divided into 9 elements. Of every surface element the heat losses per minute are calculated and added together over the entire experiment. The formulas applied in the computer programme for determining the losses are the following:

Radiation

$$Q_{rad.i} = \sigma.\varepsilon_i.A_i.(T_i^4 - T_e^4).\Delta t$$

Convection

$$Q_{conv.i} = \alpha_i.A_i.(T_i - T_e).\Delta t$$

$$\alpha_i = \frac{Nu.\lambda}{L_i} \; ; \; Nu = C_1(Pr.Gr)^{C_2}$$

$$Gr = \frac{g.L_i^3}{V^2}\left(1 - \frac{T_e}{T_i}\right)$$

Accumulation

$$Q_{acc.i} = C_{p.i}.m_i.(T_{new} - T_{old})_i$$

The stack gas losses are divided into two parts, one part represents the sensible heat carried away by the hot gases, the second part is the latent heat loss due to CO-formation.

Sensible heat

$$Q_{sens} = \phi_{fl}.C_{p.fl}.\rho_{fl}.(T_{bc} - T_e).\Delta t$$

Latent heat

$$Q_{lat} = H_{CO}.\phi_{fl}.[CO].\rho_{CO}.\Delta t$$

For calculating ϕ_{fl} see Sec. 6.6.4 on instantaneous volume flow. The formulas used for calculating the heat produced by the fuelbed are left out of consideration here because the calculation is incomplete.

The applied symbols used above are listed in Appendix 6.3 and the element characteristic values and material constants are listed in Appendix 6.4.

APPENDIX 6.3.

List of symbols used in the formulas presented in Appendix 6.2.

$Q_{rad.i}$	amount of heat lost by radiation by an element i	[J]
$Q_{conv.i}$	amount of heat lost by convection by an element i	[J]
$Q_{acc.i}$	amount of heat accumulated in an element i	[J]
Q_{sens}	amount of sensible heat lost by the dry flue gas in the bottom of the chimney	[J]
Q_{lat}	amount of heat lost due to CO-formation	[J]
T_i	temperature of an element i	[K]
T_e	temperature of the environment	[K]
T_{new}	temperature of an element i measured during the minute following the minute at which T_{old} is measured	[K]
T_{old}	tempeature of an element i measured during the minute preceding the minute at which T_{new} is measured	[K]
T_{bc}	temperature at the bottom of the chimney	[K]
A_i	surface area of an element i	[m^2]
Δt	time interval over which the heat loss is	[s]
m_i	mass of an element i	[kg]
L_i	characteristic length of an element i	[m]
$C_{p.i}$	specific heat of an element i	[J/kgK]
$C_{p.fl}$	specific heat of the dry flue gas	[J/kgK]
Nu	Nusselt number	[-]
Pr	Prandtl number	[-]
Gr	Grasshof number	[-]
C_1 & C_2	constants dependent on the geometry of surface element i as well as the value of Gr.Pr	[-]
g	acceleration due to gravity	[m/s^2]
H_{CO}	heating value of CO	[J/kg]
[CO]	concentration of CO	[-]

σ	Stefan Boltzmann constant	$[W/m^2K^4]$
ε_i	emission coefficient of an element i	[-]
α_i	heat transfer coefficient for convection of a surface element i	$[W/m^2K]$
λ	thermal conductivity of air	[W/mK]
ν	kinematic coefficient of viscosity	$[m^2/s]$
ρ_{fl}	density of the dry flue gas	$[kg/m^3]$
ρ_{CO}	density of CO	$[kg/m^3]$
ϕ_{fl}	volume flow of dry flue gas	$[m^3/s]$

APPENDIX 6.4.

$\sigma = 5{,}7 \cdot 10^{-8}\ W/m^2K^4$

stove body (stainless steel): $\varepsilon = 0{,}24$
pan lid (aluminium) : $\varepsilon = 0{,}04$

$g = 9{,}81\ m/s^2$
$\lambda = 3{,}0 \cdot 10^{-2}$ W/mK of air at 80° C
$\nu = 2{,}1 \cdot 10^{-5}\ m^2/s$ of air at 80° C
$Pr = 0{,}69$ of air at 80° C

	C_1	C_2
top surface of the stove :	0,15	0,33
stove wall (vertical cylinder):	0,59	0,25
stove bottom :	0,58	0,20

stove body (stainless steel): $C_{p.i} = 0{,}46 \cdot 10^3$ J/kgK
pan (aluminium) : $C_{p.i} = 0{,}896 \cdot 10^3$ J/kgK

$C_{p.fl} = 1{,}012 \cdot 10^3$ J/kgK, for this value the C_p of air at 100° C is taken.
$\rho_{fl} = 1{,}293\ kg/m^3$, for this value the density of air is taken at 0° C and 1 bar.
$H_{CO} = 9{,}43 \cdot 10^6$ J/kg of white fire.
$\rho_{CO} = 1{,}25\ kg/m^3$ at 0° C and 1 bar.

7. A SURVEY OF TEST RESULTS ON WOOD STOVES

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7.1 Introduction

Until now the W.S.G. has produced descriptions and test results of several stoves. A comparison of several stoves and an attempt to explain the differences among them have not been done.
One exception has to be mentioned. Claus and Sulilatu (1982) compared three different stoves. However their study was limited to the comparisons of efficiency, heat balance and combustion quality of the stoves.
A more logical way of studying stoves is to relate their design features to the efficiency and combustion quality as was demonstrated by the tests. The present study is thus an attempt to elaborate on the work initiated by Claus and Sulilatu.

Tests on nine stoves (including the open fire) carried out over a period of 3 years and in two labs are included in this study. Due to this fact, several important restrictions arise during comparisons. The first one is that a number of people were involved in conducting the tests. In spite of the relatively simple, but highly regimented testing procedure, this fact no doubt influences the results obtained. For example tests at Eindhoven show that differences up to 10% could be obtained in tests conducted by two people on the same stove under seemingly identical conditions. Presumably this influence could be minimized by performing repeated tests under the same conditions on the same stove. Due to the very large number of variables considered in these tests, such a procedure is too time consuming. Hence the present study does not concern itself with absolute performance values but only with trends of results.

A second difficulty encountered in the study can be attributed to the evolution of thinking about stoves in the course of 3 years. It was not always easy to isolate two variables and study the relationship between these for all the stoves concerned. Hence we have had to satisfy ourselves in most cases with plotting all the measured points and represent them by an agreed average on an ad-hoc basis. Sometimes even this was difficult, a cloud of points emerged on the plot due to the large dispersion in the results.

Some of these problems are characteristic of all large scale testing enterprises. This situation must be superimposed on the fact that we are interested in woodburning stoves for domestic use. These are small naturally aspirated combustion devices. The demands on accuracy and reliability from instrumentation of such systems are quite high and not easily attained. Added to this is the complexity of non steady operation associated with these devices. The following is a simple example of the limitations of instrumentation. In none of the tests considered in this study there has been a direct measurement of air flow into the stove. It has always been inferred from the carbon balance technique using gas analysis and the ultimate analysis of the fuel. Both the latter are subject to considerable uncertainties. Thus we have not been surprised at the lack of elegance in the emerging comparisons.

Finally, we should like to point out a set of restrictions on the present analysis. It does not consider any aspects of materials and constructional techniques for comparison of stove designs. Furthermore, caution is necessary while interpreting the results of this analysis. The results used in this analysis have not been obtained on stoves that could be considered optimum designs in any sense of the word. As the results of Sulilatu et al (see elsewhere in this report) shows the Nouna Stove is capable of much higher efficiencies with relatively small modifications. The purpose of the analysis is therefore quite limited in scope. It attempts to elucidate the factors that could promote better performance from stoves. <u>It does not propose to be a consumer service</u> suggesting <u>stove designs for wide scale adoption</u>.

The study is divided into several sections. The following section provides in tabular form the most important dimensions of the stoves tested. This is followed by fuel consumption estimates derived from efficiency figures of three stoves for specific cooking tasks. We then move on to delineating the differences between one pot and two pot stoves. A logical way of distinguishing between design features and their influence on performance with plausible explanations for the observed behaviour is next considered. Finally some tentative conclusions and suggestions for further work are presented.

7.2 Survey of several stoves and their most important dimensions

The stoves dealt with in this report, are all stoves which have been tested by the woodburning stove group itself. Although a classification can be made in categories like open fires, shielded fires and closed fires this is not done, mainly because only nine stoves are dealt with. Separate treatment of the three groups would give too little data for comparison. That is why all stoves are treated without any distinction based on the design.

A summary of the stoves and the most important dimension is given in table 7.1; in this table also the most important design characteristics are given. Figures 7.1 to 7.9 give sketches of the stoves. In table 7.1 quantities such as P_{max}, P_{design} etc. are also tabulated; these numbers are derived from graphs which are presented later. They are presented here to see, whether it is possible to derive some simple design rules. One could think of ratios like power and grate surface area, power and combustion volume, pan load etc.

In the Woodstove Compendium (De Lepeleire et al. 1981) tentative figures for these ratios have been given. These are however not based on detailed performance testing of several stoves and cannot be considered reliable.

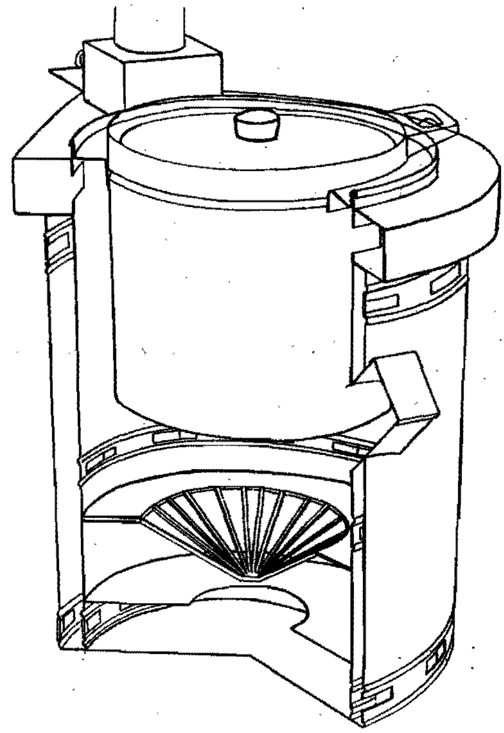

Figure 7.1: The Experimental Stove.

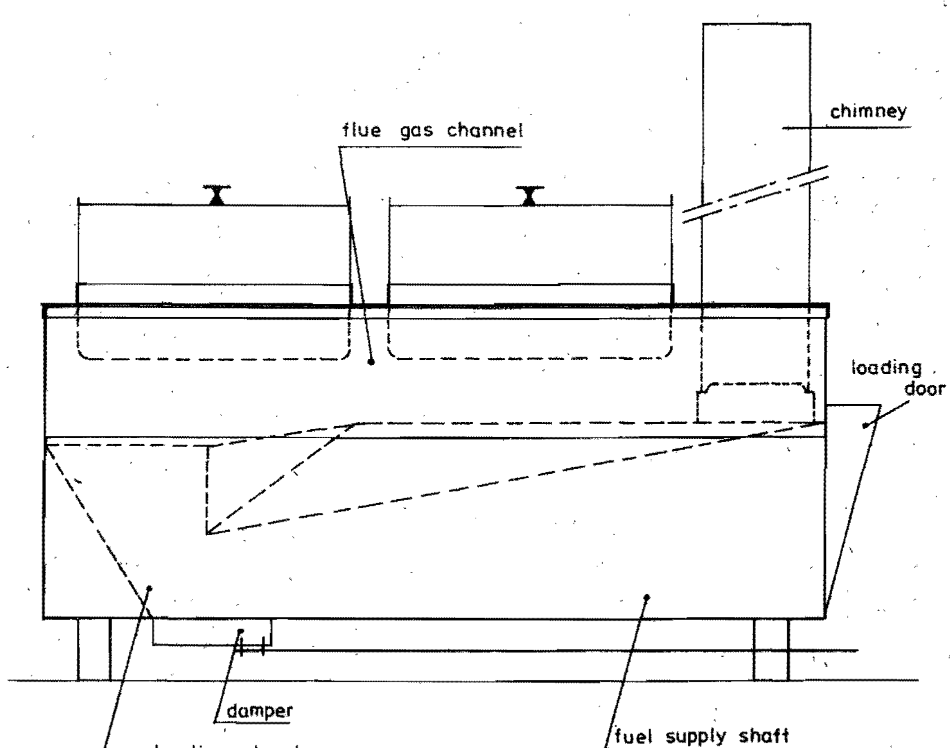

Figure 7.2: The de Lepeleire - van Daele Stove.

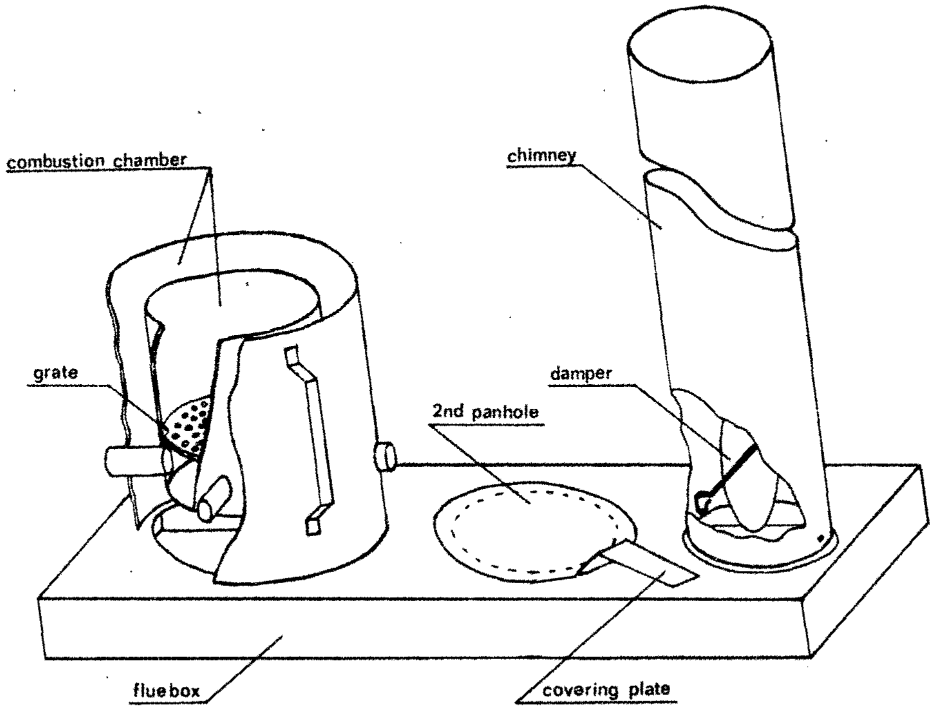

Figure 7.3: The Family Cooker.

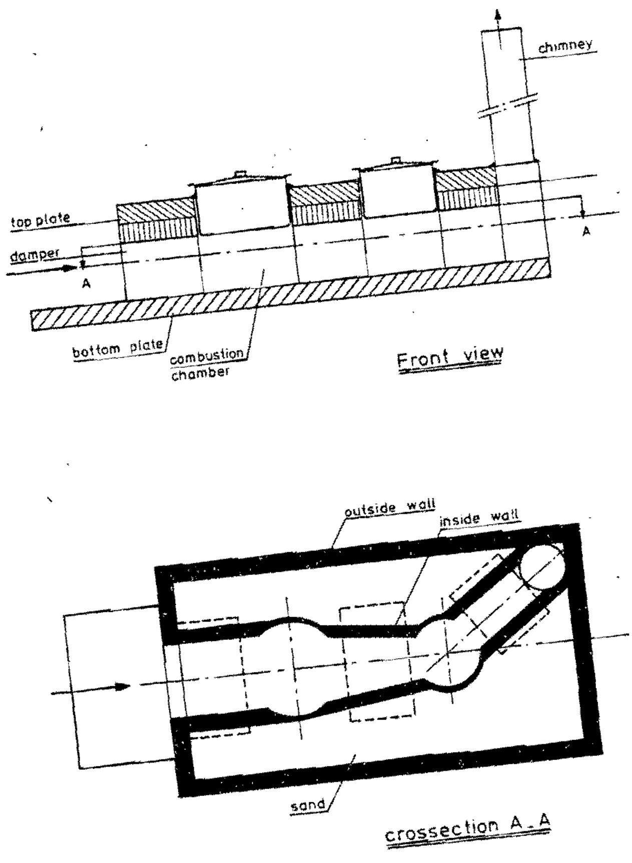

Figure 7.4: The Nouna Stove.

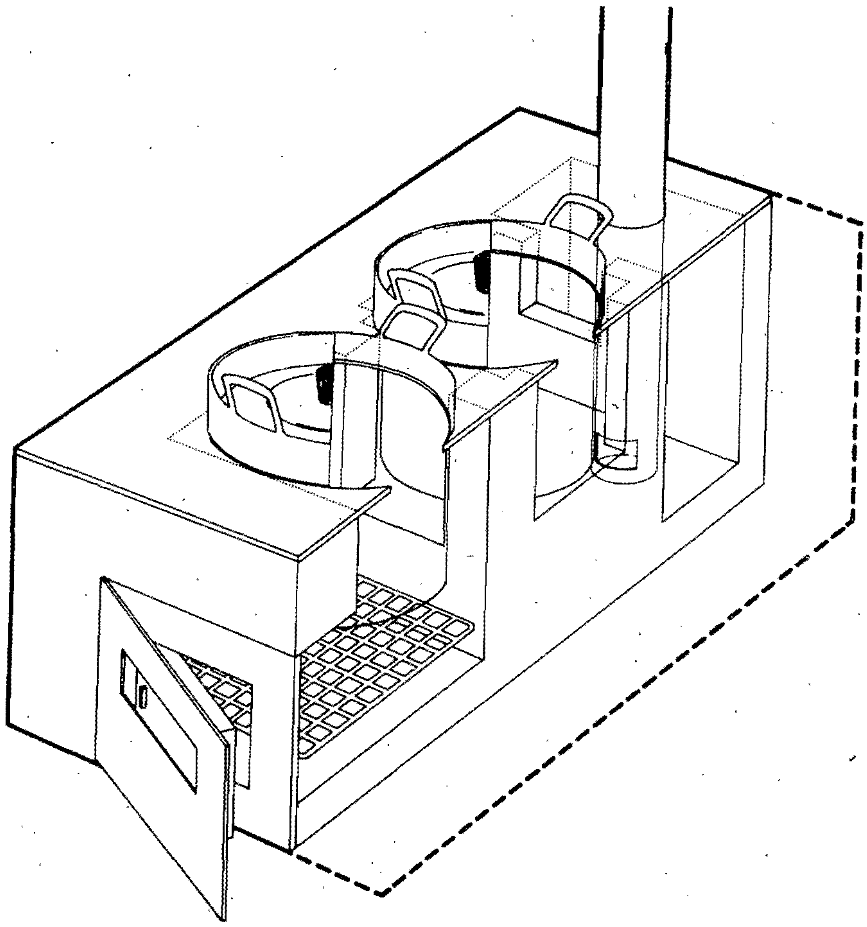

Figure 7.5: The Heavy Experimental Stove.

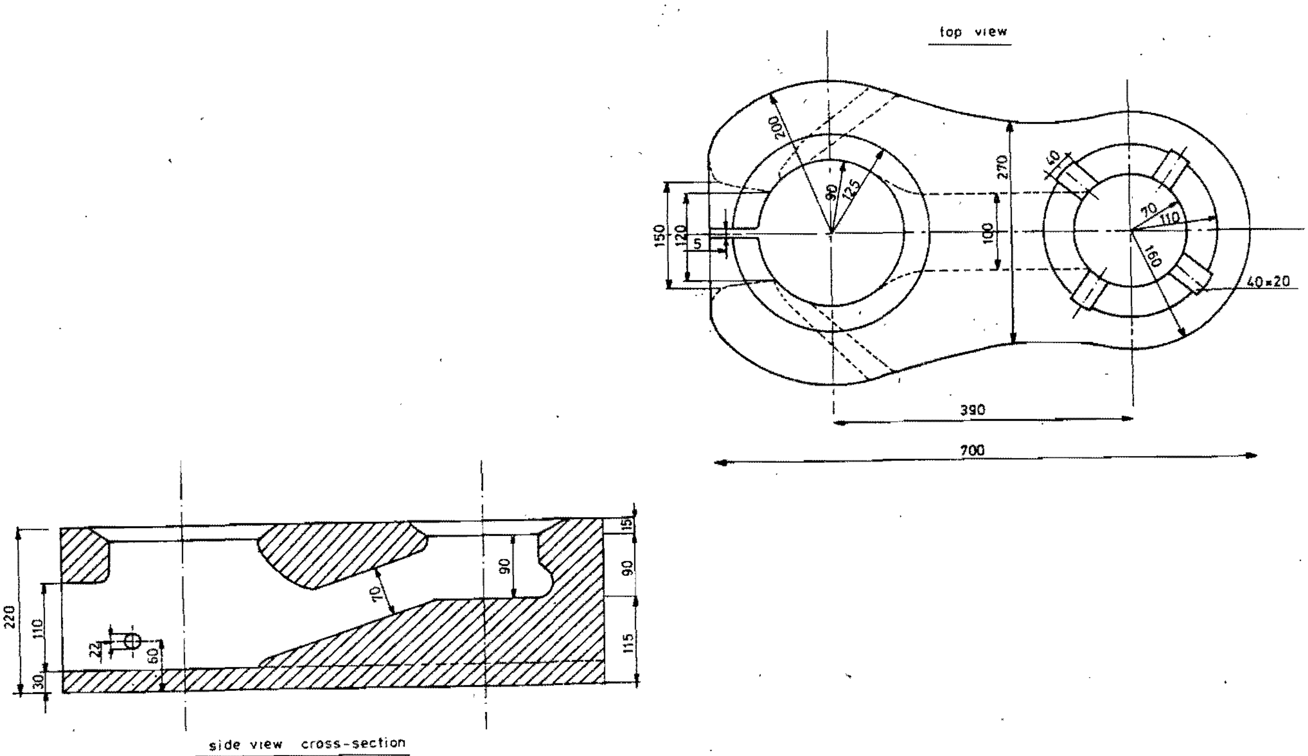

Figure 7.6: The Tungku Lowon Stove.

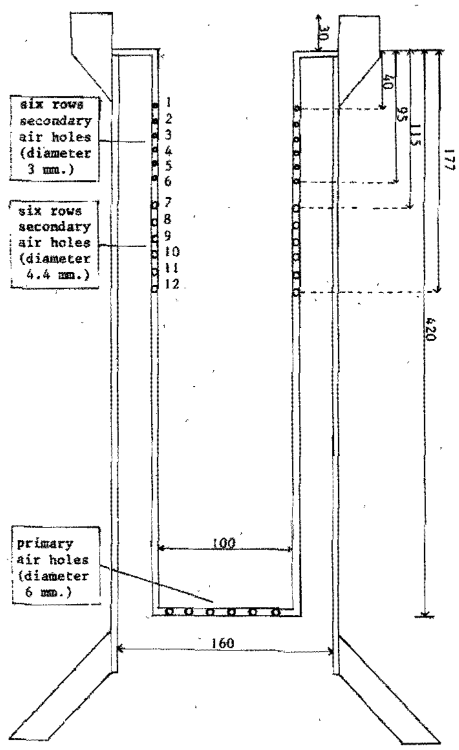

Figure 7.7: The Cylindrical Combustion Chamber.

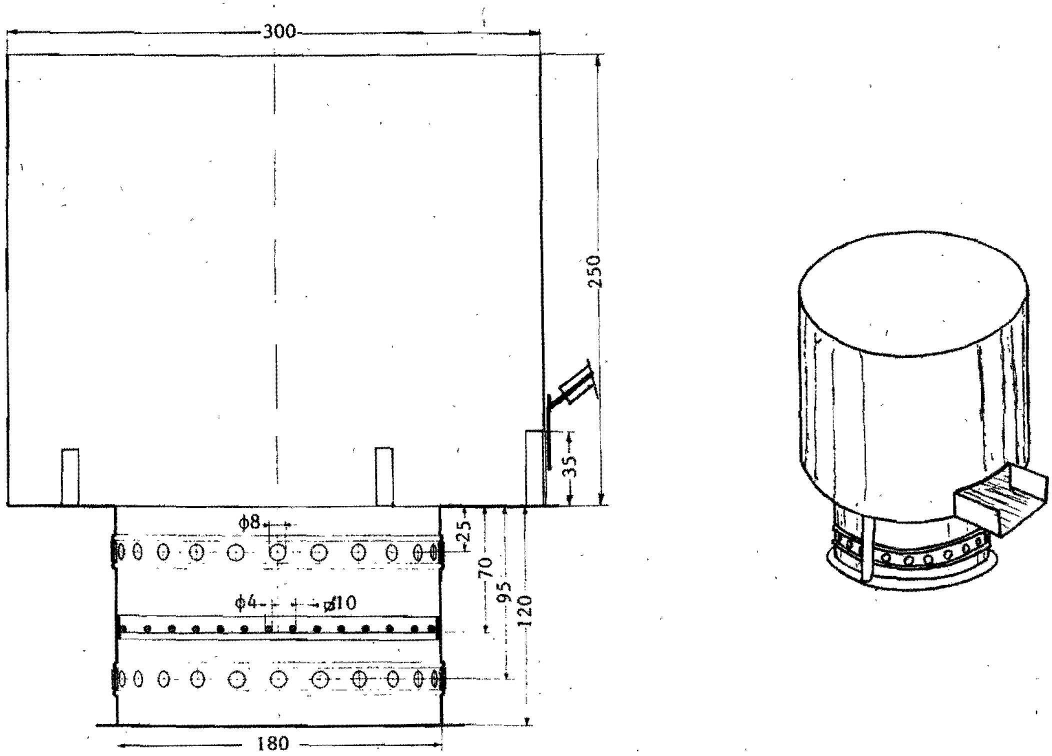

Figure 7.8: The Shielded Fire.

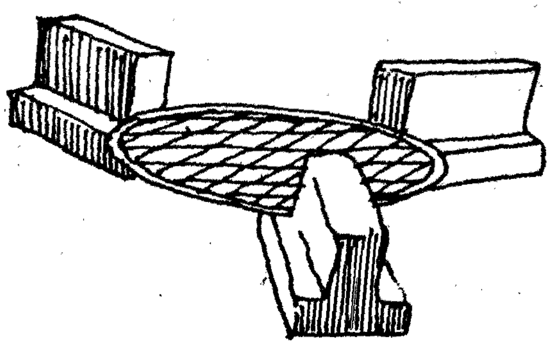

Figure 7.9: The Open fire with grate.

	Experimental Stove	De Lepeleire Van Daele Stove	Family Cooker	Nouna Stove	Heavy Experimental Stove	Tungku Lowon	Cylindrical Combustion Chamber	Shielded Fire	Open Fire
Reference	Vermeer 1982	Prasad 1981 (a)	Prasad 1981 (b)	Prasad 1981 (a) Sulitatu 1983	Prasad 1981 (b) Engberts 1983	this report	Prasad 1981 (b)	Visser 1982	Bussmann 1982
Number of pots	1	2	1	2	2	2	1	1	1
Chimney									
- length (m)	1	.95	2.1	1	1	-	-	-	-
- diameter (mm)	70	100	110	100	70	-	-	-	-
Damper									
- primary	x	-	x	—	x	-	x	x	-
- secondary	x	x	-	—	x	-	x	x	-
- chimney	x	-	-	xx	-	-	-	-	-
Grate surface (cm^2)	430	27	132	490	625	255	79	255	255
Combustion volume (l)	3.44	.51	1.4	9.8	7.5	4.6	2.7	3.1	3.1
Outer dimensions (m)	ϕ.4 * .56	.72 * .30 * .24	.60 * .25 * .23	1.10 * .60 * .30	.93 * .52 * .45	.70 * .40 * .22	ϕ .20 * .50	ϕ .30 * .38	ϕ .28
Power									
- max (kW)	8.5	13	6.5	9.5	12	7	4.16	7	8
- nom (kW)	6	7.5	3	6.5	6	3.5	-	5	4.5
- min (kW)	3.5	4.5	2	4	2.5	1.8	-	2.5	2.5
Combustion volume/ grate surface (l/kW)	.40	.04	.22	1.03	.63	.65	.65	.44	.39
Power/ grate surface (W/cm^2)	19.8	481	49.2	9.6	19.2	27.5	52.7	27.5	31.4
Pan size (m)	.24 * ϕ .28	.155 * ϕ .26 (2x)	.24 * ϕ .28	.145 * ϕ .25 .115 * ϕ .20	.26 * ϕ .28 (2x)	.115 * ϕ .225 .095 * ϕ .20	.24 * ϕ .28	.24 * ϕ .28	.24 * ϕ .28
Pan load (W/cm^2)	13.8	24.5	10.6	19.4	19.5	17.6	6.8	11.4	13.0

Table 7.1: Survey of the stoves and some important dimensions

The figures given in the Compendium are:

Volume of the combustion chamber: 0.6 l/kW

Combustion intensity on a grate : 50 W/cm^2 with chimney
10 W/cm^2 without chimney.

The quantities listed in table 7.1 vary however from .04 l/kW upto 1.03 l/kW for the ratio between combustion volume and power and between 4.81 W/cm^2 and 9,6 W/cm^2 for the combustion intensity on the grate. One has to take into account, that in table 7.1 stoves with and without grate are mentioned. The stoves without grate (Nouna Stove, Tungku Lowon Stove) seem to have a somewhat lower combustion intensity compared to the stoves with a grate. Regarding the simple relations used to characterize the stove designs we can state that the method presented here is not satisfactory and other ways should be developed. Some definitions stated in this text or in table 7.1 have to be explained.

Grate surface; this quantity gives the magnitude of the grate area in cm^2. In most stoves the grate surface area is equal to the base of the combustion room. Where this is not the case (for example in the Experimental Stove) the area of perforated surface is taken. For stoves without a grate (Nouna and Tungku Lowon Stove) a figure between brackets is given; the figure indicates the maximum surface area on which wood can be burned.

Combustion volume; this volume is the product of the grate surface area and the distance between grate and pan. Although other definitions are possible this is the most simple and distinct one. This definition includes however that the volume under the pan in the Experimental Stove (for example) is much larger than the defined combustion volume.

In this study we will mention several power definitions. For clarity we will list them here. They agree with the definitions stated in Bussmann et al. (1982).

Nominal power ; the power as imposed by charge size m charge interval time t and the combustion value of the wood B; $P_{nom} = m \cdot B/t$

Minimal power ; the lowest possible nominal power

Maximal power ; the highest possible nominal power

Design power ; the nominal power which gives the highest efficiency

Average power ; the power obtained by looking at the total burning process. The time interval between lighting of the fire and the instant the water stops boiling is taken for the time and the total amount of wood used for the mass of fuel to determine this average power. The average power will always be less than the nominal power.

The burning intensity on the grate and the ratio between combustion volume and power are both calculated for the maximal power.

7.3 Some calculations on cooking tasks and fuel consumption

The stoves, tested until now by the Woodburning Stove Group, show efficiencies varying from 15 to 50 percent (see figure 7.10, this figure will be explained in detail in section 7.5).
Although features like combustion quality, fire control etc. are important we will neglect these for the moment and concentrate on the efficiency of a stove and the influence of this efficiency on the fuel consumption. First we will have a look at a cooking task, which consists of bringing to boil 5 kg of water. The time needed to perform this is important for the decision whether or not a stove is used. If this time is too long probably a stove is not used at all. A reasonable time which can be stated to perform this task is about half an hour. When we know the efficiency of the stove, we can estimate the required power of the stove, and thus the required wood supply.

The required power P is given by

$$P = \frac{1}{\eta} \frac{m_w c_w \Delta T}{t} \qquad (7.1)$$

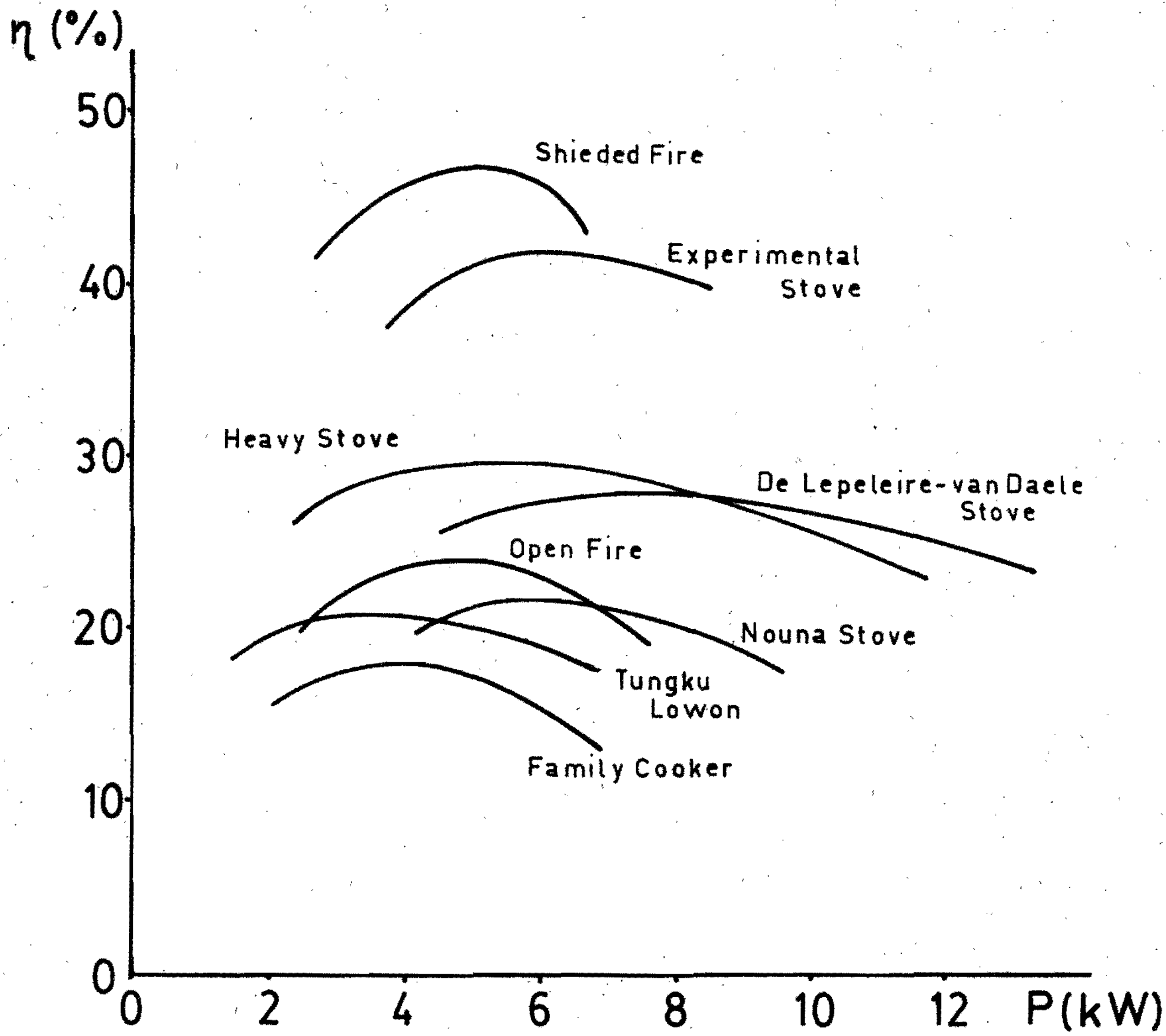

Figure 7.10 : The influence of the power range on the efficiency of the stoves.

In this formula m_w is the amount of water (5 kg); c_w is the heat capacity of the water ($4.2\ 10^3$ J/kg.K); ΔT is the temperature difference between the end and the beginning of the experiment (80 K) and t is the time interval (half an hour). Substitution of these figures into (7.1) yield for the power of the stove.

$$P = \frac{1}{\eta} * 93.33 \qquad (7.2)$$

In this formula P is expressed in kW and the efficiency η in a percentage. A stove with low efficiency ($\eta \approx 15\%$) must have at least a power of 6.2 kW, which, as can be seen from figure 7.10, is realized for every stove. Wood consumption differs however for the various stoves. This wood consumption M can be expressed by the formula

$$M = P \cdot \frac{t}{B} \qquad (7.3)$$

Substitution of the correct values for the cooking time t (1800 s) and the combustion value B ($18.7\ 10^6$ J/kg) yields.

$$M = P * .0963 \qquad (7.4)$$

here the power P is again expressed in kW, while the mass of wood M is expressed in kg.
Substitution of equation (7.2) yields

$$M = 8.99/_{\eta} \qquad (7.5)$$

For the same cooking task a stove B, with an efficiency of half the efficiency of stove A, will consume twice as much wood as stove A does (if they each burn for half an hour).

For heating of periods less than half an hour (this will be the case for higher power levels), the fuel consumption for a stove can be even less. This can be explained as follows. From figure 7.10 we see, that the efficiency first increases and later decreases with increasing power. At the highest efficiency the wood is used as efficiently as possible for that stove.

A stove even uses less wood when a higher efficiency is obtained by a power higher than the required power for half an hour's cooking. One remark however has to be made for figure 7.10. This figure contains the total efficiencies for the stoves, which means that the efficiency for the first pan of a two pot stove is less than given in figure 7.10.

Another cooking task, which is closer to reality is the following task. Bringing a given amount of lentils and water to boil and let it simmer for one hour. This example will stress the importance of the power range of a stove. First of all we will assume a constant efficiency for each stove. From figure 7.10 this seems to be not a bad approximation and it will simplify our calculations (we will only use the maximum and minimum power of the stove). Say we will cook m_l kg of dry lentils; after cooking this will give us $2.44 * m_l$ kg of cooked lentils (see Prasad, 1982); (calculations here are similar to the calculations in Prasad (1982)). For heating and simmering m_w kg of water is needed, which is partially absorbed by the lentils ($1.44 * m_l$ kg) and partially evaporated as steam (m_s kg) during the simmering period.

The stove, as already indicated, will have a constant efficiency and will burn at two powers; namely the maximum power P_{max} (kW) during the heating up period and the minimum power P_{min} (kW) during the simmering period. It is assumed that the meal will keep simmering at P_{min}, which can be confirmed from a cooling test of a pan. This test gave a heat loss of 300 to 600 W in the temperature range of 90^oC to 70^oC. Realizing that this test was done on a free standing pan and that a pan in or on a stove at least from one site is not exposed to the surroundings we can accept the statement that the pan's contents will keep simmering at P_{min}. (lowest power of a stove is about 2 kW; lowest efficiency is about 20%, which means that at least 400 W is transfered to the pan.)

We can now make the following calculation on fuel use and cooking time. The energy needed to evaporate m_s kg of water during the simmering period (one hour) can be calculated from the energy equation

$$Q_{into\ pan} = Q_{steam}$$

$$P_{min} * 10^3 * \eta/_{100} * t_s = m_s \ h \qquad (7.6)$$

In this formula t_s stands for simmering time (s) and h for the specific heat of evaporation (2.257 10^6 J/kg).
The amount of steam formed is thus

$$m_s = P_{min} * \eta * 1.595 \ 10^{-2} \qquad (7.7)$$

The initial amount of water m_i (kg) is the sum of the steam produced and the water absorbed by the lentils.

$$m_i = m_s + 1.44 * m_1 \qquad (7.8)$$

Now we can calculate the duration of the heating up period t_b (s) from an energy balance

$$P_{max} * 10^3 * \eta/_{100} * t_b = (m_1 * c_1 + m_i * c_w) \ \Delta T \qquad (7.9)$$

Here ΔT stands for the temperature difference, which is taken as 80 K; c_1 and c_w for the specific heat of lentils and water respectively (1.84 10^3 J/kg and 4.2 10^3 J/kg). For the time t_b this gives

$$t_b = \frac{(m_1 * 6.31 + m_s * 3.36) * 10^4}{P_{max} * \eta} \qquad (7.10)$$

The total amount of wood consumed during this task (M_{wood}) is

$$M_{wood} = \frac{P_{max} * t_b + P_{min} * t_s}{B} \qquad (7.11)$$

The combustion value of the wood is indicated by B (J/kg). Substituting equations (7.6) till (7.10) for the total wood consumption ($B = 18.7\ 10^6$ J/kg)

$$M_{wood} = \frac{m_1}{\eta}\ 3.374 + P_{min} * .2212 \qquad (7.12)$$

The total time (t_{total}) needed to perform this task is the sum of the heating up period and the simmering period.

$$t_{total} = t_b + t_s \qquad (7.13)$$

With formulas (7.7), (7.8) and (7.10), found for the simmering and heating up period, (7.13) can be expressed in the variables of the stove and cooking task

$$t_{total} = \frac{m_1}{P_{max}\eta} * 6.312\ 10^4 + \frac{P_{min}}{P_{max}} * 5.3592\ 10^3 + 3.6\ 10^3 \qquad (7.14)$$

The total required energy is found by the mass of wood from formula (7.12) and the combustion value of the fuel used.

We see, that the total fuel consumption is determined by the minimum power, the efficiency of the stove and the amount of lentils to be cooked. The maximum power of the stove has no influence on the fuel consumption. This value only influences the cooking time. In fact this is true because we stated a constant efficiency, in reality the maximum power will also influence the total energy consumption per cooking task.

We will now apply these formulas to three stoves for different amounts of lentils. The three stoves chosen are the Shielded Fire, the Heavy Experimental Stove and the Tungku Lowon Stove. Reasons for the choice are: the Shielded Fire has a high efficiency, the Heavy Experimental Stove has the greatest power range and the Tungku Lowon Stove has the lowest minimum power.

For these three stoves the following data have been used

	P_{min} (kW)	P_{max} (kW)	η(%)
HES	2.2	12	26
TL	1.8	7	17.5
SF	2.5	6.7	43

Table 7.2: The efficiency and powers of the three stoves.

In figure 7.11 the total cooking time and the total energy consumption is plotted as a function of the amount of lentils to be cooked. It is remarkable, that for small amounts of lentils the Shielded Fire has the highest fuel consumption. This can be explained by noting that during the simmering time, due to the high efficiency, more water is evaporated compared to other stoves. This implies that during the heating up period relatively more water must be heated if we desire the same final quantity of cooked food. Another striking fact is the exceptionally long cooking time for the Tungku Lowon Stove when large amounts of lentils are cooked. This is only due to the long heating up period (simmering period is taken constant) and thus due to low efficiency and low maximum power of the stove.

In practice there will be some departures from the results presented above. Firstly non constant efficiencies with varying powers will result in some changes. Secondly it is improbable that the same pan will be used when m_1 is changed from 1 to 10 kg. This will have a much stronger effect on the efficiency.
In spite of these limitations, the calculations show the importance of designing stoves for low power levels so that the simmering operation could be carried out with minimum amount of fuel.
The several graphs of figure 7.11 all start with 0.5 kg of lentils. This amount of lentils is arbitrarily chosen. The maximum amount of lentils which can be cooked by each stove varies. This variation is due to the pan size which has been used for the stoves.

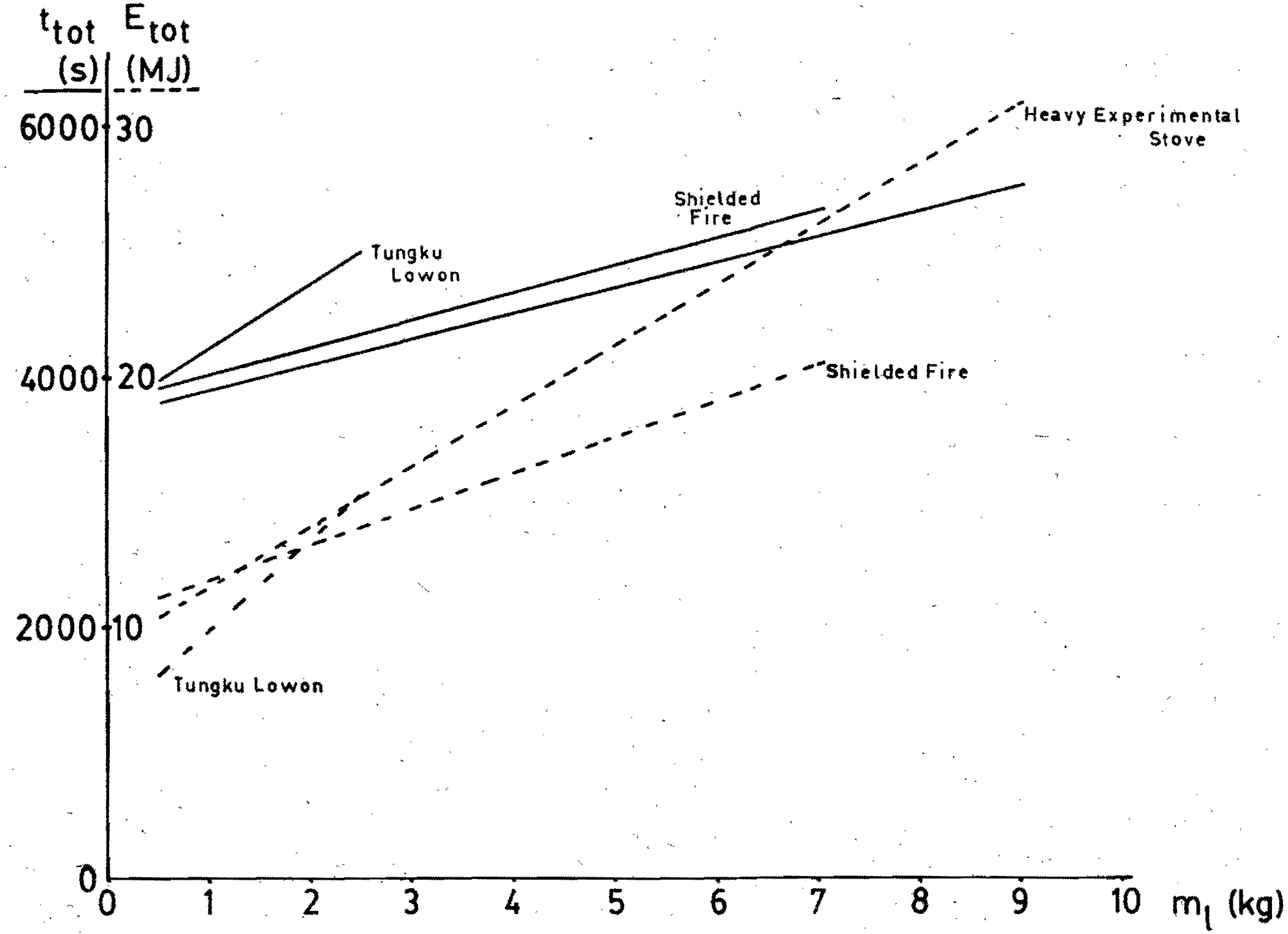

Figure 7.11 : Cooking time and energy consumption for cooking m_l kg of lentils

It was assumed, that in the beginning each pan could not be filled for more than 80% of the total volume. With the derived formulas it was possible to calculate the maximum amount of lentils which could be cooked in one pan. The density of the lentils is taken 1800 kg/m^3 (the volume between the lentils has taken up water!)

7.4 One and two pot stoves

As can be seen in table 7.1 there are stoves with one and two holes. Two pot hole stoves are intended to cook two pans simultaneously. In that case the two cooking processes should be such, that the heating up period as well as the simmering period are equal in duration. This will never be the case so that the total efficiency has no practical value. When one has two non "identical" cooking processes it can happen that the second hole is only used part of the time. No use of the second hole is made when one likes to heat some water or milk when there is no real reason to heat anything else. In former days in Europe there was always need of hot water and during colder periods of the year (± 6 months) also need for space heating. The climate in most developing countries does not demand such use. In contrast with stoves, which are intended for cooking two pots simultaneously, there are also two pot stoves which have a second pot to enhance the total efficiency. This is really a bad way of improving stoves. Probably in most cases two one pot stoves (one with a rather high power and one with an intermediate power) perform better when they are used simultaneously (for detailed calculations see Prasad 1982). Other advantages of one pot stoves is the lighter weight, compared to two-hole stoves, so they are portable and in addition one can still use one stove when the other must be repaired, while when the two pot hole stove breaks down one has to retreat to the open fire.

To get an idea of the ratio of efficiency between the second pot and the first pot figure 7.12 was drawn. In this figure the ratio between second pan efficiency and total efficiency and the ratio between second pan efficiency and first pan efficiency are plotted.

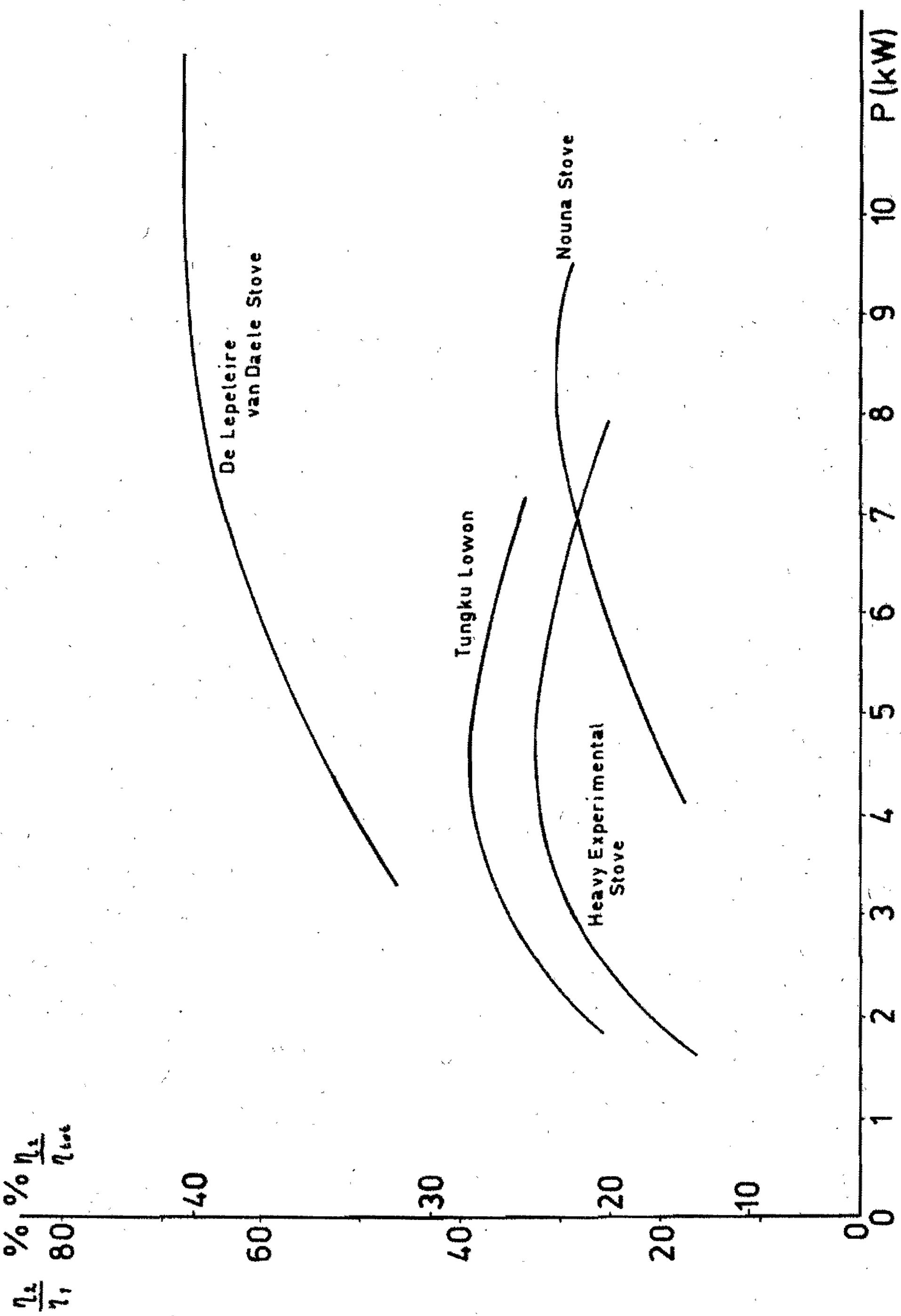

Figure 7.12 : Ratio of the efficiency of the second pan and the total efficiency.

We can see that only the De Lepeleire van Daele stove gives high figures; higher than about 35% for the total efficiency ratio. Because the total efficiency is about 30% this means that the efficiency of the second pan is about 12%, which indicates a rather low efficiency of the first pan of about 18%. From the point of view of the efficiency two pot stoves do not perform better than one pot stoves (figure 7.10), at the same time it appears that the second pot efficiency is not higher than 40% of the total efficiency (figure 7.12), which is not higher than 30 percentage points. This together with the doubts stated before on the effective use of the total efficiency suggests that one pot stoves seem to hold more promise for saving fuel than two pot stoves.

7.5 Some remarks on the combustion quality of stoves

The combustion quality of a stove is a term indicating the completeness of combustion. Products of incomplete combustion can be classified into

(i) carbon monoxide

(ii) hydro-carbon (oxygen?) components

(iii) soot and tar.

The heavy carbon hydrogen components will be partly deposited on the (cold) pan and stove, just like most of the soot and tar and partly leave the stove as smoke. The CO gas is completely emitted by the chimney or other devices by which the smoke can escape.

In general incomplete combustion represents a loss in efficiency. However the general impression that emerges from the testing programme in Eindhoven and Apeldoorn is that products of incomplete combustion account for at most 5 to 10% of the fuel heat content. Thus improvement of combustion efficiency in a stove will not result in significant improvements in fuel economy. However there is another aspect to the problem of combustion quality. Many components of the products of incomplete combustion of wood are known to be carcinogenic. Generally it is assumed that at the present levels of health in the third world countries, this is not a serious problem. The other product of combustion involved is carbon monoxide (CO). This can be serious threat to life. When CO is breathed, this molecule will combine with the oxygen carriers in the blood in such a way, that no oxygen can be transported any more. Figure 7.12a shows the poisoning effect for various CO concentrations. A reasonable number of stoves is operated in closed rooms, mostly small and the smoke is released in the same room. From figure 7.12a it will be clear that severe poisoning can result. In figure 7.13 measurements of CO concentrations are shown for various stoves as a function of the power output. When one states that the CO concentration in the flue gasses may not be higher than .5% the maximum power of the stoves need to be restricted. We see from figure 7.13, while the Nouna Stove is not affected by this consideration, the other two are affected quite seriously. Tungku Lowon is the worst of the three in this respect. Thus the designer is required to specify a new power rating of the stove, which may be called the carbon monoxide limited power level (analogous to the smoke limited power output of a diesel engine).

The practical implication of this is that it is inadvisable to operate the Tungku Lowon indoors to bring 5 litres of water to boil in half an hour. If this is an important function that a customer desires, it is preferable to scale up the design so that twice the present power level from the stove could be obtained.

7.6 Influence of several stove parameters

7.6.1 *Introduction*

In addition to power range and combustion quality investigations have also been done on the influence of several stove parameters on the efficiency, formation of products of incomplete combustion and excess air factor. As explained in section 7.3, it is very important that a stove has a power range and that this power range matches a cooking task.

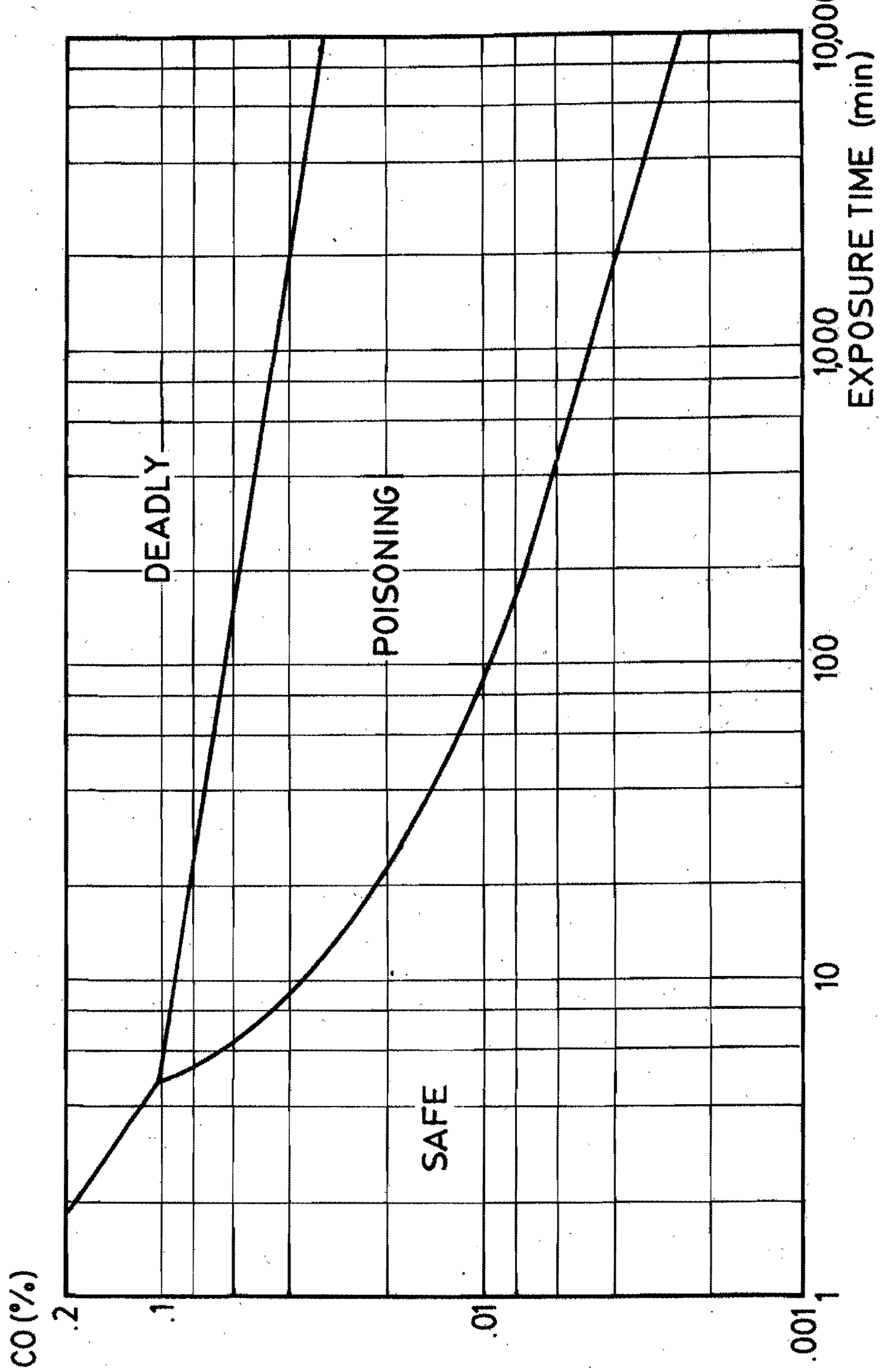

Figure 7.12a : Toxicity of CO concentration as a function of the exposure time.

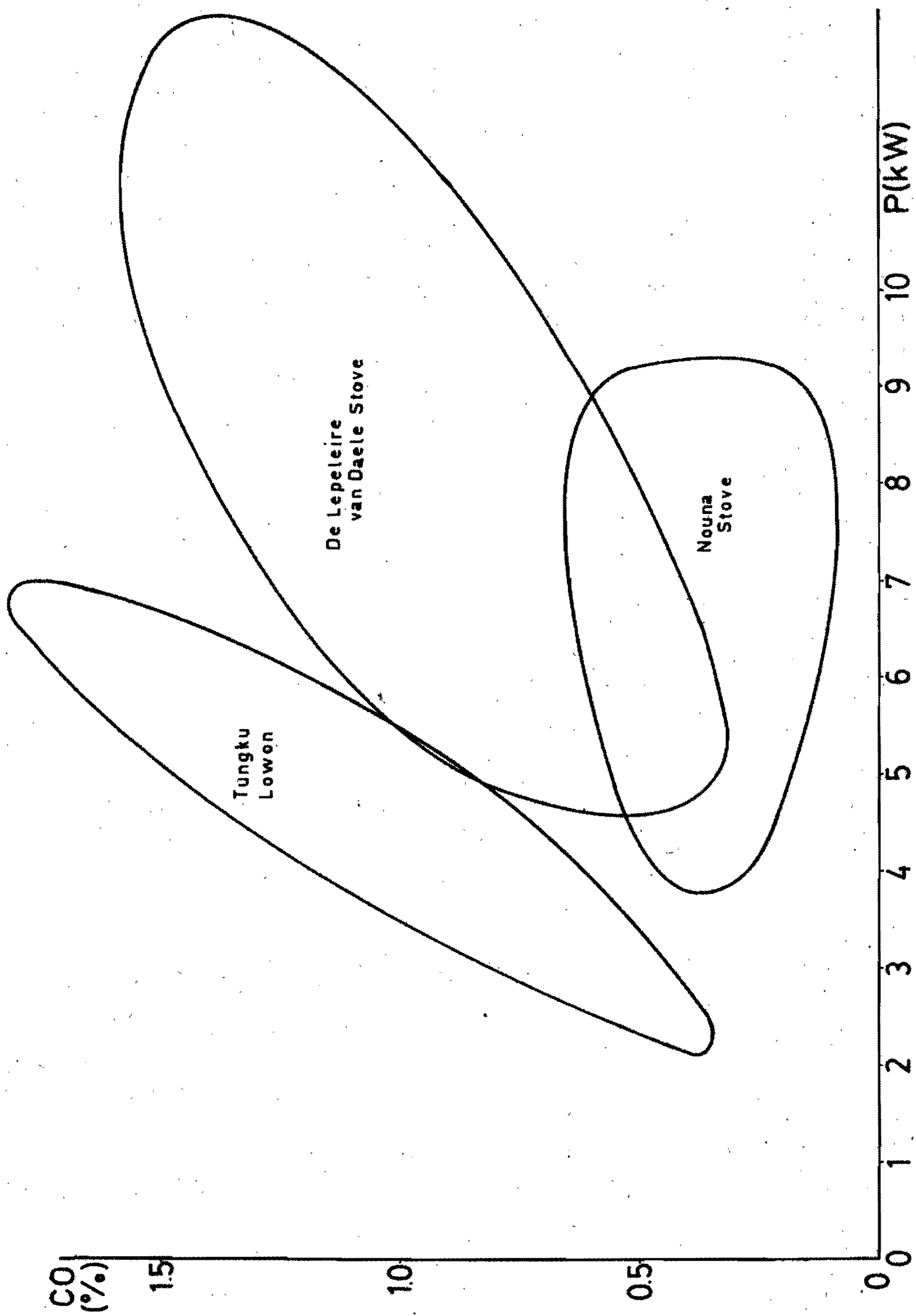

Figure 7.13 : The CO concentration in the flue gas as a function of the power.

In this section however only tests are described with a constant power input. This constant power input means that several charges of wood of the same weight are added at certain fixed intervals of time. When the water starts boiling maybe one or two more charges are added. When the water stops boiling, the test is finished.

Each of the three characteristic stove magnitudes; power range efficiency and combustion quality, can be influenced by several parameters. These parameters can be stove dependent parameters or parameters which depend on the fuel use.
The parameters related to the stove geometry are:

- distance between (first)pan and grate
- depth of the pan in the stove
- air supply
  - primary
  - secondary
- chimney
  - length
  - damper
- distance between second pan and baffle (if present)

The parameters related to fuel use are

- fuel species (wood)
- moisture content of the fuel
- dimensions of the wood blocks
- charge size (Δm_f)
- charge time (t_c)
- power input

The last three parameters are related through the following relation:

$$P = \frac{\Delta m_f \cdot B}{t_c}$$

It is thus possible to vary P by either varying Δm_f holding t_c constant or vice-versa. From the point of view of the user, it is much more convenient to have a stove that permits large values of t_c.

The implication of this is that the stove does not require frequent attention to maintain a power level that is demanded by the cooking process. Thus the study of the influence of Δm_f on the stove performance is quite important.

7.6.2 *The distance between pan and grate*

The influence of the distance between the pan and the grate on the efficiency of an open fire has been studied in detail by Bussmann et al (1982). Here we shall restrict ourselves to a qualitative discussion of the problem in closed stoves. For this purpose we shall consider a hypothetical stove with a fixed grate position but with the possibility of varying the depth of insertion of the pan into the stove body. When we permit that the grate can be placed on several places an analogous treatment can be given of the observed effects. Now we will first look at effects on the radiative heat transfer when we change the distance L between the pan and the grate. First of all the view factor from the fuelbed to the pan will change. This view factor is a measure of the radiation leaving the fuelbed and incident on the pan. When L becomes larger, the view factor will become smaller (see Eckert and Drake, 1959 for details on the view factor between two concentric circles).

Another view factor which will also become larger is the view factor between fuelbed and stove wall and thus stove wall and pan when the distance h becomes larger. However the enlargement of the surface A_w is probably more important, because the product of view factor and surface area determine the total radiated energy from the stove wall to the pan. This radiation from the side wall is not negligible compared to the radiation from the fuelbed to the pan because of the rather high wall temperatures. A last item related to radiation is the change of the ratio of the surfaces of the pan inside (A_g) and outside (A_L) the stove. The higher this ratio, the more pan surface is exposed to the environment, the more heat can be lost by radiation to this environment. Delsing calculated the effect of the changing heat flux to the pan when the distance between pan and grate changes (Prasad 1981(b)). However he did not do a proper heat balance on the fuelbed and he calculated a too high fuelbed temperature.
Besides changes in view factor and surface area, which influence the radiation heat transfer there are also changes in gas flow and gas temperature, which influence the convective heat transfer. The relative importance of these two heat transfer mechanisms have been studied by Bussmaan et al. (1982) for the open fire. Similar work for closed stove is being attempted now by the group.

The expected changes in gas flow and temperature are caused by the changing flow resitance in the stove (between pan and stove wall for example). This will cause different excess air factors with different dilution of the gases, temperature changes etc.. When the excess air factor increases there will be more air drown into the stove. This larger amount of air may affect the combustion. Firstly the gas temperature will become lower which could mean worse combustion of the volatiles. The dilution however also lowers the concentration of the carbon monoxide. Secondly it is possible that a higher excess air factor will provide a better chance for complete combustion because there is more oxygen available. This also will cause a lower CO concentration.

The absolute amount of CO produced in the stove depends on the product of the CO concentration and the excess air factor. In first instance it is not clear wether this total amount of CO will be larger with a higher air flow through the stove. In figure 7.14 the influence is shown of the distance between pan and grate for the open fire and several other designs. All these measurements were done on metal stoves and not on brick stoves. From the graphs it is seen, that the distance between pan and grate has a big influence on the efficiency of the stove. It is presumed, that in heavy stoves one also needs to pay attention to this dimension.

In figure 7.15 the effect of the insertion depth of the pan on the efficiency is shown. Only the experimental stove has a fixed distance between pan and grate while the insertion depth H_s is varied. For the other stoves this means, that identical curves are shown in figure 7.14 and 7.15.
In figure 7.16 and 7.17 the excess air factor k and the CO concentration respectively are shown against distance between pan and grate. We see that the excess air factor increases with increasing distance h. The variation of k with the CO concentration is as stated before.

7.6.3 *The air supply control*

Control of combustion air is mainly done by changing the flow resistance in the stove. In principle the flow resistance must be balanced by the chimney draft. We will not discuss in detail how the flow resistance influences the amount of air which flows through the stove but confine ourselves to the different devices which are designed to exert this influence. These devices are the primary air holes, the secondary air holes, the chimney length and chimney damper. Also the distance between the second pan and the baffle influences the flow resistance. When the draft is assumed constant, flow resistance is only dependent on the chimney length and temperature of the gases, we see that the air flow will be higher when there is less resistance. This results in a higher excess air factor with all its possible consequences for the combustion.

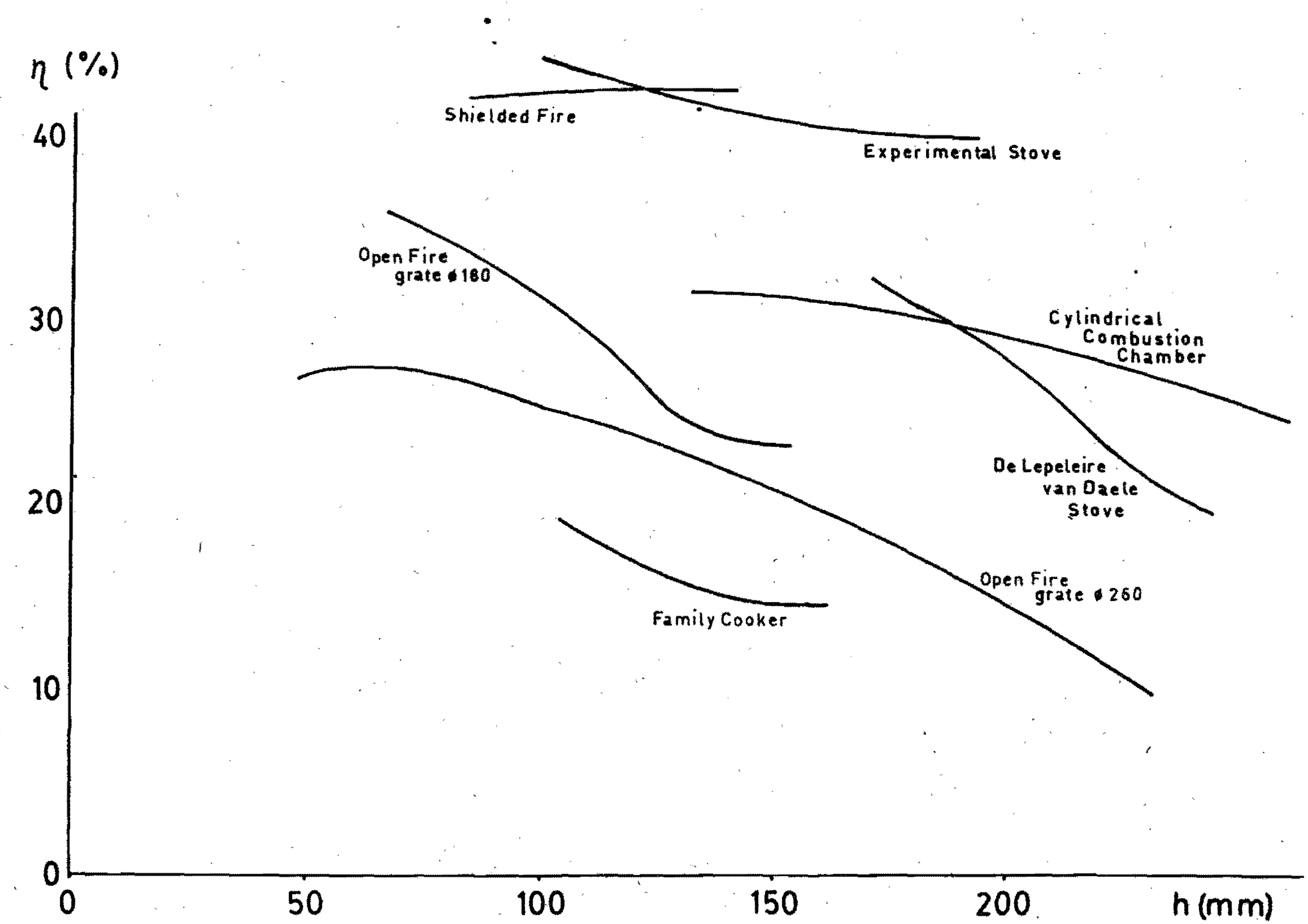

Figure 7.14 : Influence of the distance between the pan and the grate on the efficiency.

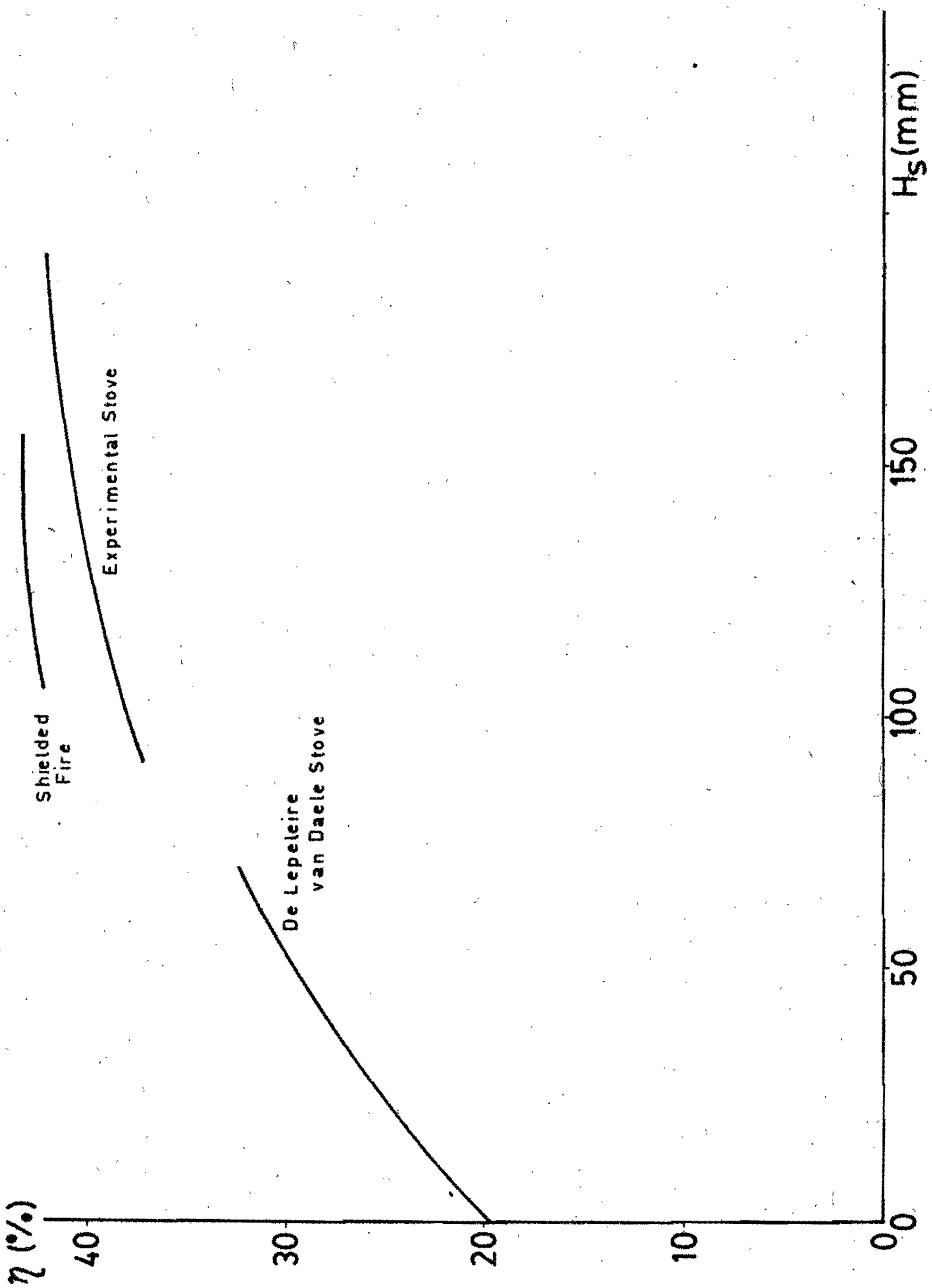

Figure 7.15 : Influence of the depth of the pan in the stove on the efficiency.

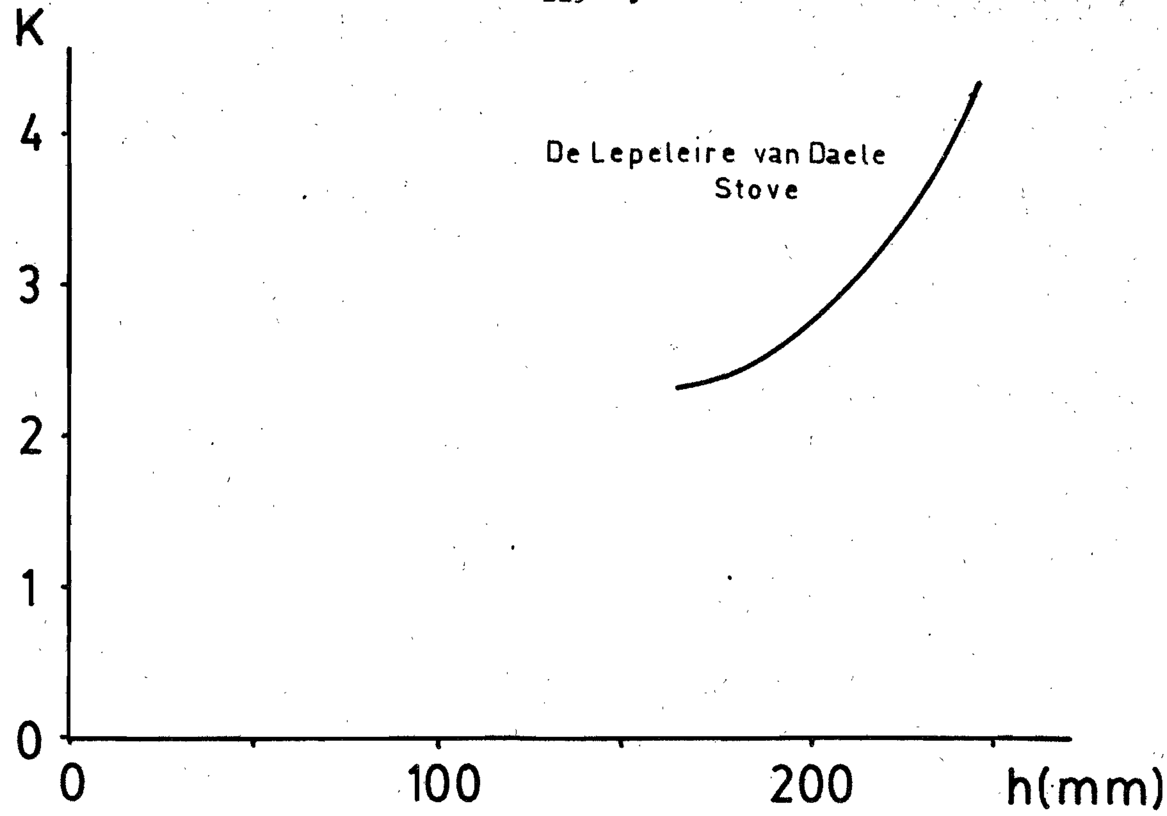

Figure 7.16 : Influence of the height h on the excess air factor.

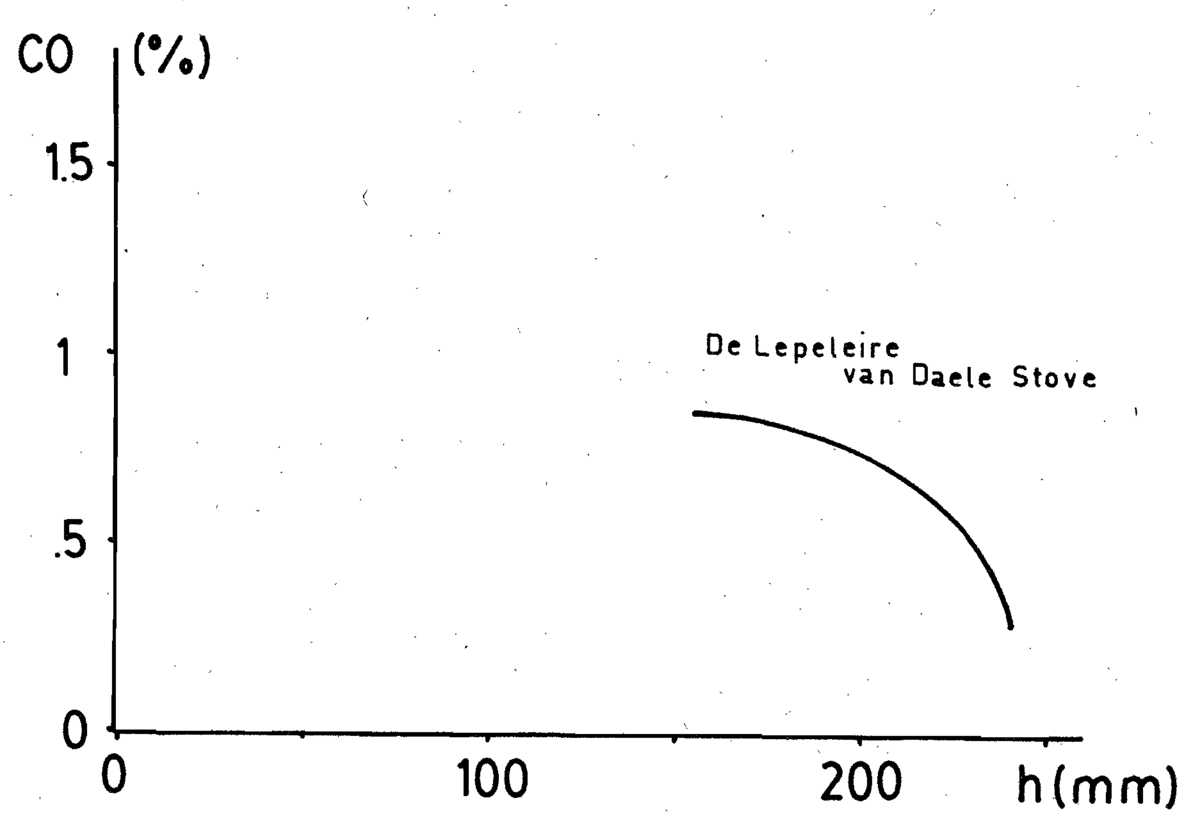

Figure 7.17 : Influence of the height h on the CO concentration.

None of the experiments takes into account, that the air control might have a significant influence on the power range of a stove. In the previous section we already discussed the influence of the excess air factor on the formation of CO and on the concentration of carbon monoxide.

The influence of the primary air control on the efficiency is shown in figure 7.18. The primary air is expressed in the relative opening of the primary air holes. From the graphs it will be clear, that for the designs considered this parameter has little influence on the stove efficiency. In figure 7.19 the efficiency is plotted against the secondary air supply. Also in this graph no influence is seen. The data for the Nouna Stove show considerable scatter and as such it is shown as a region in figure 7.19.
From figures 7.18 and 7.19 we see that the air flow through the stove is unaffected by the air controls. This can also be seen from the excess air factor k. This parameter is plotted in figure 7.20 as a function of the air hole opening. (Only the Nouna Stove and the De Lepeleire van Daele Stove have air hole control possibility). Except for very small openings of the air holes the excess air factor is practically constant. Figure 7.21 shows the CO concentration. Althrough the scatter in results with the De Lepeleire van Daele Stove is quite large, similar trends are observable for both Nouna and the De Lepeleire van Daele Stove.

In figure 7.22 the influence is shown of the chimney damper on the efficiency. Until now these experiments have only been done with the Family Cooker. In the graph 0% open means totally closed and the smoke can only escape by leaks. Except for very small openings no influence is seen; below a threshold value of the damper opening, efficiency drops with decreasing damper openings. In general one can conclude that the devices especially made to control the air flow through the stove do not work. Compared to the distance between the pan and the grate this influence is negligible and the distance h seems a more useful device to control the air flow through the stove.

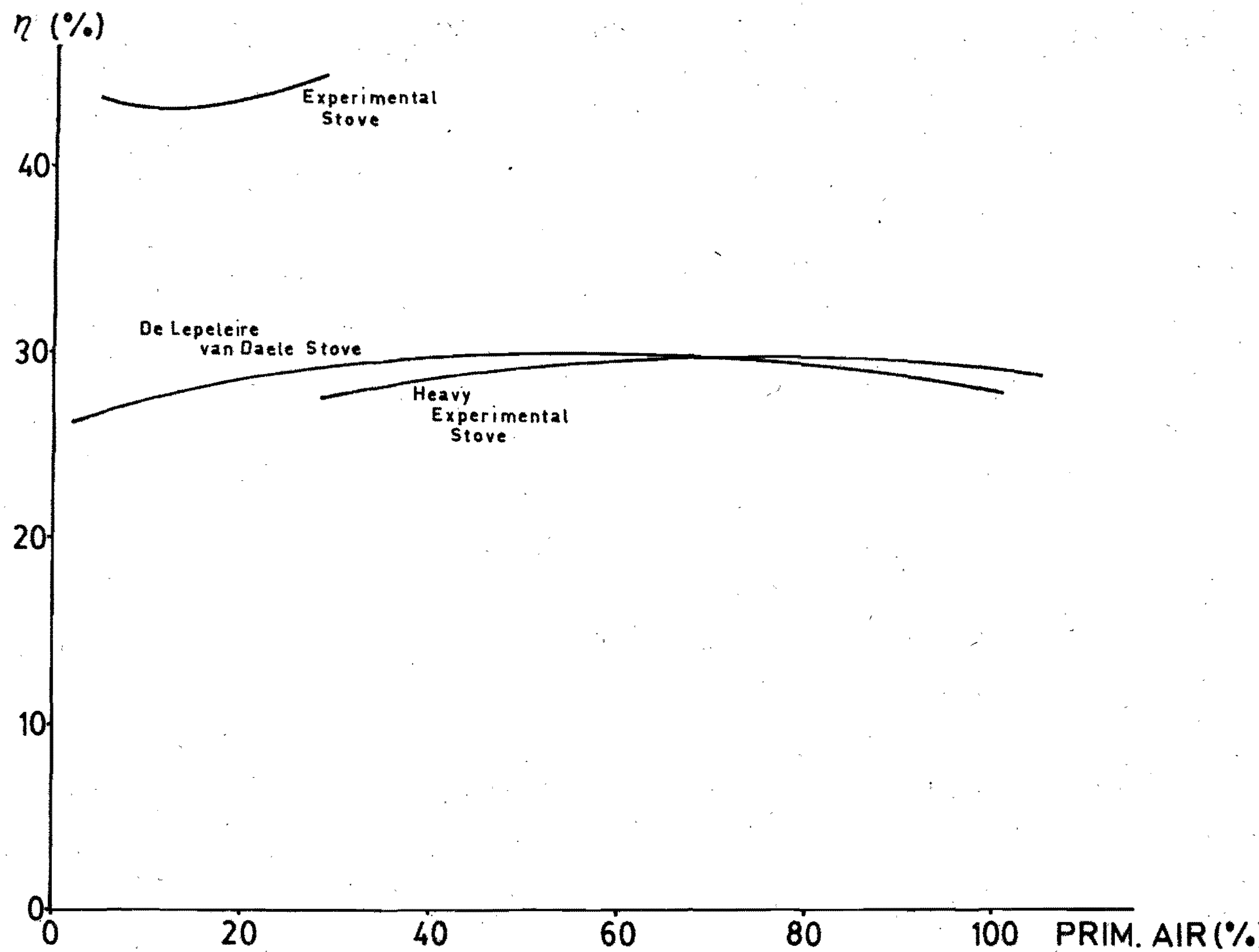

Figure 7.18 : Influence of primary air openings on the efficiency.

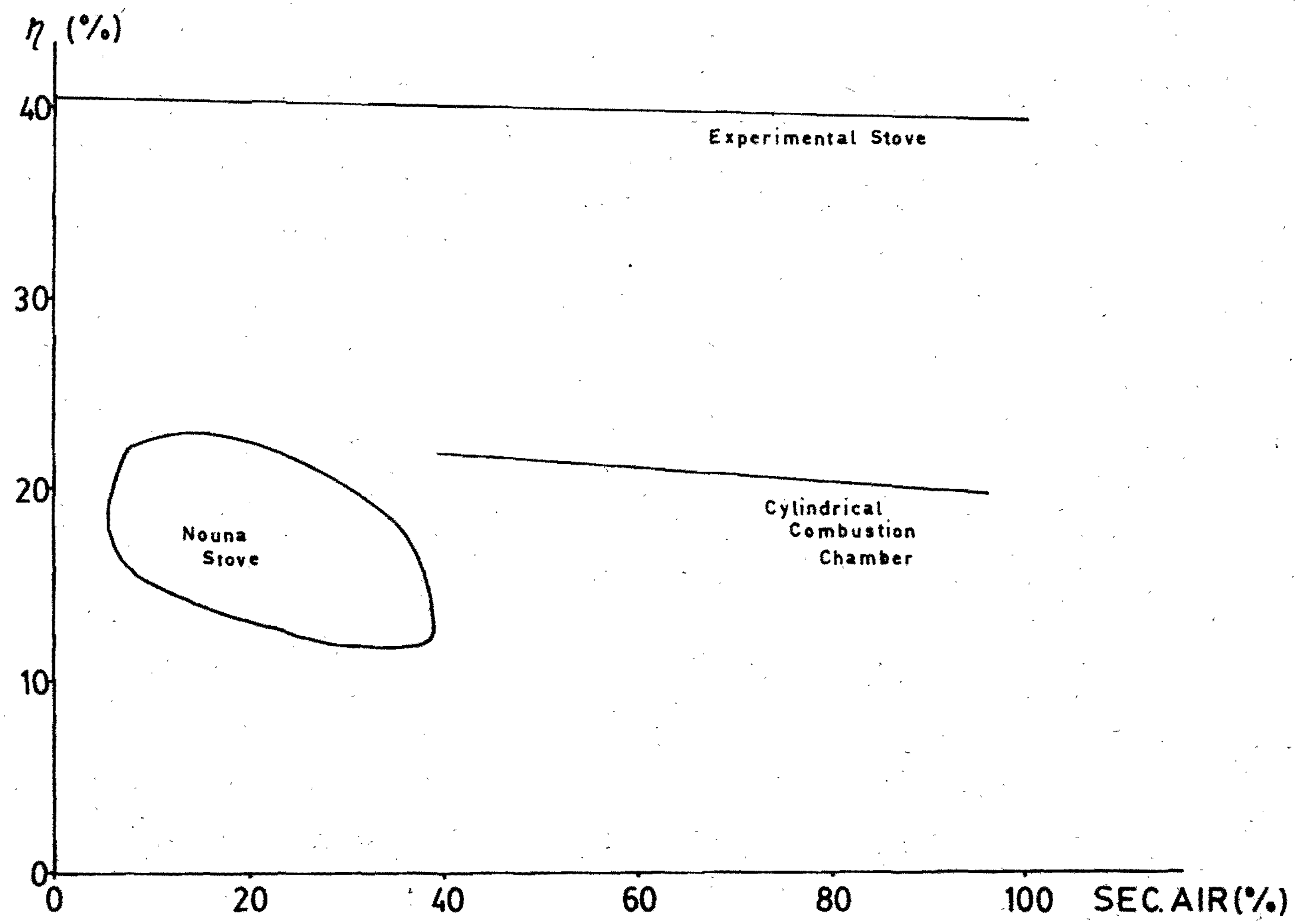

Figure 7.19 : Influence of secondary air openings on the efficiency.

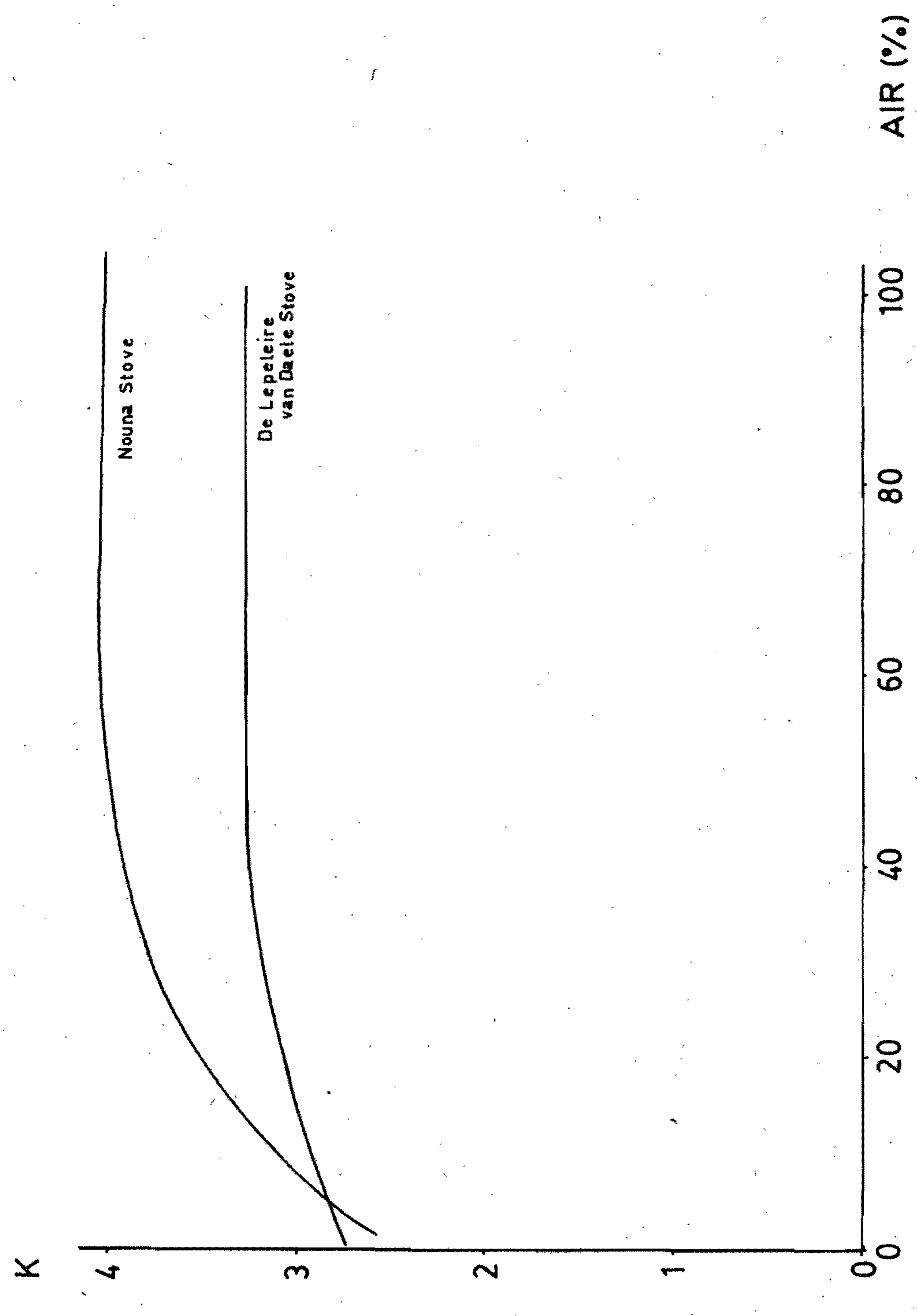

Figure 7.20 : Influence of air control holes on the excess air factor.

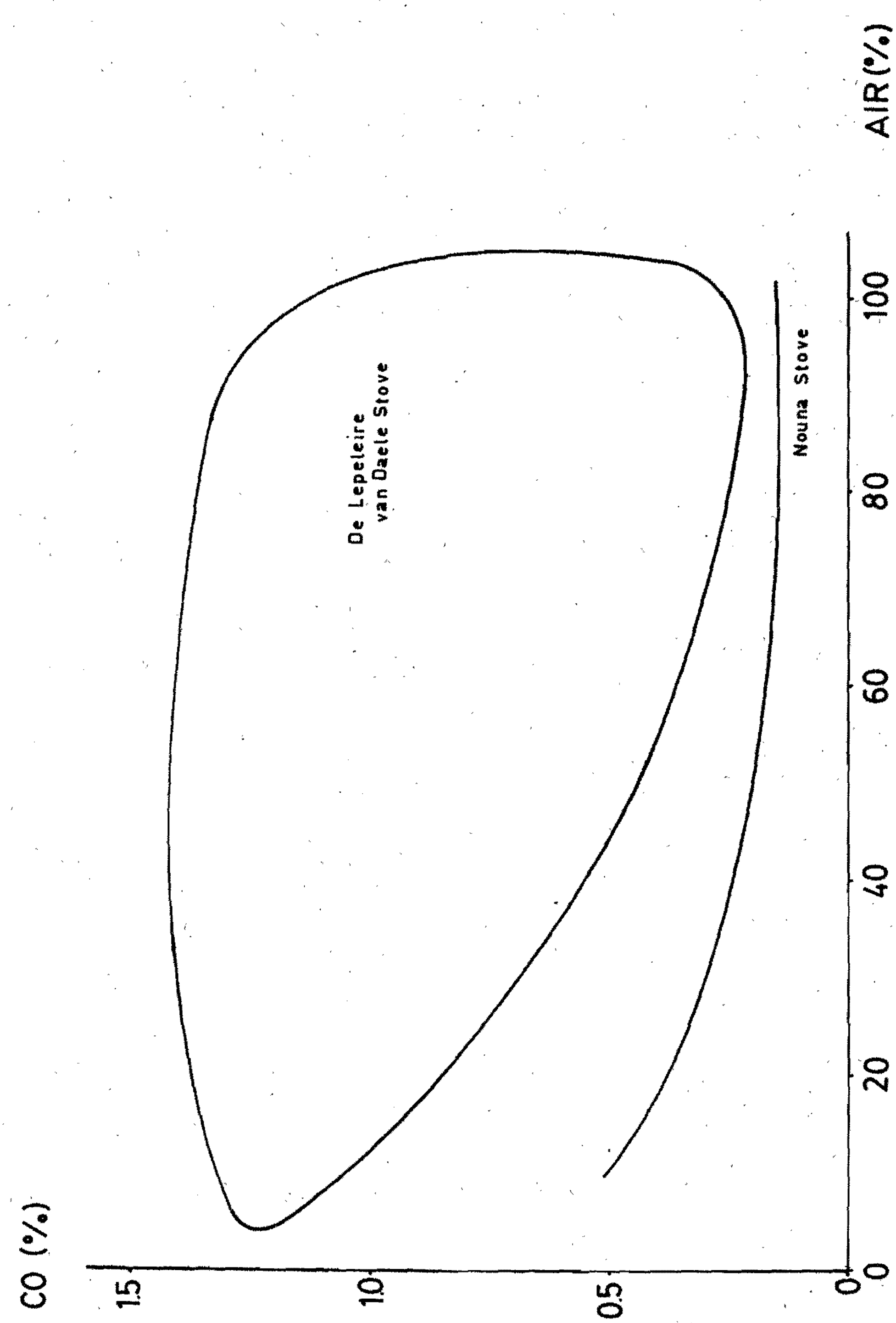

Figure 7.21 : Influence of the air control holes on the CO concentration of the flue gases.

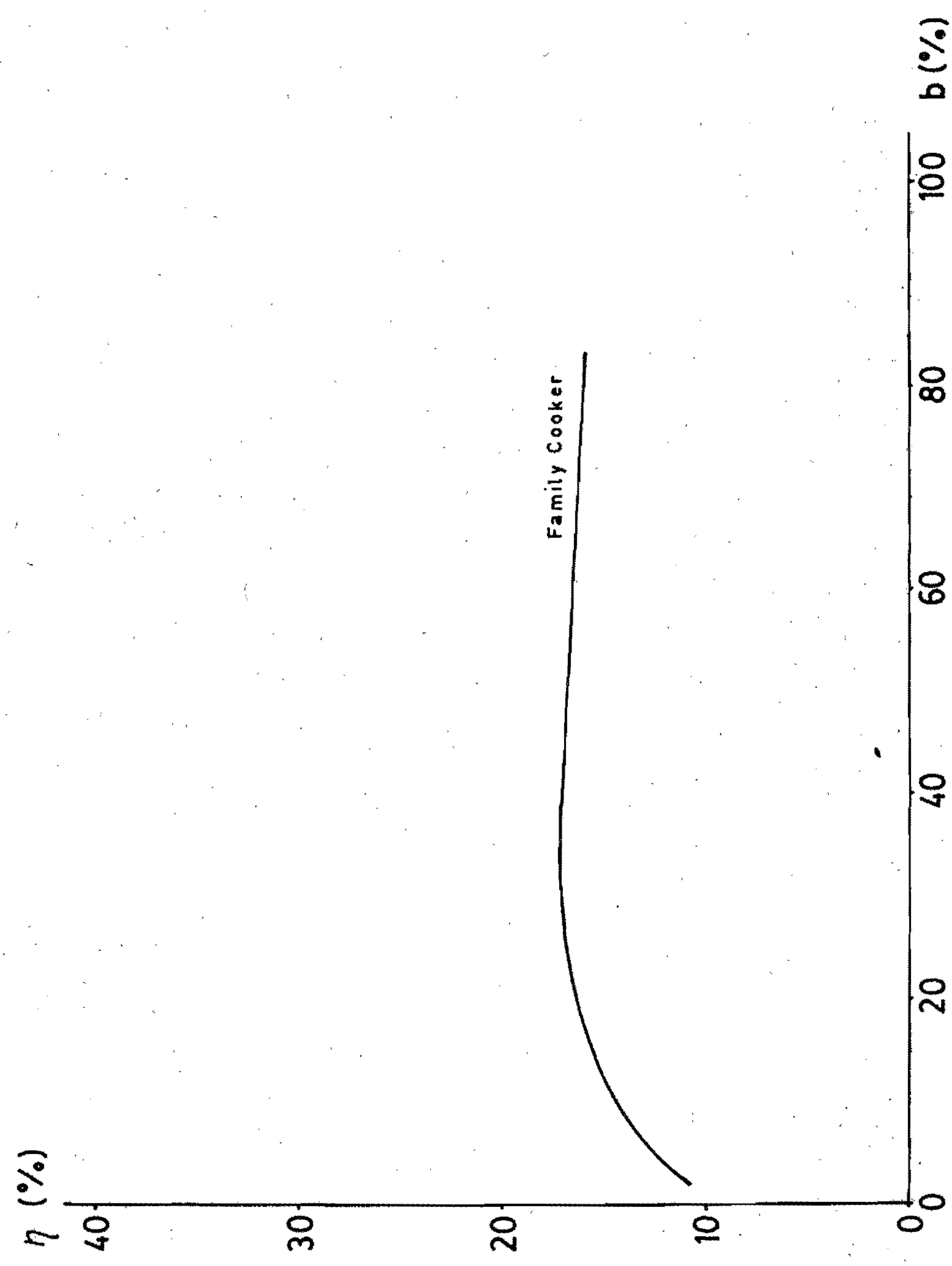

Figure 7.22 : Influence of chimney damper on the efficiency.

Until now only few preliminary tests have been done with varying chimney length. Tests which have been done with the Heavy Experimental Stove, are shown in figure 7.23. We see an increase in efficiency with increasing chimney length. No measurements on the excess air factor have been performed along with these tests, so no explanation of the increasing efficiency can be given. It is expected, that more air passes the stove because of the higher draft.

The distance h_b between second pan and baffle is varied for the Heavy Experimental Stove for four experiments only. The results are shown in figure 7.24. From the graph it is seen, that in particular the second pan efficiency is influenced by this parameter. The efficiency of the first pan is hardly effected.
A distance of about 20 mm seems to be an optimum value for this parameter. Probably with smaller values the flow resistance influences the efficiency in a negative way. Here again no excess air factor is measured, so that the air flow through the system, resulting from the resistance under the second pans cannot be shown.*

7.6.4 *Wood properties*

In this section we will deal with the influence of several wood properties on the stove efficiency and combustion. In Chapter 9 this will be done in detail for the Nouna Stove. Properties which we will discuss are the moisture content, the wood species and the dimensions of the wood pieces.
Bussmann et al. (1982) already paid much attention to the effect of moisture content of the wood on the efficiency and on the problems of lighting the fire. Test on stoves showed, as far as efficiency is concerned, the same tendency with varying moisture content. Hardly any influence has been noticed. In figure 7.25 this effect is shown. In figure 7.26 the influence of the moisture content on the CO concentration is shown. We see that these measurements are only performed for the Nouna Stove. The CO

* See for more details on the effect of this parameter on the Nouna Stove (Chapter 2).

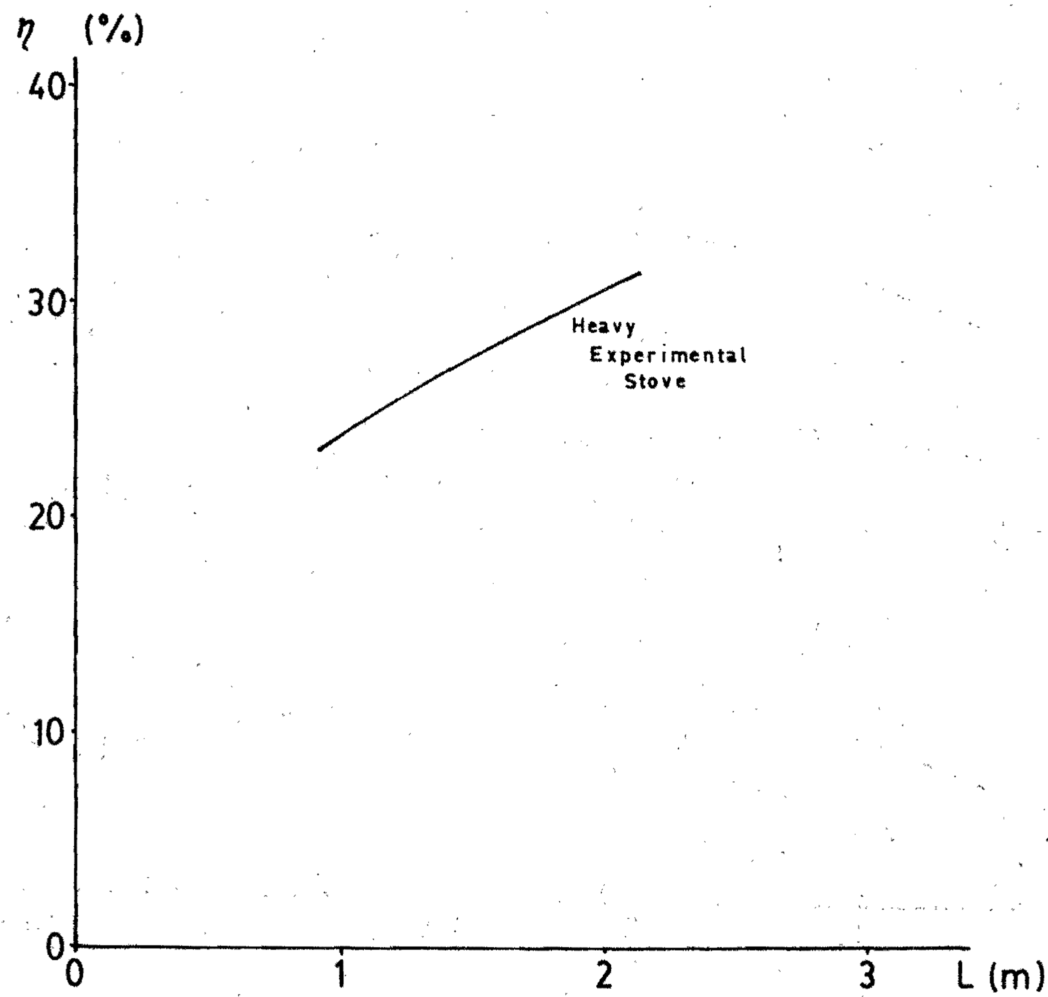

Figure 7.23 : Influence of the chimney length on the efficiency.

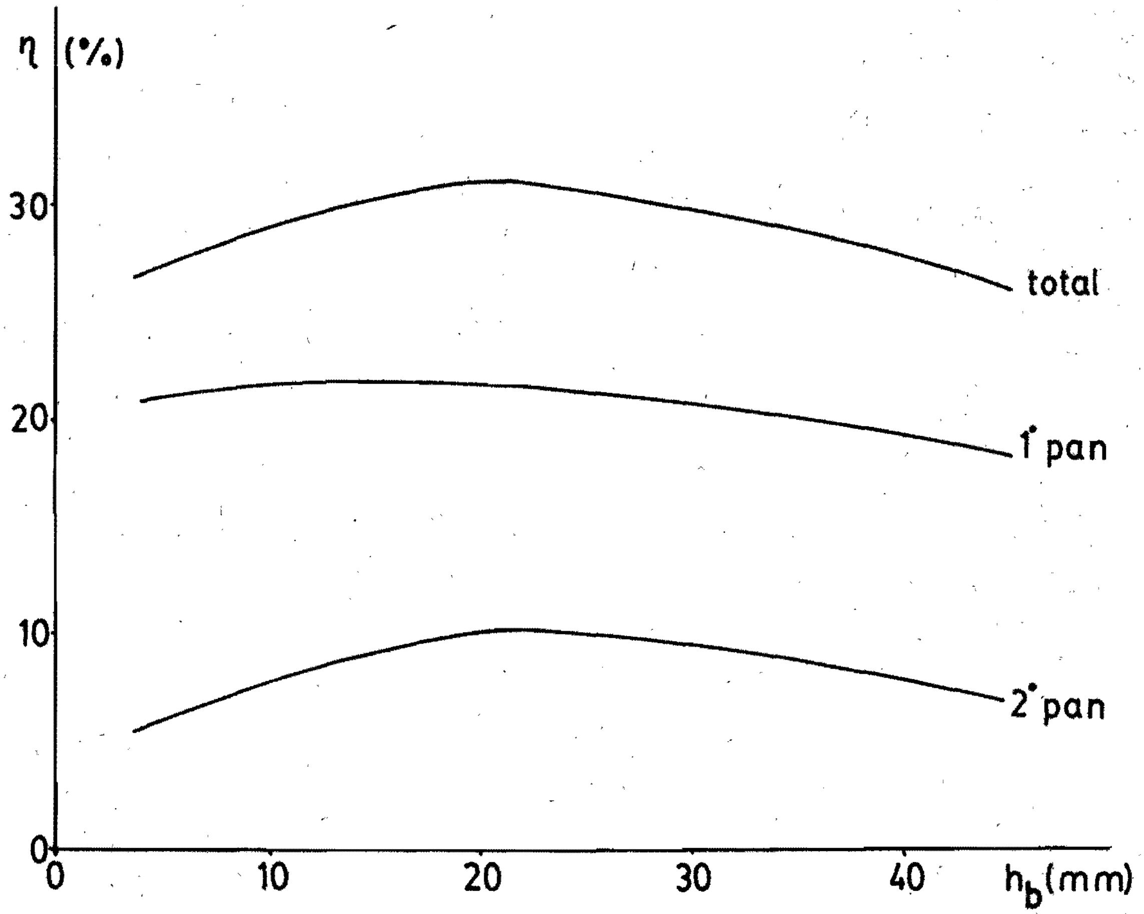

Figure 7.24 : Influence of baffle 2^e pan distance on the efficiency.

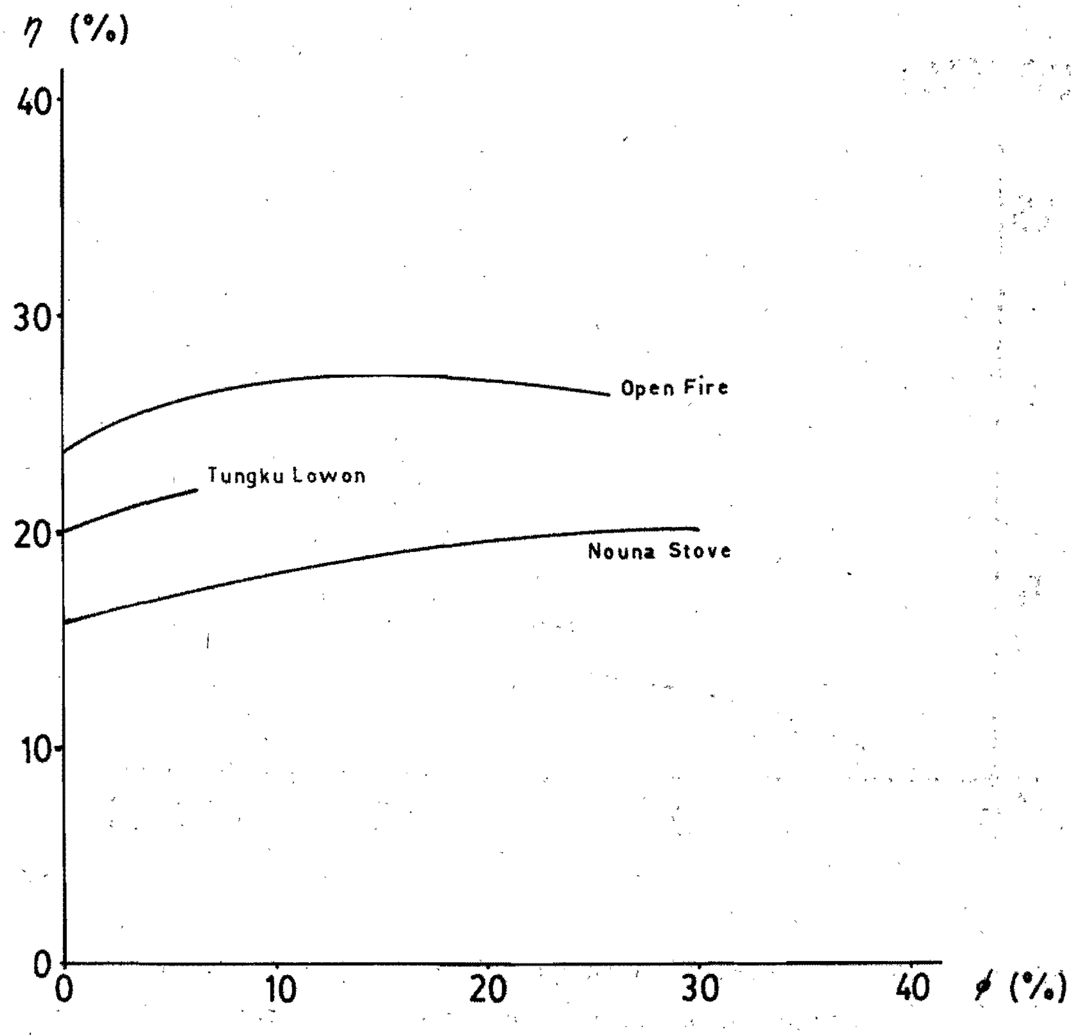

Figure 7.25 : Influence of the moisture content of the wood on the efficiency.

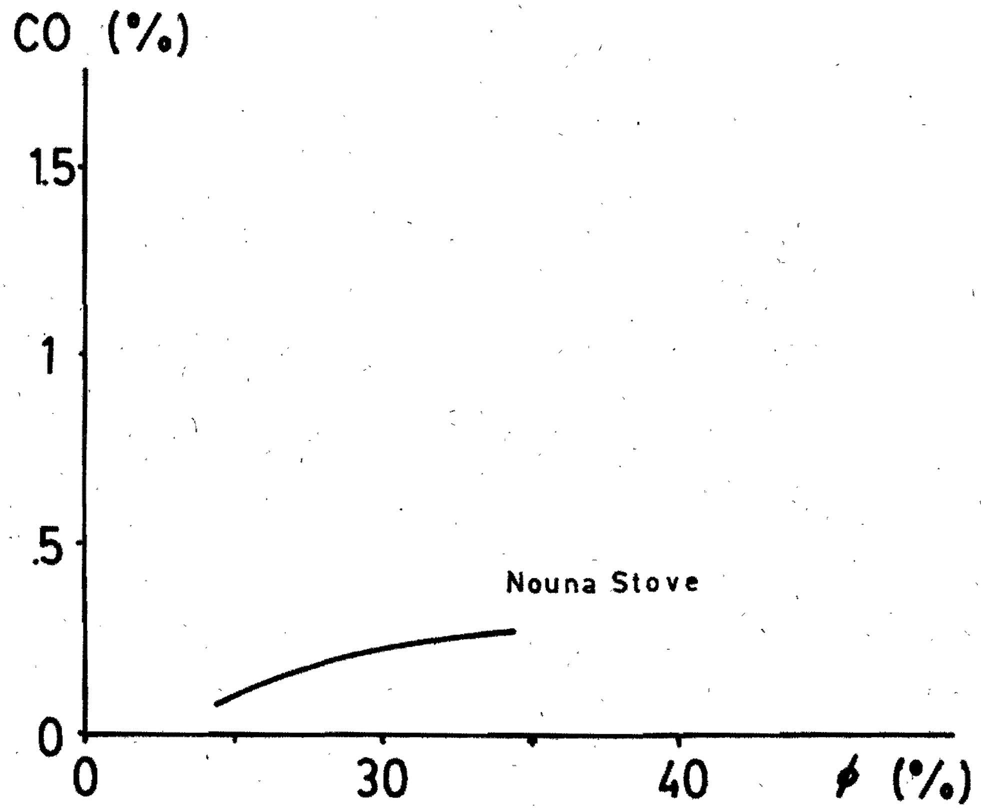

Figure 7.26 : The variation in the CO concentration of the flue gases for varying moisture content of the wood.

concentration increases somewhat with increasing moisture content. In figure 7.24 and 7.25 the moisture content ϕ is expressed as a percentage on a dry weight basis.

The wood species can be characterized by its density. In figure 7.27 measurements are shown for three stoves. Table 7.3 shows the various wood species used, their density, their combustion value and the stoves in which they have been tested. The effect on the combustion quality of the density of the wood has been studied systematically only for the Nouna Stove. Measurements however show so much scatter, that no clear trend could be observed (see figure 7.28).

Wood species	Density kg/m^3	Combustion value MJ/kg	Open fire	Tungku Lowon	Nouna Stove
White Fir	400	18.7	x	x	x
Jelatong	440	17.8	x	x	
Iroko	580	18.1	x		
Meranti	600	18.3	x		x
Oak	620	16.9	x		
Beech	650	17.0	x		x
Merbau	850	18.1	x		x

Table 7.3 The wood species, their characteristic dimensions and the stoves in which they have been tested.

The wood block dimensions are varied for three stoves. (F.C., N.S., L.v.D) and the open fire. It is possible to present the results in two ways, efficiency versus length of the wood or versus volume-surface ratio. The last option is chosen and the graphs are shown in figure 7.29.
One can see that hardly any influence can be shown for the Nouna Stove, while for the Family Cooker and De Lepeleire van Daele Stove a higher efficiency is measured with bigger wood blocks. Results for the De Lepeleire van Daele Stove showed much scatter.

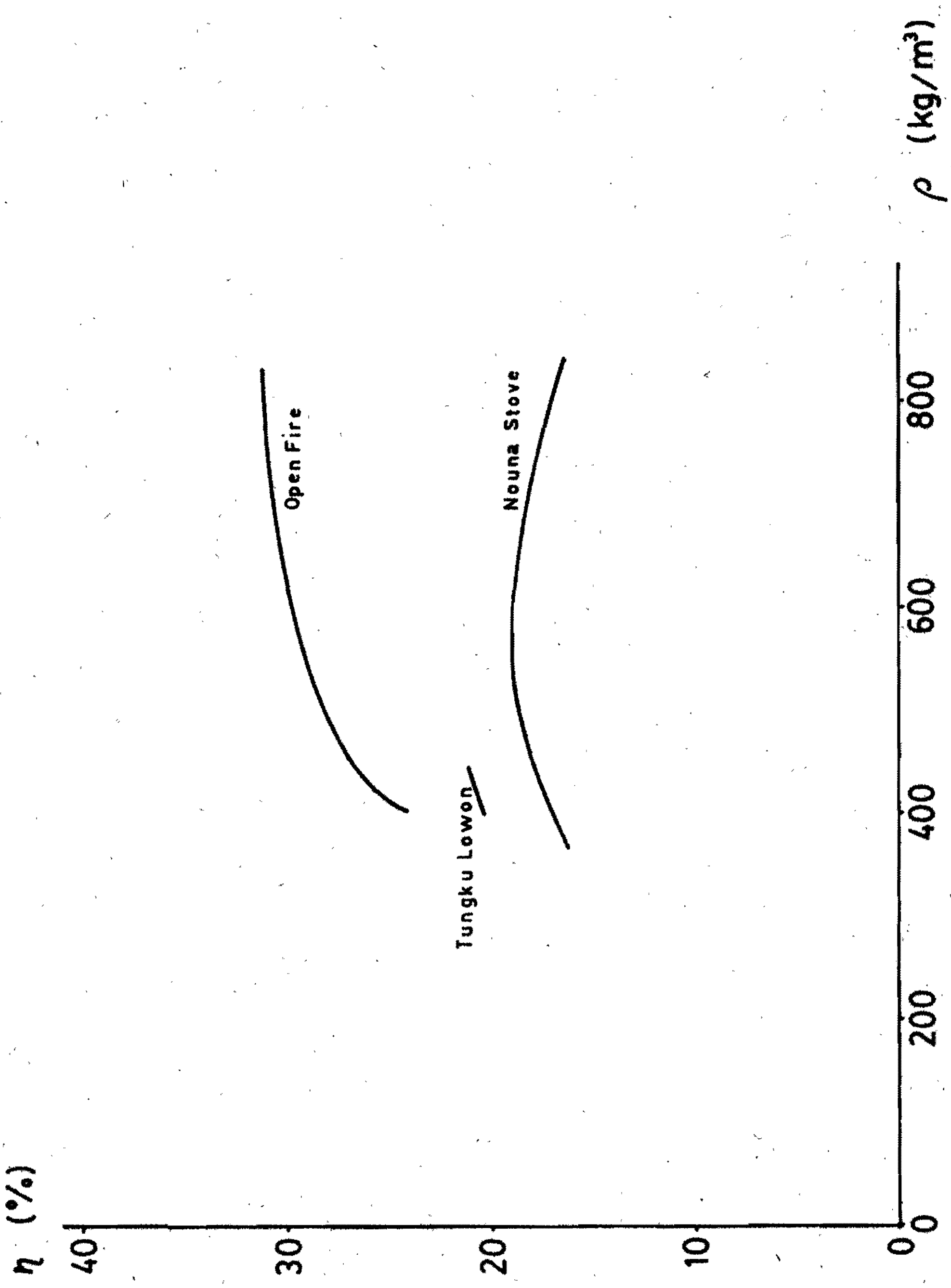

Figure 7.27 : Influence of the wood species on the efficiency.

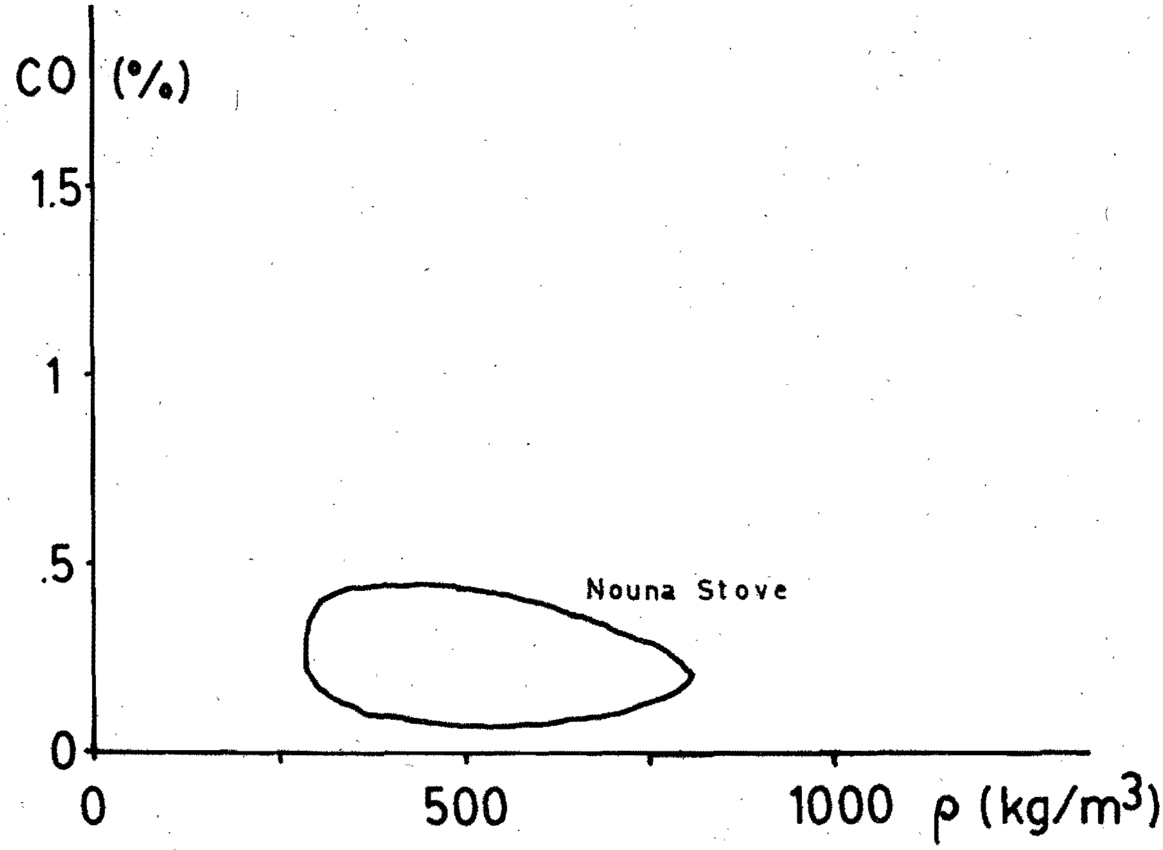

Figure 7.28 : Influence of the kind of wood on the CO concentration.

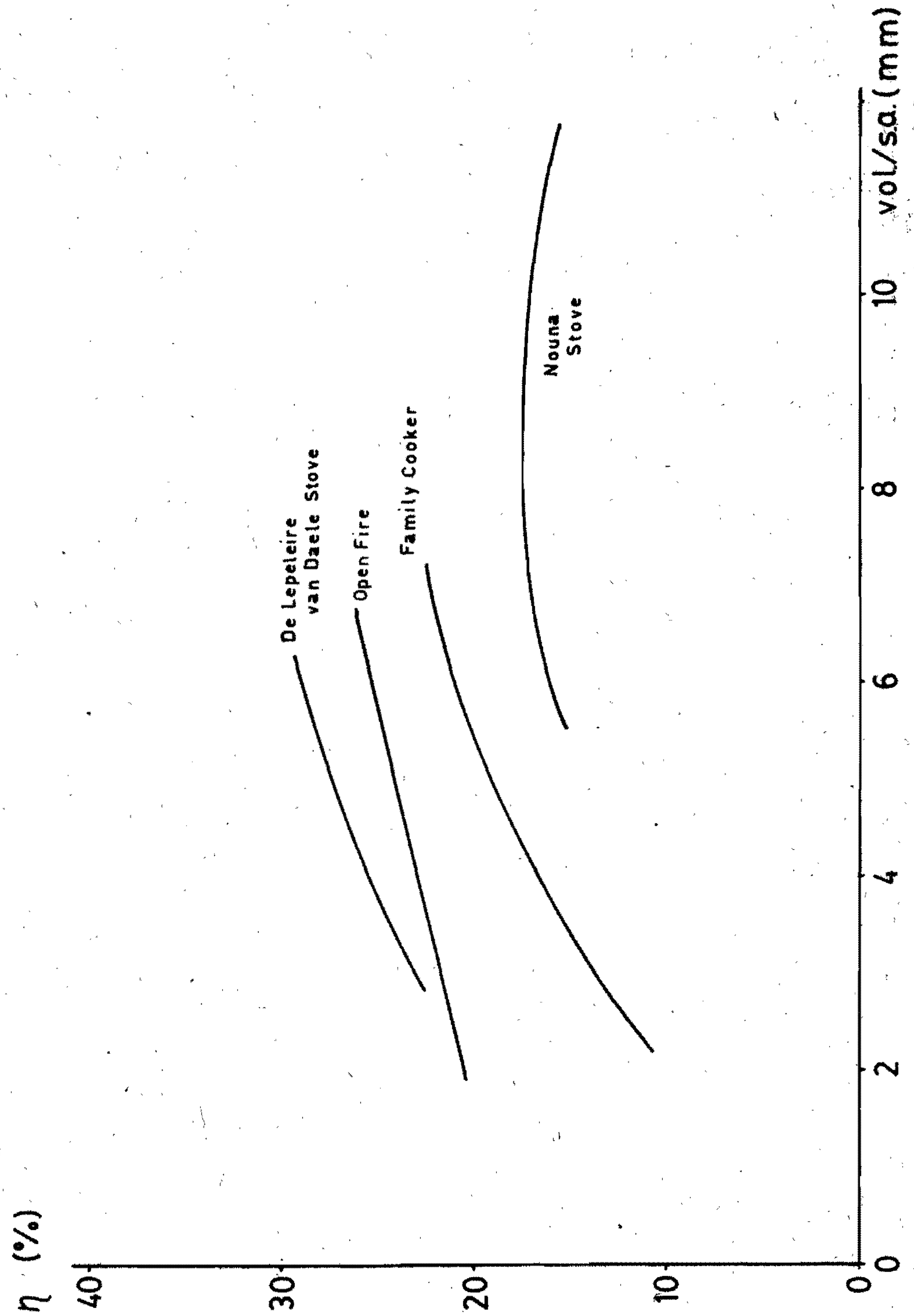

Figure 7.29 : Influence of the fuel dimensions on the efficiency.

The graph for the open fire is based on two measurements, that is why a straight line is drawn. Effect on the combustion quality of the fuel dimensions is only measured for the Nouna Stove. The results are presented in figure 7.30. The curve is based on measurements with three kind of volume-surface ratios. Two ratios however were so close to each other (7.75 and 7.9) that the line is based on two measurements only.

7.6.5 *Power range and charging intervals*

Before we give the results for all the stoves, we first concentrate on the De Lepeleire van Daele Stove, just for the simple reason that a lot of experiments have been done with this stove.
In figure 7.31 all power-efficiency results are shown together with a curve which fits these points. This curve gives a maximum efficiency for a power between 7 and 8 kW. To find out what the best charge and chargetime for this power is, we plot the charge and chargetime versus efficiency for all the measurements with a power between 7 and 8 kW. This is shown in figures 7.32 and 7.33. From these figures we find an optimal charge of 230 grams and an optimum chargetime of 550 seconds. These two numbers correspond to a power of 7.8 kW, which corresponds well to the design power as was deduced from figure 7.31. This seems to be a reasonable procedure to determine the optimal charge and chargetime for a stove. It is however not done for the other stoves because the experiments are too few, so no reliable optimum can be determined.

In figure 7.34 all power efficiency curves are shown (this figure is identical to figure 7.10). The curves are drawn on a identical way as was done in figure 7.31. As one can see some stoves have a lower efficiency than the open fire. Power ranges of the stoves vary from less than three to six kW (Heavy Experimental Stove). The dependence of excess air factor and CO percentage with the power of the stove is shown in figures 7.35 and 7.36 respectively. The magnitude of the CO percentage and the possibility of introducing a CO percentage dependent power has already been discussed earlier.

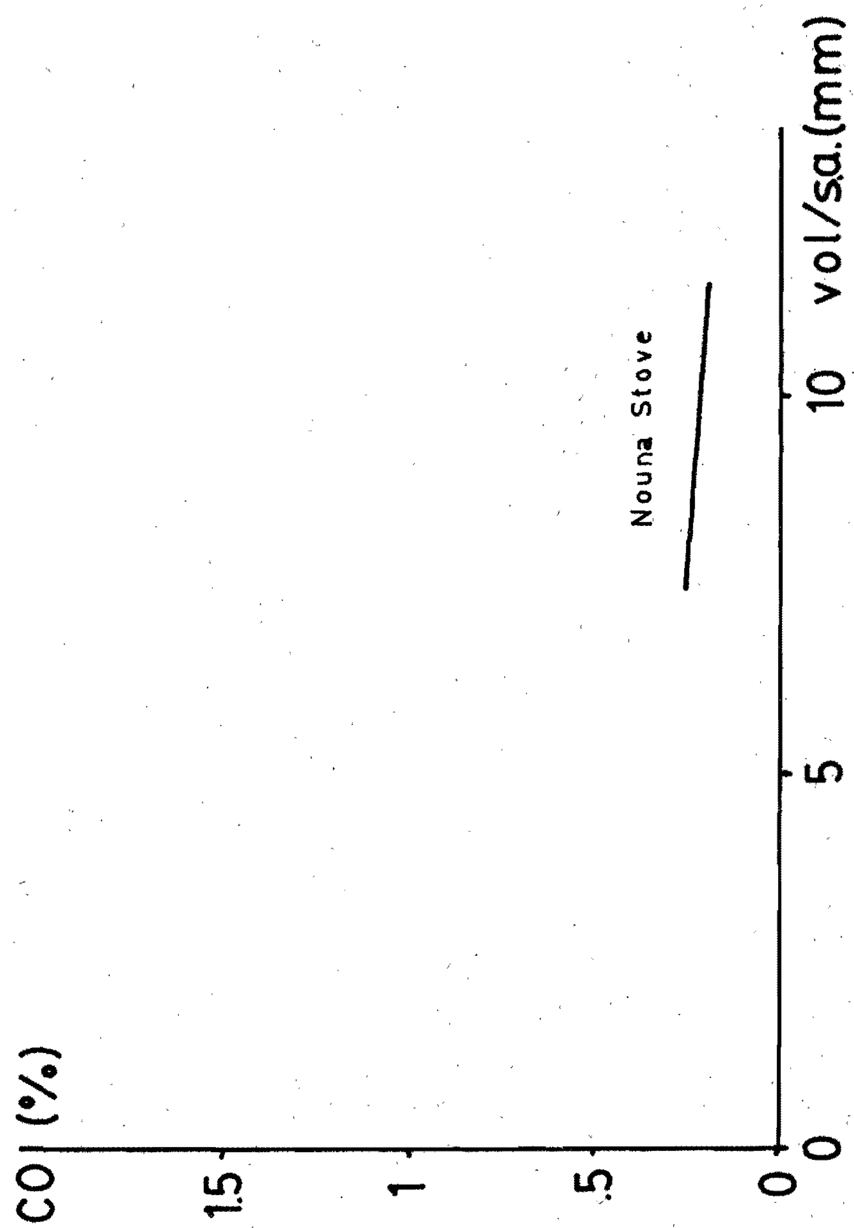

Figure 7.30 : CO concentration variation with increasing volume area ratio of wood blocks.

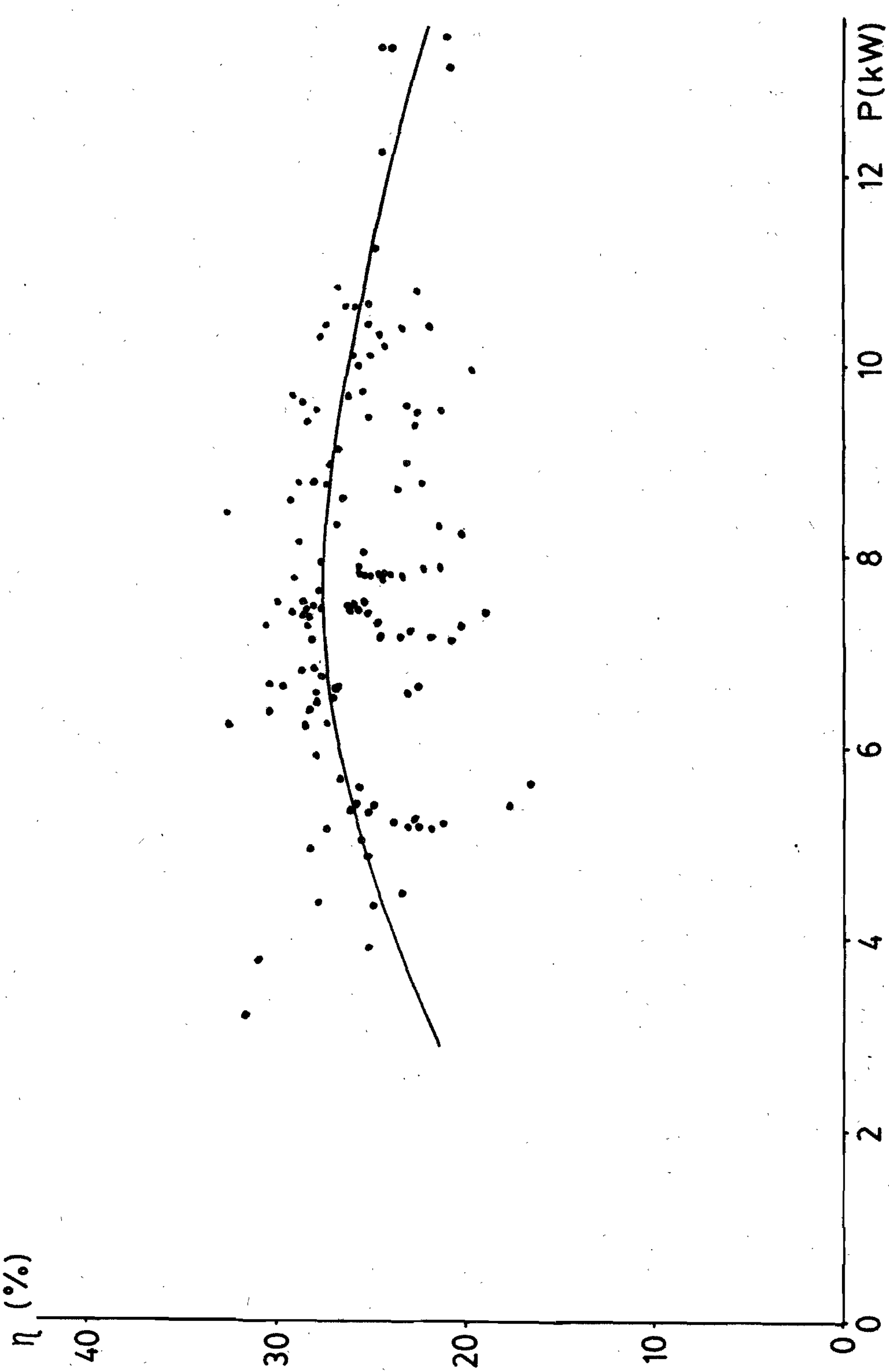

Figure 7.31 : Overview of all efficiency-power relations
by the De Lepeleire-van Daele Stove.

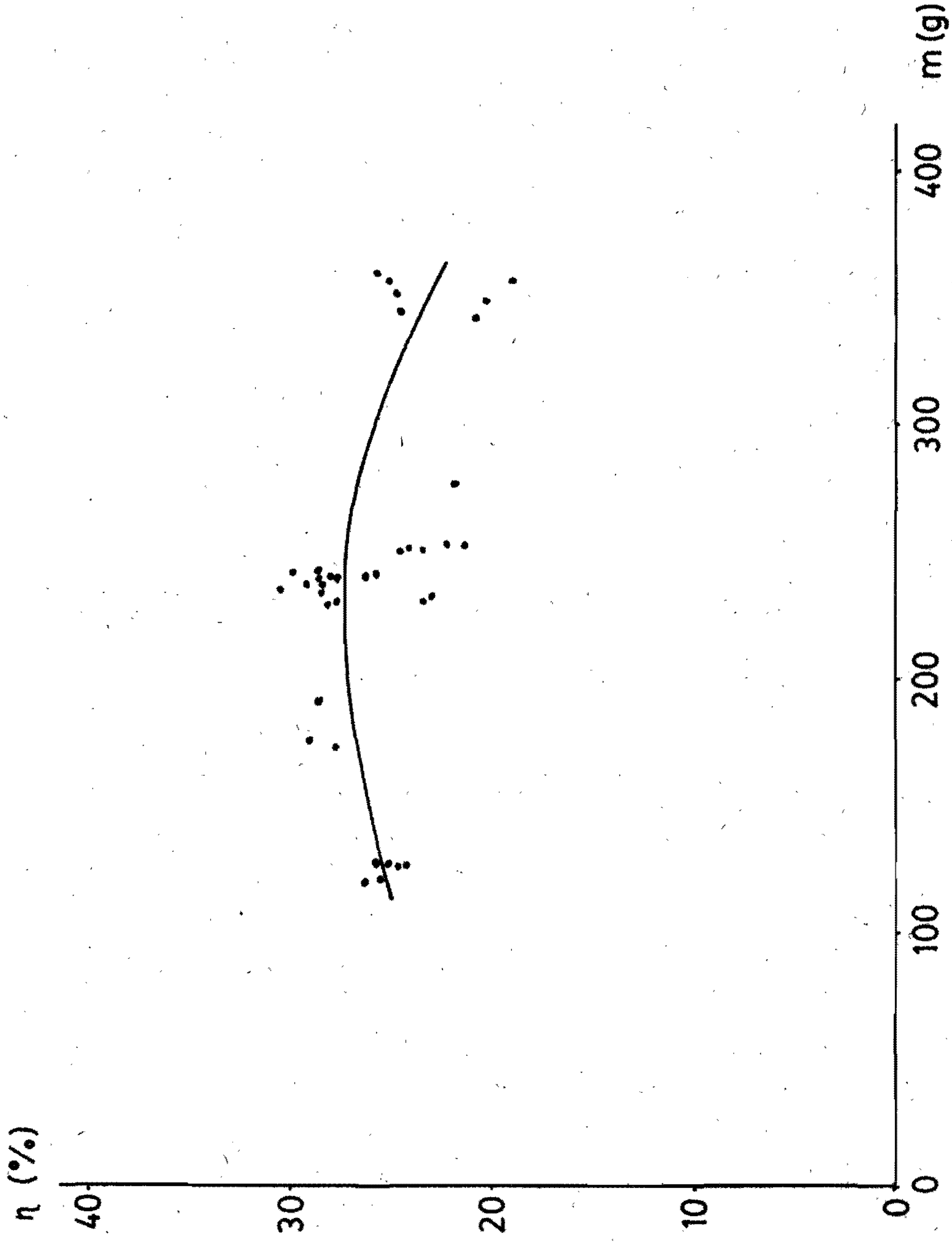

Figure 7.32 : Efficiency versus charge size for the experiments with a charging interval of 550 seconds with the de Lepeleire-van Daele Stove.

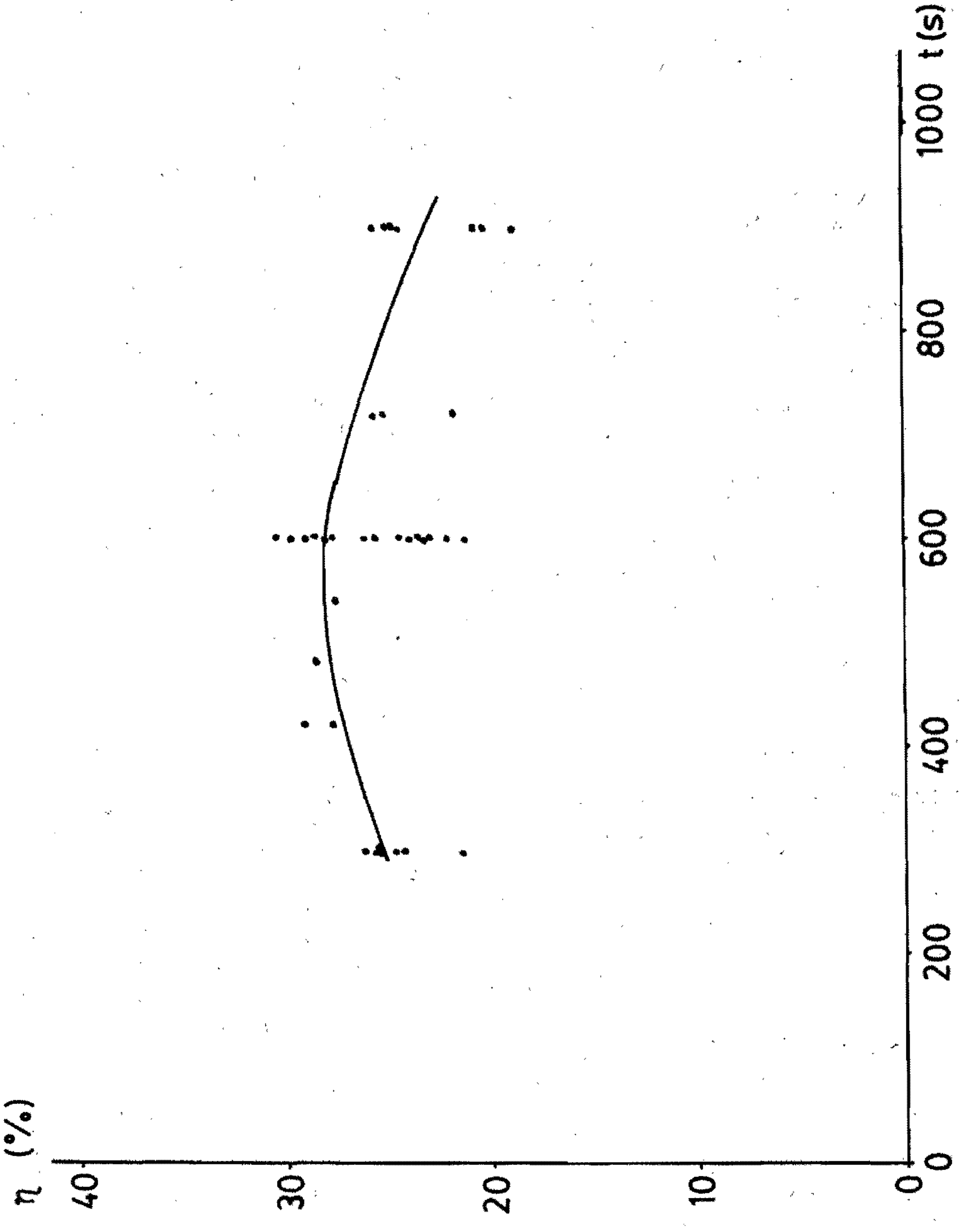

Figure 7.33 : Charge time versus efficiency for the experiments with a charge of 230 g with the De Lepeleire-van Daele Stove.

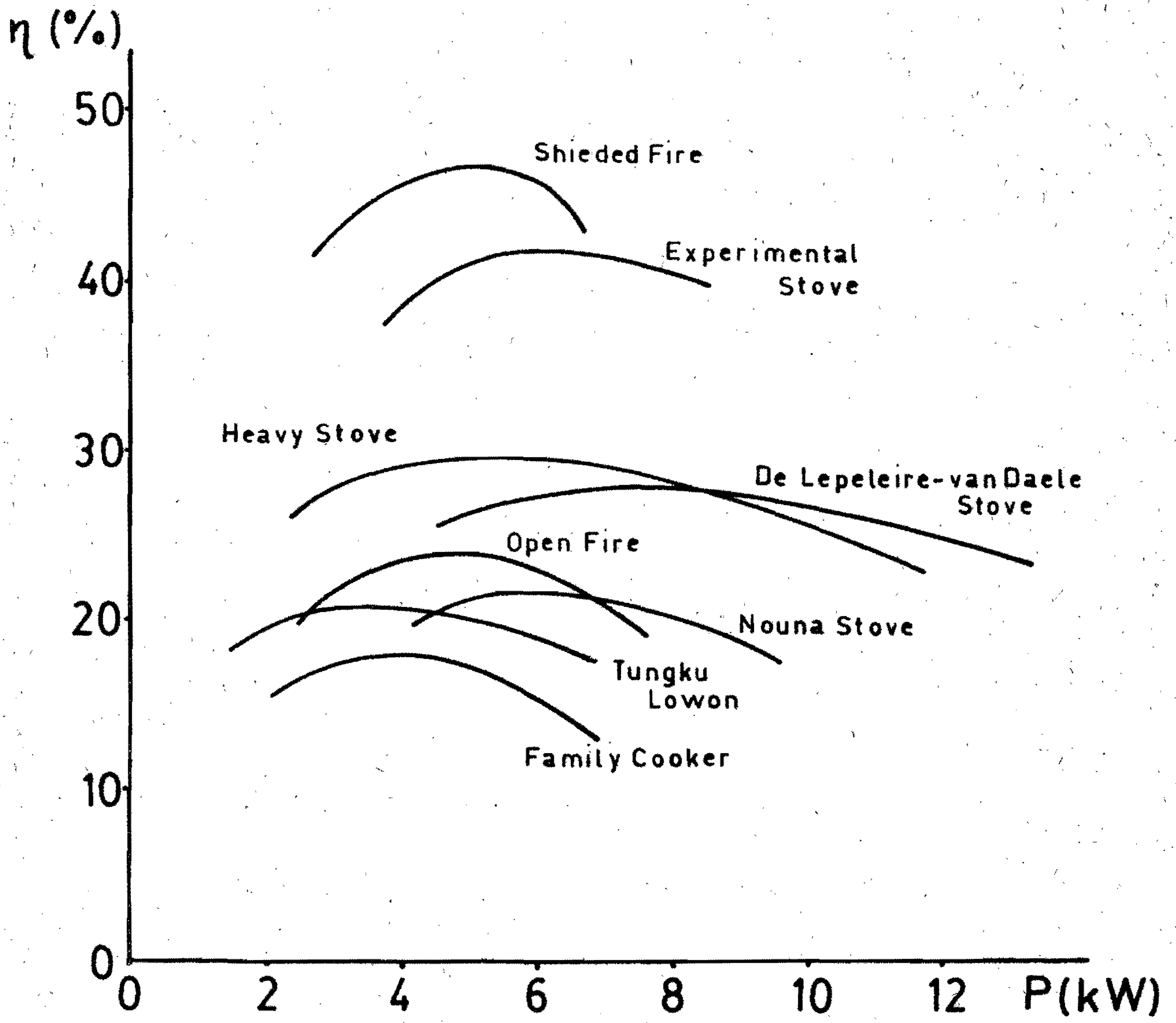

Figure 7.34 : The influence of the power range on the efficiency.

7.6.6 *Combustion quality*

The combustion quality is reflected in the composition of the flue gases, i.e. the amount of CO, C_xH_y and soot in the fluegas. These products are due to imcomplete combustion and thus their quantities contribute to the latent heat losses. Compared to the measurements of soot and C_xH_y it is much easier to measure CO concentration and that is why only for two stoves the soot and C_xH_y concentrations are measured, while CO is measured for 4 stoves. The contribution of the C_xH_y concentration to the heat balance is between .1 and 2.2 percent for the Nouna Stove and between 2.4 and 9 percent for the Tungku Lowon Stove. In order to apply these results to other stoves a relation is sought between the CO and the C_xH_y part of the heat balance and between the CO and C_xH_y concentrations in the fluegases. In figure 7.37 the C_xH_y concentration (in ppm) is plotted versus the CO concentration (in %). This is done for all measurements on the Nouna Stove and the Tungku Lowon. It is possible now to fit a curve with the least square method through all the points.
The formula we tried was:

$$C_xH_y = a\ CO^b,$$

with C_xH_y in ppm and CO in percentage points.
The values we found for the coefficients a and b are

$$a = .023$$
$$b = .438$$

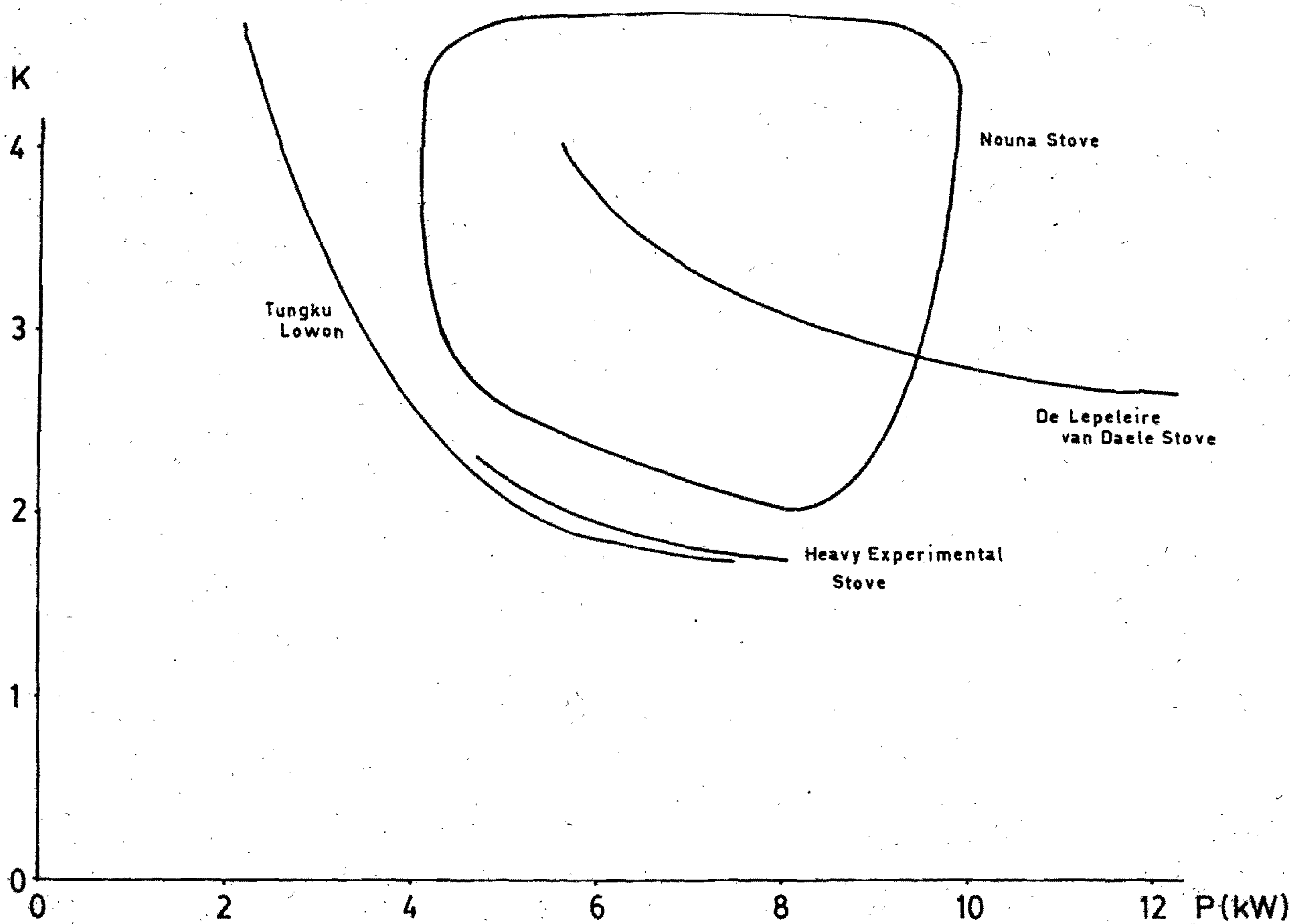

Figure 7.35 : Excess air factor as a function of the power.

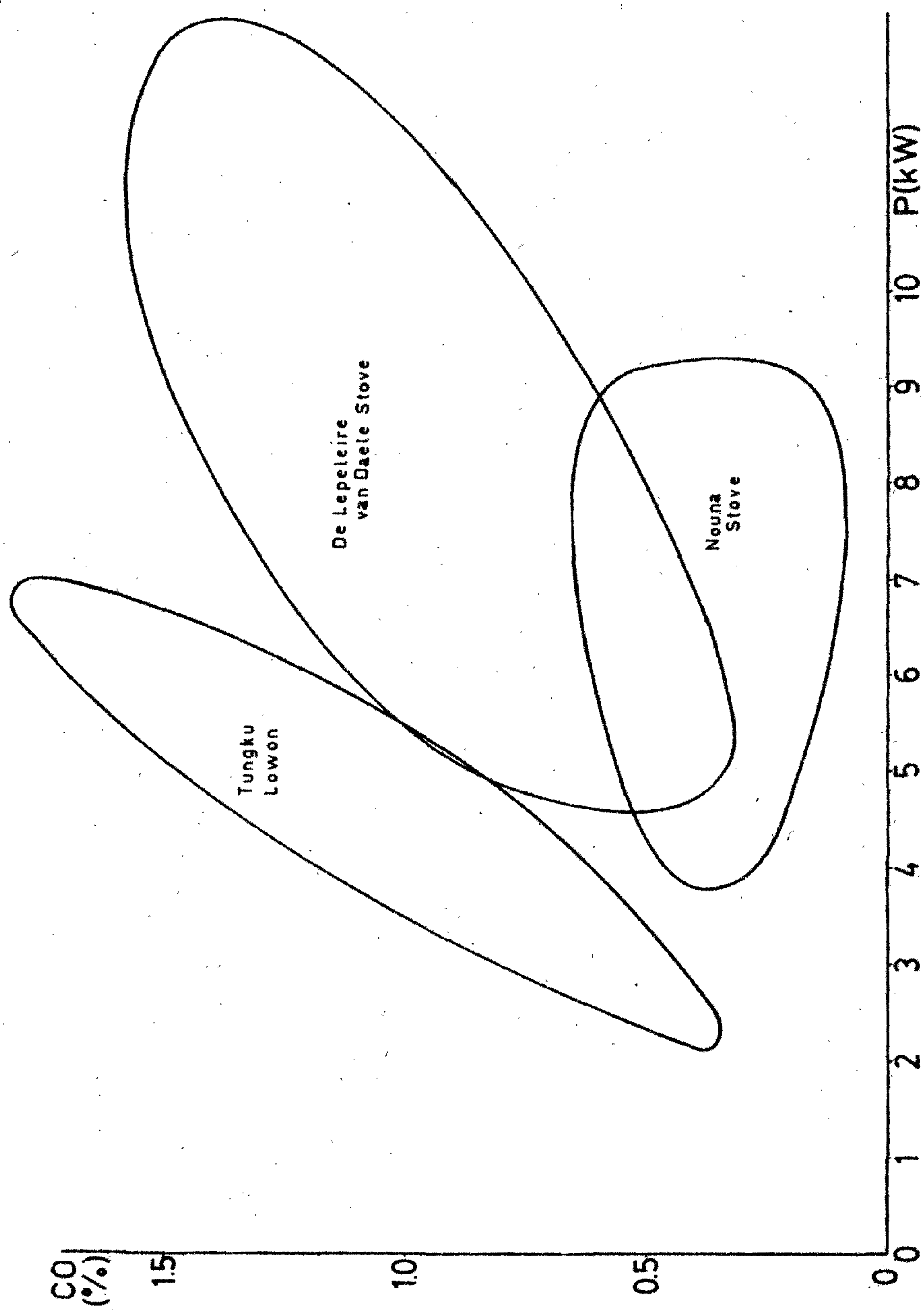

Figure 7.36 : CO concentration as a function of the power.

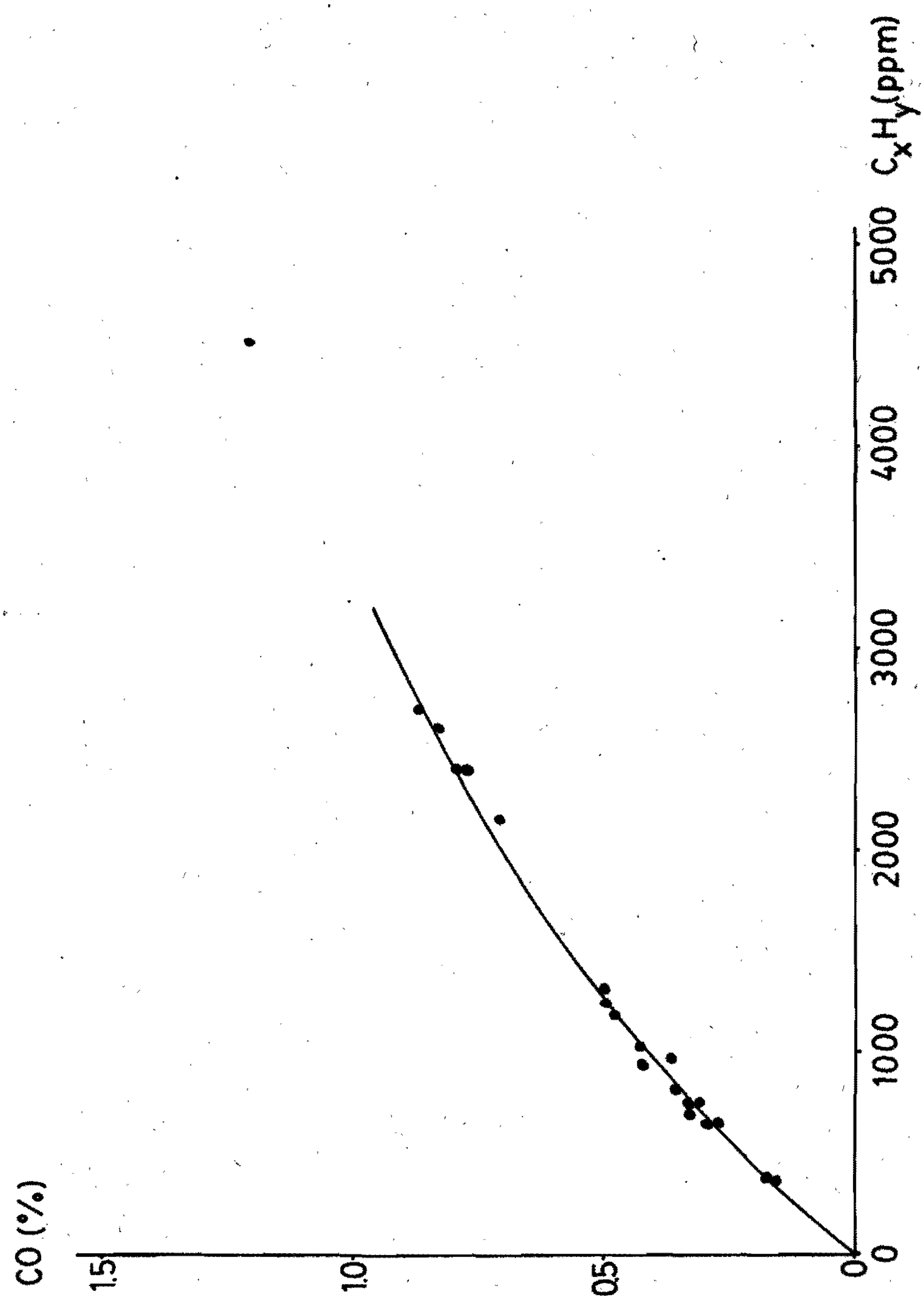

Figure 7.37 : The relation between the CO concentration and the C_xH_y concentration.

7.7 Discussion

The results collected together in this study - while forming a rather incoherent picture - do assist us in delineating a few generalized remarks on the performance of woodburning stoves. It seems that four factors could be used as performance indicators for characterizing a stove. These are:

a. maximum power level of a stove, P_{max}
b. power range of a stove, $R \equiv P_{max}/P_{min}$
c. efficiency, η
d. CO concentration, [CO]

P_{max} provides an indication of the speed with which a certain cooking task, say, bringing a mixture of water and lentils to boil, can be accomplished. R and η together determine the fuel economy, particularly for those cooking operations that require prolonged periods of simmering. [CO] is introduced as a separate performance indicator because it can be a source of serious health hazard.

We can write down a few generalized functional relationships showing the dependence of these indicators on the design and operating parameters.

$$P_{max} = P_{max}\{a_b, f_1, [CO]\}$$

Where a_b is the area of the fire place, f, contains fuel species and condition. As far as can be gathered a_b is the single parameter that determines P_{max} of a stove. As higher efficiencies becomes possible, it is obvious, that we will require smaller a_b. However it has not been possible to establish a clear relationship between P_{max} and a_b. As stated in section 7.4 it is possible to restrict P_{max} by claiming that the CO concentration does not exceed a given limit

$$R = R\ \{m_c, f_1, \lambda\}$$

where m_c is the charge weight and d is the excess air factor.

The excess air factor in its turn can be given by

$$\lambda = \lambda \{P,C\}$$

P is the power of the fire and C is the construction of the stove (air control, chimney length etc.). The excess air factor influences the combustion of the stove, the exact mechanism is not clear however. A higher excess air factor provides a lower CO concentration.
The efficiency η is dependent on many factors. A general relationship is given by

$$\eta = \eta \{P,\lambda,f_1, m_1,c\}$$

The relationship between the efficiency and the stove design is of special interest, because this relationship can be a tool to design a stove. Other parameters like power, fuel species etc. are so called "operational" variables and are only determined by the person who uses the stove. Some of the influences of the stove design on the efficiency are shown in the figures presented in the study.
Some relationships, like the distance between first pan and grate, show a general tendency for all stoves. Other relationships are only measured for one stove, like the chimney length, and the figures show only preliminary measurements. In the future more attention should be paid to determine the influences of the stove design parameters on the efficiency.

The last performance indicator is the carbon monoxyde concentration of the flue gases

$$[CO] = [CO] \{\lambda,P\}.$$

In contradiction to the earlier mentioned parameters, the maximum [CO] concentration is determined by general rules for health, and not by the stove itself. In consequence the [CO] concentration may restrict the power output of a stove (as discussed in chapter 5).

Considerable amount of data have been presented in the preceding pages. But, unfortunaly, they fall for short of what is required to give general relationships for every class of stoves. Very much more work both of a theoretical and an experimental nature are necessary before we are in a position to have the necessary relationships. Such relationships will be very useful to understand the behaviour of a stove and to predict the influence of a change in the design of a stove. Further the process of designing will become easier and more efficient.

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8. A SHORT REFLEXION ON WOODBURNING COOKING STOVE PERFORMANCES, EFFICIENCIES AND FUEL SAVING

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8.1 Introduction

Efficiency is not a physical fact: it is an engineering concept which can be useful to evaluate the performances of technical equipment. In principle the efficiency is a dimensionless ratio of an actually measured performance and a computed one, associated with a reference or theoretical model. Of course this reference must be clearly defined as well as the test conditions. Steady state conditions are preferred if possible but if necessary a well defined cycle may do.

For example: the efficiency of an engine running at a given speed with a given torque is the ratio of the mechanical power measured at the shaft to the power that would result if all of the energy content of the consumed fuel was converted into mechnical power.

Often a system is an assembly of different components, each with their own efficiency. The overall system efficiency is the the product of all the component efficiencies. The other way around: the efficiency of a machine can be split up in partial efficiencies. This allows identification and localisation of separate losses*.

For example: the efficiency of a pump may be considered as a product of:
- Hydraulic efficiency (which considers pressure losses in the fluid flow)
- mechanical efficiency (reflecting friction losses outside the fluid flow)
- volumetric efficiency (reflecting leakage losses)

* Partial efficiencies are very popular among engineers; users however limit their attention to the overall efficiency.

The efficiency concept does not work in those cases where any energy input is converted into losses and no useful energy output is produced. This happens for example with a brake. The energy efficiency of a brake would always be zero; this is obvious nonsense.

Many technical systems have to operate under varying conditions for example at high or low power, speed, etc.. Therefore the performance cannot be expressed in one single efficiency figure. Instead efficiency plots are needed to get a correct impression. For example a fan will show an efficiency plot as in figure 8.1.

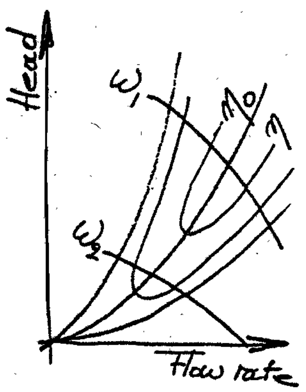


Figure 8.1 Fan efficiency plot.

Anyhow: efficiency is a man made concept which needs a clear definition. Among other things one has to define a reference to compute the theoretical performance. Many times this includes a consensus on what is considered useful and what is a loss.
As a matter of fact all energy flows in technical systems in the end turn into loss. Therefore it is important to state the point at which an ongoing process will be checked, and to define the assembly of equipment that will be considered...

8.2 Efficiency Definitions

An energy flow diagram in woodburning cooking stoves might look like figure 8.2. Charging and burning wood is a batch type process; a strictly steady state situation is therefore not feasible. One can however imagine periodic quasi steady states.

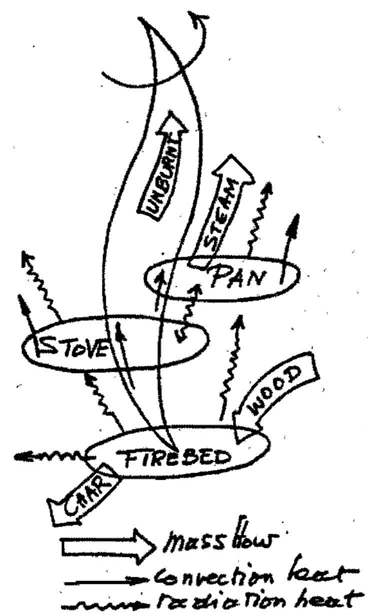


Figure 8.2 Mass and energy flows in a woodstove.

Heat is generated by (partial) combustion of wood. This heat is partly transferred by radiation from the firebed and by convection from the flue gas. It is transferred to the stove body, the pan and/or the environment.
In the end heat left in the flue gas (both sensible and latent) is lost in the environment by dilution.

When starting a cold stove things are complicated by the fact that heat is stored in the stove body in the beginning to be lost later...
In the meantime the cooking pan receives heat in many ways and loses heat to the environment. When heating up part of the received heat is stored in the pan contents (food) and a fraction is lost to the environment by radiation and convection from the pan surfaces. When simmering some latent heat is supposed to be

absorbed in physico-chemical changes in the food (see 8.5).
A major part however of the net heat input is wasted in evaporation of water which acts as a thermostat, and induces a quasi steady state.

Many different efficiencies can be suggested.
For example:

1) Combustion efficiency η_C

$$\eta_c = \frac{\text{heat generated by combustion}}{\text{combustion heat (in fuelwood)}} = \frac{\text{consumed H.-unburnt losses}}{\text{combustion heat}}$$

2) Heat transfer efficiency η_T

$$\eta_T = \frac{\text{gross heat input into the pan}}{\text{generated heat}} = \frac{\text{generated heat - stove loss}}{\text{generated heat}}$$

3) Pan efficiency η_P

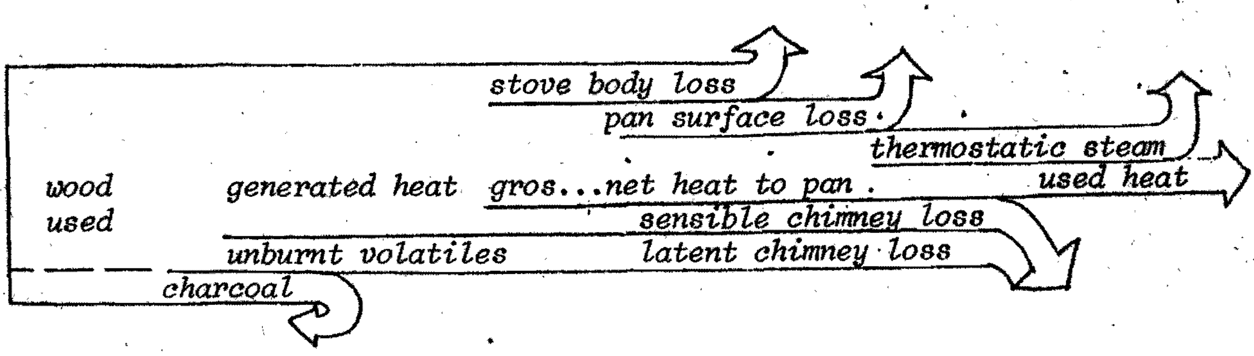
$$\eta_P = \frac{\text{net heat input into the pan}}{\text{gross heat input}} = \frac{\text{gross input - pan surface loss}}{\text{gross input}}$$

4) Control efficiency η_R

$$\eta_R = \frac{\text{heat absorbed in the food mix}}{\text{net heat input into the pan}} = \frac{\text{net input - evaporation}}{\text{net input}}$$

5) Overall cooking efficiency

$$\eta = \frac{\text{heat absorbed in the food mix}}{\text{combustion heat (in fuelwood)}} = \eta_C \cdot \eta_T \cdot \eta_P \cdot \eta_R$$



Note that this is not the only possible layout. For example one might suggest a stove efficiency including combustion and heat transfer: $\eta_S = \eta_C \cdot \eta_T$
However, this concept is not likely to be popular as it is not easily measured.
An efficiency definition which is often used is

$$\eta' = \eta_C \cdot \eta_T \cdot \eta_P$$

Some people call it an overall stove efficiency, as it is indeed a product of three partial efficiencies.

8.3 Water boiling tests and simulation of cooking

Water boiling tests have been done in the lab for example in quasi steady state conditions. The power rate of the stove can be controlled by the rate of charging and burning wood. The "useful" heat transfer to the pan is computed from the evaporation rate.
Note that this is a simulation where water simulates the food mix, and the evaporation represents the storage or absorption of heat in the same mix.

Pan surface losses as such are not measured and not considered useful. Along the suggested efficiency concepts the reported result thus includes combustion, heat transfer and pan efficiency:

$$\eta' = \eta_C \cdot \eta_T \cdot \eta_P.$$

This engineers approach - which is easy to do in a well equipped lab - results in a power-efficiency plot as shown in fig. 8.3, very familiar to engineers indeed. This plot contains a lot of information on the stove tested, for example the power limits Pmin and Pmax, the stove "flexibility" Pmax/Pmin etc.. Pmin is the lowest power the stove can deliver: at a lower firing rate the fire might die. Obviously at a stove power level Pmin the net heat input into the pan is Pmin . η'

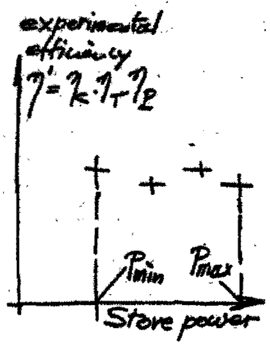


Figure 8.3 Experimental stovepower - efficiency plot.

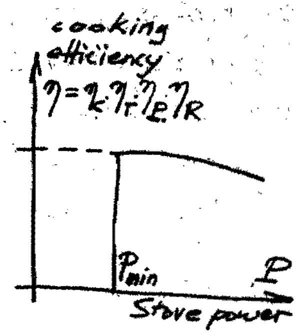


Figure 8.4 Extended stove power - cooking efficiency plot.

If the heat demand - in actual cooking - is lower than $P_{min} \cdot \eta'$ the excess heat input is lost in thermostatic steam generation (see 8.5). The control efficiency drops, proportionally with the reduced heat demand, even down to zero.

As a consequence the overall cooking efficiency of a simmering stove operating with a heat demand below $Pmin \cdot \eta'$ can be anything between zero and the measured efficiency η'. The experimental power-efficiency plot can be extended as shown in figure 8.4. A better image of what happens results when efficiencies are plotted as a function not of the stove power, but of the useful heat transfer or heat demand $P' = P \cdot \eta$ as shown in figure 8.5.

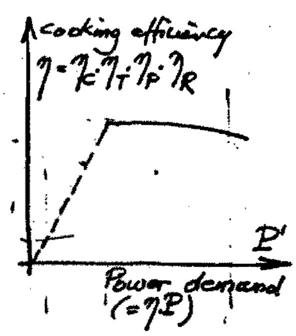


Figure 8.5 Extended heat demand efficiency plot.

It appears that at very low power demand P' the energy efficiency of the cooking system depends not on the measured efficiency but rather on the slope of the curve as in figure 8.5, that is the ratio (η'/Pmin).
It appears that to save fuel, efficiency is not the only point.

This ratio can be improved in different ways:
- by an increase in efficiency (obvious, but perhaps difficult)
- by a decrease of Pmin, through either a reduction of the nominal power Pmax or by an increased flexibility (Pmax/Pmin) or both.

Perhaps the extended efficiency plot including control efficiency explains some of the contradictions that may exist between lab results and field experience...

A funny question arises here. If simmering is frequently met in the cooking program, and if this involves net heat demands below Pmin, why do we worry about the measured efficiency? The fuel consumption is dictated by the minimum stove power, and has nothing to do with the measured efficiency η' as such. Why do we measure it? Engineers may have good reasons to do so, but the user has not.

Water boiling tests have been done in the field, often with some preferred scenario for example: bringing to the boil and then simmering for 30 or 60 minutes.
The result in this case is not a power-efficiency plot as just discussed but a single efficiency figure associated with the complete cycle. The power rate is known only as an average. Obviously it will not be possible to make an extended plot either (as in figure 8.4 and 8.5). The ratio (η'/Pmin) is unknown.

Recently for example two charcoal stoves have been tested in Nairobi, showing interesting results, as in the table below. Two liters of water were heated to the boiling point, and then simmered for thirty minutes. Water evaporation in principle was considered useful (ITDG August 1982).

stove	JIKO	UMEME
time to boil (t_B) in mins.	26	17
charcoal consumed (CC)	.150 kg	.168 kg
water evaporated (WE)	.150 kg	.564 kg
ΔT = TB − To	79°C	78°C

From these data:

energy consumed (EC)	4800 KJ	5376 KJ
sensible heat to pan (SH)	663 KJ	655 KJ
latent heat to pan (LH)	339 KJ	1275 KJ
total heat to pan (TH)	1002 KJ	1930 KJ
heat left in the pan (HL)	614 KJ	470 KJ
efficiencies PHU2 = $\frac{TH}{EC}$	.21	.36
PHU1 = $\frac{SH}{EC}$	.138	.122
cooking eff. = $\frac{HL}{EC}$	.128	.087
average power	1.43 KW	1.9 KW
simmering power excess	.19 KW	.71 KW
SSC (draft standard)	.10 kg/kg	.15 kg/kg

Note that the overall cooking efficiency of the process is not zero. However if the separate simmering period was considered the cooking efficiency would be zero indeed if at least the latent heat absorbed in the food mix is negligible. Remember that with a haybox one can simmer without any heat supply or fuel consumption.

From the above table it appears that there is no conflict at all between an efficiency approach and the specific consumption concept, if at least the overall efficiency is considered including the control efficiency.

On the other hand it appears that there is a contradicition indeed between the said overall cooking efficiency - or the SSC - and the PHU, be it PHU1 or PHU2; these figures do not reflect the fuel saving ability of a stove.

Here is a disseminators question: why should a stove user be impressed more by some partial efficiency figure than by the actual fuel consumption or the overall cooking efficiency?

If there is a conflict between "engineers" and the users viewpoint, something is wrong with the engineering. In this case, notice that the PHU concept forgets the control efficiency. The dramatic importance of this may be clear from another case. Imagine a car with poor or no power control. To go slowly one uses the brakes to compensate the excessive torque of the engine, which may have an excellent efficiency. What about the fuel economy of this type of car?

8.4 Field tests

It has been suggested to proceed almost as in lab tests, with quasi steady state operation at a controlled power level. A complete burnout of the stove to mark the end of the test would avoid possible errors in the results. These errors result from the fact that distinction of wood and charcoal is difficult, and that the consumed wood is computed as a difference of two quantities: the charged wood minus the recovered equivalent wood.
In fact this lab testing procedure departs from normal cooking practice for the sake of accuracy of the results. This is quite normal and an endless discussion can be set up or a compromise decided on. Beyond the compromise there is a question of feasibility. How to test a stove which explicitly uses long wood, and where the removal of fuel is the main way to control the fire? Notice that this is what happens in an open fire, in charcoal stoves and in most "improved" stoves now in the field.
Finally, even if a complete power-efficiency plot is feasible, there is a question about the "effort efficiency" of this approach in field operations.

For many current cooking programs the main things are the performance at maximum and minimum power level, and the flexibility of the stove. This flexibility is to be understood in two ways: the "static" flexibility (Pmax/Pmin) as mentioned above and the "dynamic" flexibility, that is the ability to switch from one power level to another ($\Delta P/\Delta time$).

A set of two boiling tests (a short and a long one) gives a lot of information, as high - and low power are involved and a transition too. This has been realised by many field testers. How to handle the results?

If any efficiency concept is to be used, it should be an overall efficiency, if at least the figure is to be relevant to users and field workers. It has been shown that the overall cooking efficiency by nature is close to zero for a simmering process even with a good stove. This reduces the concept to nonsense in this case.

It is true that the overall efficiency for a full cooking operation including heating, as simulated in a water boiling test, is not zero. Still the "spot efficiency" is zero in the simmering phase. This explains why many people are disappointed by the fact that the overall cooking efficiency (on the same stove!) drops with increasing simmering times. Whereas increased fuel consumption is readily accepted and understood.

In fact, the specific consumption as suggested in the Marseille meeting, is the simplest, shortest and most flexible way to meet all the said problems.

The consumption concept (S.S.C.) is closer to the users own experience, and closer to the specific day consumption as used in kitchen performance tests or in any energy demand research in general.

The SSC concept is more flexible too to be used afterwards, to make guesses on actual fuel consumption for cooking scenarios which are different from the W.B.T.-standard.

For example, if a stove is to be used for processes where evaporation is involved (for example to distill rhum) the short WBT which is a high power test can be used to guess the expected fuel consumption. If a stove is to be used for simmering over a period of time other than the 60's standard the expected fuelwood consumption (M) can be derived from test data as

$$M = (Mo)_{\text{long test}} - (Mo)_{\text{short test}} \text{ kg/h.}$$

Perhaps it is worthwhile to report explicitly the minimum (and maximum) power as a test result , expressed in kg/h.

8.5 Closing remarks

The cooking process by boiling in water has been investigated by the food processing industry. Extensive experiments on sterilisation - and cooking effects have shown that the "cooking time" needed to get a suitable consistency (a mechanical concept) of the food depends both on the kind of food and the processing temperature. It is generally accepted that the cooking time at a temperature T can be written as

$$\text{time } (T) = \text{time } (100) \, . \, 10^{\frac{100-T}{Z}}$$

where time (100) and Z are typical food parameters. Z ranges from about 17 to 35^{o}C. This means that the cooking time doubles for a decrease in temperature of 5 to 10^{o}C and vice versa, depending on the kind of food. No influence has been reported from evaporation rates of the boiling medium: the evaporation merely acts as a thermostat.
All this has been shown too by common experience with hay boxes and pressure cookers.

Some latent heat is absorbed in the cooking process. However detailed data on the heat quantities involved are rather rare. Probably this has to do with the fact that the measurement of these latent heats is difficult. Anyhow they must be relatively small. This appears for example from the fact that in a haybox the necessary heat is supplied by limited conversion of sensible heat...

If F kg of food are to be cooked with at least W kg of water to give (F+W)kg of cooked mix on a stove which evaporates DW kg of water in the process, one has to start with (W+DW)kg of water, or (F+W+DW)kg of mix.
The extra amount of water DW requires an extra amount of heat input DH:

$$DH = DW\ (c.\ \Delta T + L)$$

where c represents the specific heat, L the latent heat of vaporisation. This extra heat DH probably comes from extra fuel consumption. To save fuel DW should be as small as possible. However, if the extra heat DH is considered useful, it will increase some poorly defined efficiency figures.

9. THE INFLUENCE OF WOOD PROPERTIES ON THE PERFORMANCE OF THE NOUNA WOOD STOVE

by
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9.1 Experimental details

The experiments to determine the influence of wood properties on the cooking efficiency, have been carried out in the original Nouna Wood Stove in hot conditions (see Chapter 3 for details). The initial amount of water was 4 l respectively 2 l for both the pans used. The experimental set up of the measurements is similar to earlier experiments that have been carried out (Prasad 1981).

The variables studied were:
1) density of wood,
2) moisture content,
3) wood size.

The effect of various wood densities on the efficiency is studied with oven dry Merbau, Beech and Meranti. The proximate analyses of these wood species are presented in table 9.1. To test the influence of the moisture content on the efficiency white fir has been used with moisture contents of respectively 10, 20 and 30%. The experiments concerning the influence of the wood size on the efficiency have been carried out with fir with a moisture content of 10%. In relation to the standard wood size of 0.02 * 0.03 * 0.2 m, the experiments have been carried out with the following wood sizes:

0.045 * 0.054 * 0.2 m	perimeter = 2 *	L = 1*
0.025 * 0.054 * 0.2 m	" = 1.6*	L = 1*
0.025 * 0.054 * 0.4 m	" = 1.6*	L = 2*

The effect of the variables has been determined for three magnitudes of the heat output with a combustion air damper position of 25%.

9.2 The influence of wood properties on the efficiency

9.2.1 Effect of various wood species on the efficiency

As already said the wood used was oven dry wood with a size of 0.02 * 0.03 * 0.2 m. The results are summarized in table 9.2 and plotted in figure 9.1. It shows the efficiency as a function of the heat output. The efficiency varies between 14% and 21% for a heat output range between 4.5 and 13 kW. Anyhow, there is no clear indication that the investigated wood species considerably influence the efficiency of the Nouna Wood Stove. This does not imply that the wood burns away in the same manner. For example the Merbau burns with short flames, when compared with white fir.

9.2.2 Effect of the moisture content of wood on the efficiency

These experiments have been carried out with white fir and a size of 0.02 * 0.03 * 0.2 m. The moisture content of the wood varies between 0% and 30%. The experiments are summarized in table 9.3 and are plotted in figure 9.2. This graph shows the efficiency as a function of the heat output. The efficiency of the stove varies between 15 and 22% for a heat output between 2.7 and 10.5 kW. The experiments that have been carried out with a 30% moisture content of the wood, show that the minimum heat output can be reduced to a value of 2.7 kW. It is possible that a heavy wood species in combination with a high moisture content can lead to remarkable results concerning the minimum heat output of a stove. However, further experiments would be necessary to proof this contention. Anyhow, in general the conclusion can be drawn that the moisture content has no remarkable influence on the efficiency. Only the experiments with a 30% moisture content have the tendency to give higher efficiencies. However this result requires greater consideration.

9.2.3 Effect of the wood size on the efficiency

To see the influence of the wood size on the efficiency a series of experiments have been carried out with variations in perimeter and length in relation to the standard wood size of 0.02 * 0.03 * 0.2 m. With the wood sizes used the stove could still be fired with the combustion air damper in the wood entrance. The results of the experiments are summarized in table 9.4 and are plotted in figure 9.4. The figure shows that no remarkable improvement of the efficiency can be observed by firing the stove with the various wood sizes. The efficiency varies between 12 and 19% for a heat output range from 4.4 to 9.6 kW.

To see the influence of the wood properties on boiling time, a figure is made in which the boiling time of the first pan is plotted as a function of the heat output (figure 9.5). The experiments with the various kinds of wood show no remarkable influence on the boiling time. The boiling time has the tendency to rise for higher moisture contents of the wood. No conclusions can be drawn from the experiments with the different wood sizes.

9.3 Effect of various wood properties on the combustion performance

For each of the experiments the flue gas composition has been measured. Because no noteworthy facts are observed, the results of the experiments are limited to the presentation in table 9.5.

9.4 Experiments with wood from Upper Volta

In order to find out what would be the effect of firing the stove with wood from Upper Volta, an experiment was carried out in the "C" Nouna Stove with a spiece of wood, so called Detarium Microcarpum. This wood was one of the seventeen wood samples supplied by field-workers of GTZ in the framework of the collaboration with the wood-burning Stove Group.

The length and diameter of the used Upper Volta wood was 0.25 m and 0.06 m respectively. The weight of the wood was 0.353 kg with a density of 700 kg/m^3 and a moisture content of about 8%. Because only one piece of the Upper Volta wood was available, the experiment started with oven dry white fir, comparable to a heat output of about 5.2 kW. After burning out the remaning charcoal the experiment restarted with the wood from Upper Volta with a heat output of 1.6 kW. Despite the low level of this heat output, the total efficiency does not drop very much while the evaporated water of the first pan is reduced with a factor two and of the second pan even with a factor three. The results are presented in table 9.6. Two plots were made of the flue gas composition during the experiment., (see figures 9.6 and 9.7). Visual observations show, that the Upper Volta wood burns away in another manner compared with the standard wood. In contradiction to the white fir pieces, that are burned entirely, the Upper Volta wood is burning from the front of the stove to the back side. Because of this phenomenon in combination with the diameter of the wood of 0.06 m, a lot of smoke is produced during this part of the experiment. Cutting the wood in small pieces can lead to a solution of this problem and avoid the production of smoke. The low heat output of 1.6 kW can be attributed to the low burning velocity of the wood. Another problem is, that the user has to take care of keeping the fire burning. Of course more experiments with this type of wood are needed to give certainty concerning the behaviour of the wood. As already mentioned in this chapter, GTZ* supplied seventeen wood samples and four charcoal samples to the Woodburning Stove Group. For four of those samples a complete analysis has been made and for another four samples only an approximate analysis has been carried out. These values are presented in table 9.7. Table 9.8 presents the densities of the other wood samples. The samples were sent to the W.S.G. in sealed plastic packs.

* GTZ - Gesellschaft für Technische Zusammenarbeit.

9.5 Conclusions

The following conclusions can be drawn. Concerning the effect of wood properties on the performance of the original Nouna Wood Stove:

* Variations of the wood size do not influence the efficiency nor the boiling time.

* A high moisture content reduces the minimum heat output. On the other hand boiling time rises with higher moisture contents. No influence on the efficiency can be observed.

* Neither the efficiency nor the boiling time are influenced by the used of different wood species.

* Care should be excercised while using these conclusions because they are valid only for the original Nouna Wood Stove. It is possible that other stoves show another relationship between efficiency and wood properties. Hence this issue deserves further consideration.

* No conclusion can be drawn about the wood from Upper Volta because only one piece was available.

9.6 References

K. Krishna Prasad (1981)
Some studies on open fires, shielded fires and heavy stoves.
Report from the Woodburning Stove Group.

Table 9.1 Proximate analysis for various kinds of wood

Sample		Merbau	Meranti	Beech	Yelutong
Ash	[Wt %]	0.2	0.02	0.3	0.35
Volatile matter	[Wt %]	73.9	78.1	84.5	82.80
Gros calorific value	[kJ/kg]	19,550	19,775	18,525	19,375
Net calorific value	[kJ/kg]	18,050	18,250	16,975	17,775
Density	[kg/m^3]	~ 813	~ 472	~ 722	~ 355

Table 9.2 Efficiency of the Nouna wood stove as a function of the heat output of the fire for various kinds of wood.

size of the wood pieces: 0.02 x 0.03 x 0.2 m
depth of pans in stove : 0.11 m
initial amount of water: first pan 4.0 kg
second pan 2.0 kg

Symbols:

d.p. - combustion air damper position [%]
Δm_f - mass of charge [kg]
Δt - time between the two charges [min]
$\dot{Q}$ - heat output of fire [kW]
n_i - number of charges [1]
m_f - total amount of wood used [kg]
t_t - total burning time [min]
T_i - initial temperature of the water [°C]

t_b - time to boiling : t_{b1} - of pan 1 [min]
t_{b2} - of pan 2 [min]
m_s - amount of water evaporated: m_{s1} - of pan 1 [kg]
m_{s2} - of pan 2 [kg]
η - efficiency : η_1 - of pan 1 [%]
η_2 - of pan 2 [%]
η_t - total [%]
m.c. - moisture content [%]

Type and run no.	d.p. [%]	Δm_f [kg]	Δt [min]	$\dot{Q}$ [kW]	n_i [1]	m_f [kg]	t_t [min]	T_i [°C]	t_b [min]		m_s [kg]		η [%]		
									t_{b1}	t_{b2}	m_{s1}	m_{s2}	η_1	η_2	η_t
									Meranti m.c. = 0%						
1	25	0.227	8	8.65	4	0.9102		18	24	*	0.4468	0.0127	14.33	4.2	18.5
2	"	0.218	12	5.53	4	0.873	51	22	21	42	0.5585	0.0323	16.12	4.55	20.7
3	"	0.221	15	4.50	4	0.8835	65	20	33	51	0.4936	0.0480	15.2	4.82	20.0
									Beech m.c. = 0%						
4	25	0.251	8	8.9	5	1.2573	60	20	27	32	0.7268	0.1411	13.95	4.63	18.6
5	"	0.255	12	6	5	1.274	70	20	35	50	0.572	0.0737	12.15	3.86	15.9
5a	"	0.255	12	6	5	1.2732	75	21	27	52 [79°]	0.6629	0.0614	13.04	3.69	16.7
6	"	0.256	15	4.8	5	1.2808	90	23	35	50	0.8251	0.0776	14.49	3.77	18.3

* not cooked.

Table 9.2 (continued)

Type and run no.	d.p. [%]	Δm_f [kg]	Δt [min]	$\dot{Q}$ [kW]	n_i [1]	$\dot{m}_f$ [kg]	t_t [min]	T_i [°C]	t_b [min]		m_s [kg]		η [%]		
									t_{b1}	t_{b2}	m_{s1}	m_{s2}	η_1	η_2	η_t
									Merbau m.c. = 0%						
7	25	0.3479	8	13.1	4	1.3916		22	22	30	0.8034	0.028	12.4	2.85	15.2
7a	25	0.3532	8	13.3	5	1.7662	89	19	27	34	1.064	0.1745	11.8	3.35	15.1
8	"	0.3482	12	8.73	4	1.3931		19	21	42	1.2742	0.0828	16.82	3.44	20.3
9	"	0.3492	15	7	4	1.3971		22	26	56	1.1945	0.0309	15.90	2.86	18.7
9a	"	0.3599	20	5.3	5	1.7596	115	20	34.5	85	1.4166	0.0557	14.28	2.50	16.8
59	25		19	5.5	4	1.3889	86	20	45	63	0.6384	0.0191	11.081	2.84	13.9

Table 9.3 Efficiency of the Nouna wood stove as a function of the heat output of the fire for various moisture contents of white fir.

Run no.	d.p. [%]	Δm_f [kg]	Δ_t [min]	$\dot{Q}$ [kW]	n_i [1]	m_f [kg]	t_t [min]	T_i [°C]	t_b [min]		m_s [kg]		η [%]		
									t_{b1}	t_{b2}	m_{s1}	m_{s2}	η_1	η_2	η_t
"Moisture content = 0%"															
10	25	0.268	8	8.4	5	1.073		20	23	*	0.6799	0.0298	14.3	3.62	17.9
11	"	0.266	12	5.55	5	1.0661		22	36	51	0.5629	0.013	12.9	3.41	16.3
12	"	0.264	15	4.4	5	1.0582		22	36	*	0.6835	0.0176	14.4	3.12	17.5
"Moisture content 10%"															
61	25		8	8.7	4	1.0038	56	21	35.5	35.5	0.3168	0.0737	12.23	5	17.2
62	"		12	5.6	4	0.9662	57	21	33.5	43	0.3526	0.04	13.21	4.7	17.9
60	"		17	4	4	0.9703	70	21	37	63	0.522	0.0245	15.54	4.45	20.0
"Moisture content 20%"															
57	25		8	8	5	1.3687	60	20	40	40	0.461	0.1322	12	4.9	16.9
65	"		8	8	5	1.3171	60	21	33	35	0.4524	0.0663	12.2	4.2	16.4
64	"		15	4.4	4	1.079	78	18	52.5	[80°]	0.243	-	12.2	3.3	15.5
56	"		20	3.4	4	1.1194	90	20	56	[82°]	0.2472	0.0134	11.6	3.2	14.8

* not cooked.

Table 9.3 (continued) Efficiency of the Nouna wood stove as a function of the heat output of the fire for various moisture contents of white fir.

Run no.	d.p. [%]	Δm_f [kg]	Δ_t [min]	$\dot{Q}$ [kW]	n_i [l]	m_f [kg]	t_t [min]	T_i [°C]	t_b [min]		m_s [kg]		η [%]		
									t_{b1}	t_{b2}	m_{s1}	m_{s2}	η_1	η_2	η_t
"Moisture content 30%"															
66	25		15	3.86	4	1.1105	63	16	47	57	0.2614	0.0184	14.4	5.4	20
67	"		21	2.73	3	0.8255	78	19	41	[79°]	0.073	-	15	4.8	20
68	"		20	2.89	4	1.1105	82	18	53	[90°]	-	-	-	-	-
73	"		12	4.74	4	1.0928	81	19	35.5	[83°]	0.4911	-	18	4	22

Table 9.4 Efficiency of the Nouna wood stove as a function of the heat output of the fire for various wood sizes.

Run no.	d.p. [%]	Δm_f [kg]	Δt [min]	$\dot{Q}$ [kW]	n_i [l]	m_f [kg]	t_t [min]	T_i [°C]	t_b [min]		m_s [kg]		η [%]		
									t_{b1}	t_{b2}	m_{s1}	m_{s2}	η_1	η_2	η_t
						White fir	"Moisture content 10%"				0.02 * 0.03 * 0.2 m				
23	25	0.219	8	8.52	3	0.6595		25	21	*	0.1892	0	13.63	2.71	16.3
24	"	0.214	12	5.55	4	0.858		22	29	*	0.3838	0	13.53	2.76	16.3
25	"	0.215	15	4.46	5	1.0754		23	45	*	0.4682	0	11.66	2.5	14.2
											0.045 * 0.054 * 0.2 m				
26	25	0.656	38	5.4	2	1.3129	76	16	32	47	0.8513	0.1342	13.55	4.09	17.6
27	"	0.670	31	6.73	2	1.341	62	22	29	45	0.8937	0.0916	13.24	3.42	16.7
28	"	0.668	22	9.45	2	1.3368	44	22	27	33	0.8047	0.0629	12.5	3.18	15.7
											0.025 * 0.054 * 0.2 m				
29	25	0.384	15	8.0	3	1.1536	45	20	33	39	0.5714	0.0421	12.18	3.54	15.7
30	"	0.384	16	7.47	3	1.1527	48	20	28	40	0.637	0.0482	12.72	3.53	16.2
31	"	0.388	21	5.75	2	0.7761	42	19	28	*	0.2817	0	13.71	3.8	17.5
32	"	0.395	28	4.4	2	0.7892	56	18	32	*	0.4245	0	15.78	3.68	19.5
											0.045 * 0.025 * 0.4 m				
33	25	0.5543	40	4.34	1	0.5543	40	20	28	*	0.1486	0	16.14	3.22	19.4
34	"	0.7755	45	5.35	1	0.77155	45	22	29	*	0.1819	0	9.04	3.3	12.3
35	"	0.9835	47	6.60	1	0.9835	47	22	21	*	0.5497	0	13.84	3.26	17.1

* not cooked.

Table 9.5 Combustion performance of the Nouna Wood stove as a function of the heat output of the fire for various wood properties.

Symbols: d.p. - combustion air damper position [% open]
n_i - number of the wood pieces [1]
$\dot{Q}$ - heat output of the fire [kW]
T_g - flue gas temperature [°C]

initial amount of water
pan 1 : 4 kg; pan 2: 2 kg

Run no.	d.p.	n_i	$\dot{Q}$	Flue gas composition				
				CO_2	CO	CO	O_2	T
	[%]	[1]	[kW]	[%]	[%]	[g/kg]	[%]	[°C]
1	25	4	8.65	6.75	0.12	20.6	13.8	326
2	"	"	5.53	5.41	0.34	70	14.96	292.5
3	"	"	4.5	4.43	0.30	75	16.13	245
4	"	5	9	7.3	0.22	34.5	13.20	-
5	"	"	6	5.44	0.31	63.7	14.9	315
5a	"	"	-	-	-	-	-	-
6	"	"	4.8	-	-	-	-	-
7	"	4	13.1	7.04	0.18	29.4	13.5	395
7a	"	5	13.3	7.90	0.13	19.0	12.56	390
8	"	4	8.73	6.42	0.26	46	13.92	312
9	"	4	7	5.12	0.32	69.5	15.5	260
9a	"	5	5.3	-	-	-	-	-
23	25	3	8.52	6.02	0.19	36.14	13.01	373
24	"	4	5.55	4.6	0.23	56	16	246
25	"	5	4.5	3.6	0.22	68	16.8	221
26	"	2	5.4	5.11	0.21	47	16.4	271
27	"	2	6.7	5.97	0.2	38	15.5	306
28	"	2	9.45	7.6	0.21	32	-	332
29	"	3	8	5.62	0.18	36.66	14.7	322
30	"	3	7.5	-	-		-	-
31	"	2	5.75	4.76	0.23	54.45	15.8	290
32	"	2	4.4	3.93	0.22	62.62	16.74	-
33	"	1	4.34	4.6	0.25	60.89	16.9	219
34	"	1	5.35	5.13	0.25	54.89	16.4	243
35	"	1	6.60	6.23	0.28	50.81	15.3	283

Table 9.5 (continued)

Combustion performance of the Nouna Wood stove as a function of the heat output of the fire for various wood properties.

Symbols: d.p. - combustion air damper position [% open]
n_i - number of the wood pieces [1]
$\dot{Q}$ - heat output of the fire [kW]
T_g - flue gas temperature [°C]

initial amount of water
pan 1 : 4 kg; pan 2: 2 kg

Run no.	d.p.	n_i	$\dot{Q}$	Flue gas composition				
				CO_2	CO	CO	O_2	T
	[%]	[%]	[kW]	[%]	[%]	[g/kg]	[%]	[°C]
61	25	4	8.7	3.5	0.08	26.40	16.3	364
62	"	"	5.6	4.4	0.17	44	15.84	306
60	"	"	4	-	-	-	-	-
57	25	5	8	4.53	0.23	57.1	16.13	285
65	"	5	8	3.84	0.24	69.5	16.5	246
64	"	4	4.4	6.11	0.23	42.85	13.81	357
56	"	4	3.4	3.25	0.24	81.23	17.7	187
66	25	4	3.86	2.53	0.24	102	18.05	182
67	"	3	2.73	-	-	-	-	-
68	"	4	2.89	3.04	0.26	93	17.55	196
73	"	4	4.74	-	-	-	-	-

Table 9.6 Influence of original Upper Volta wood on efficiency of the Nouna "C" stove

d.p. [%]	Δm_f [kg]	Δt [min]	$\dot{Q}$ [kW]	n_i [l]	m_f [kg]	t_t [min]	T_i [°C]	t_b [min]		m_s [kg]		η [%]			Evaporated water (g/min)	
								t_{b1}	t_{b2}	m_{s1}	m_{s2}	η_1	η_2	η_t	pan 1	pan 2
25	0.252	15	5.24	3	0.7568	54	20	21	34	0.7681	0.2538	22	13.5	35.5	23.3	13
"	0.3527	-	1.62	1	0.3527	64	-	-	-	0.7113	0.273	24.3	9.3	33.6	11	4

Table 9.7 Properties of wood from Upper Volta

	Anogeissus leinocarpus	Detarium microcarpum	Eucalyptus camaldulensis	Butyrospermus parkii	Casia Siamea	Acacia Seyal	Azadirachta Indica	Tamarindus Indica
Sample No.	1	10	12	16	2	3	14	17
moisture content of original sample [Wt%]	9.95	8.1	8.7	11.1	6.2	7.1	8.8	7.2
ash [Wt%]	3.3	2	0.5	2.0	4.9	5.7	2.8	7.0
volatile matter [Wt%]	84.9	80.1	88.5	83.7	76.3	74.7	78.6	69.7
carbon [Wt%]	46.9	49.3	49. 3	49.5	-	-	-	-
hydrogen [Wt%]	6.11	6.13	6.2	6.2	-	-	-	-
oxygen [Wt%]	45.5	43.6	45.8	44.5	-	-	-	-
Gros calorific value [kJ/kg]	18,800	20,350	19,750	20,125	19,209	16,894	18,696	17,115
Net calorific value [kJ/kg]	17,375	19,000	18,325	18,725	17,659	15,394	17,196	15,565

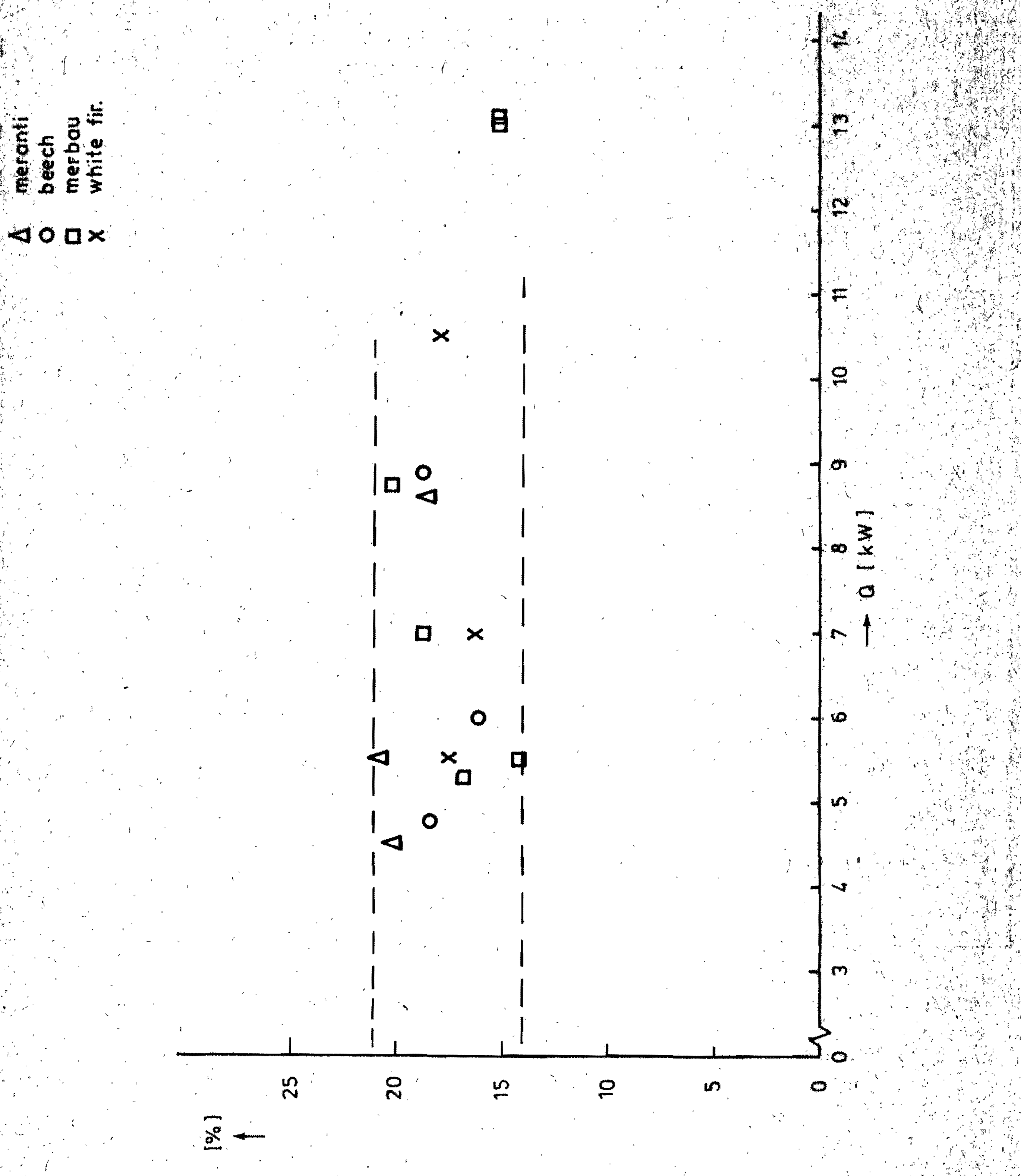
Table 9.8 **Density of Wood From Upper Volta**

	density kg/m³ 12% Mc
1°) Anogeissus leiocarpus	1020 - 1140 (910 - 1010)
2°) Cassia siamea	~ 864
3°) Acacia seyal	730 - 800
4°) " gourmaensis	-
5°) " sieberiana	650 - 720
6°) Mitragyna inermis	- (Mirubro stipulata 544)
7°) Khaya senegalensis	650 - 900 (dependent from place to place)
8°) Combretum glutinosun	~ 900
9°) Sclerocarya birrea	510 - 640
10°) Detarium microcarpum?	600 - 700 (mascocaspum)
11°) Lannea acida ?	(Lannea schimperi = 400)
12°) Eucalyptus camaldulensis	910 - 1010* 810 - 900**
13°) Gmelina arborea	460 - 500* 510 - 570**
14°) Azadirachta indica	650 - 720* 810 - 900**
15°) Balanites aegyptiaca	730 - 800 768 - 800
16°) Butyrospermum parkii	?? heavy (≅ 900)
17°) Tamarindus indica	810 - 1140 928
fruit trees	

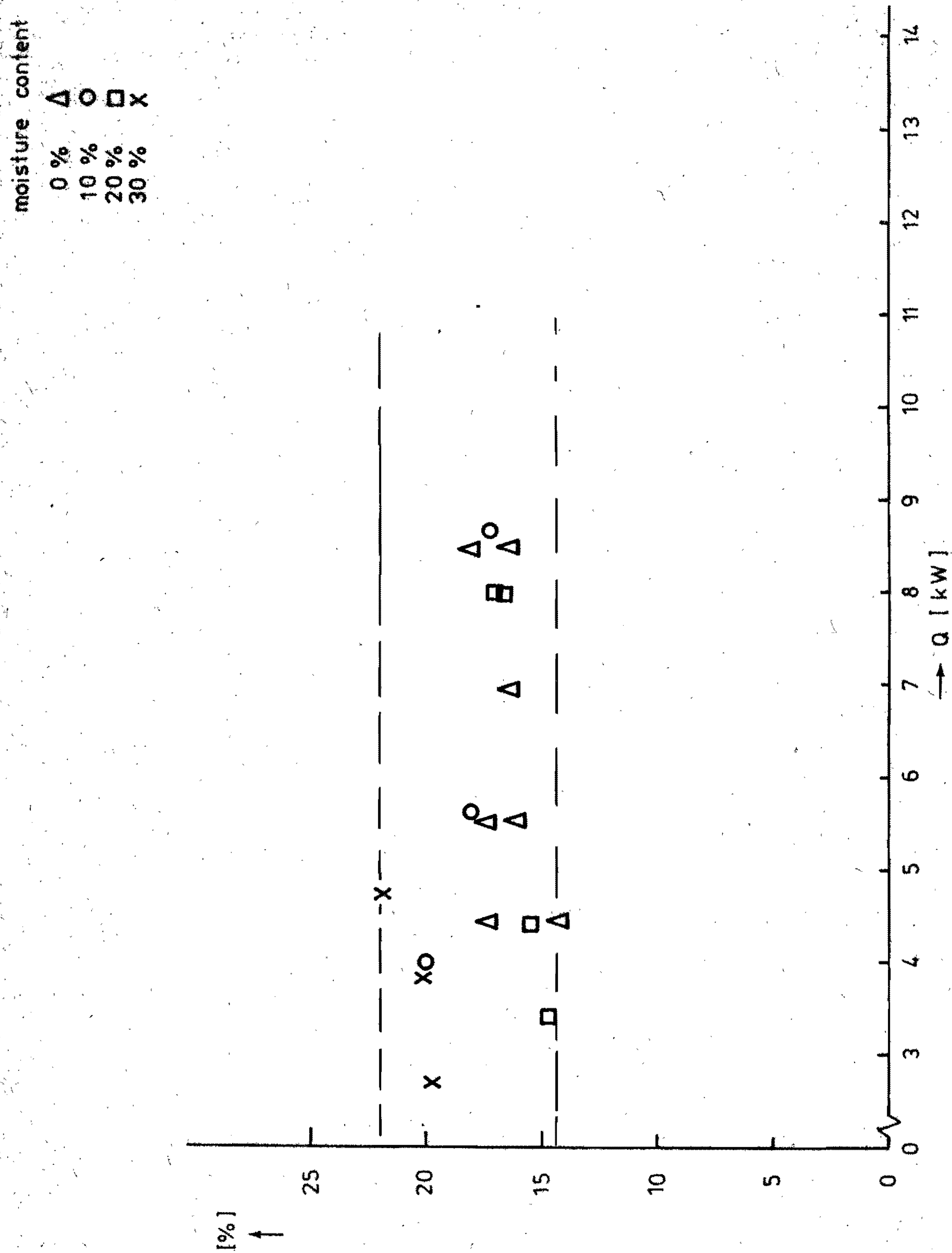
* plantation
** native

lit.: 1) Trees of Kenya
2) NEN: 1015
Handelsnaam voor houtsoorten

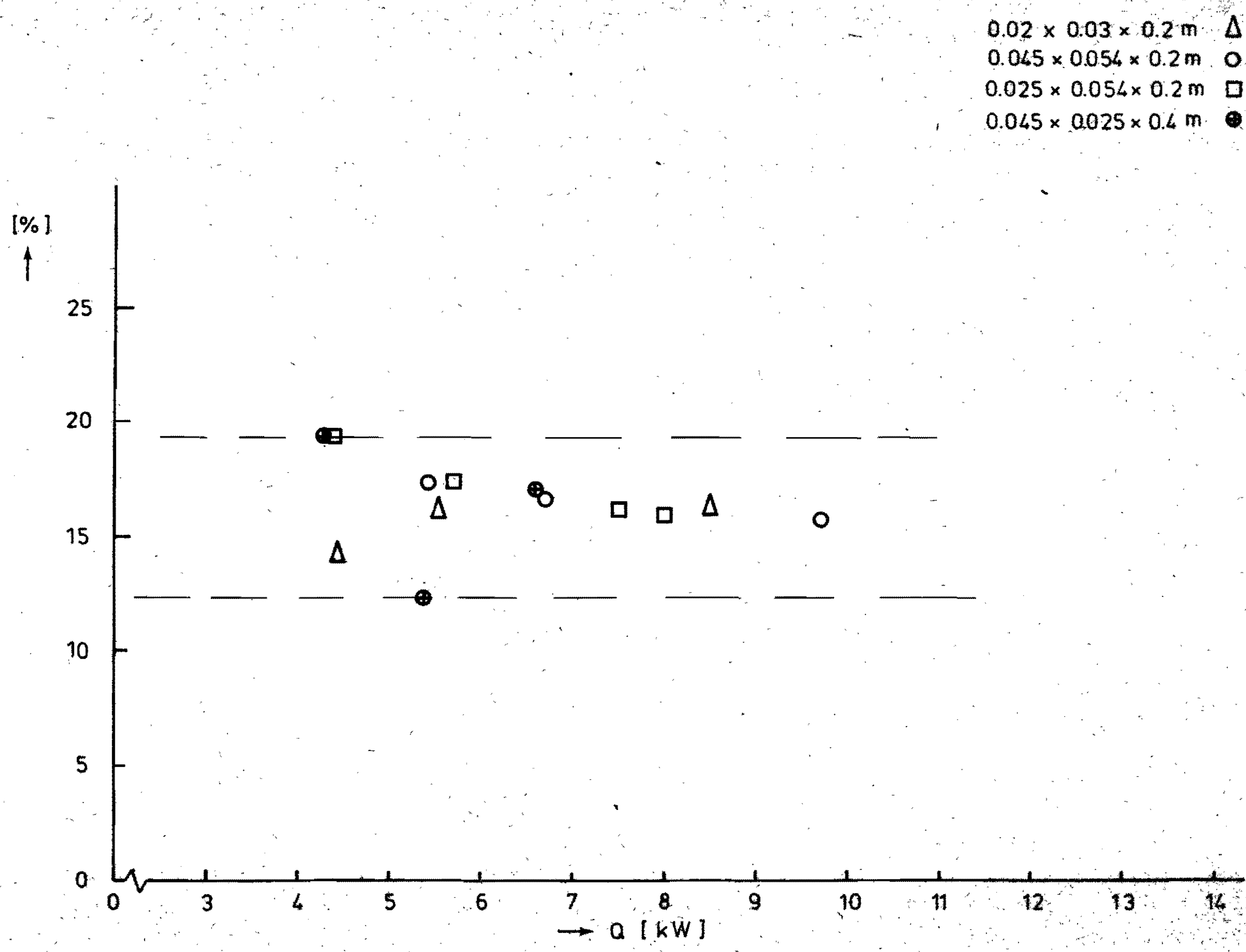
collected by: Koninklijk Instituut Voor De Tropen
Amsterdam



Efficiency as a function of the heat output of the Nouna woodstove for various kinds of wood	MT-TNO 84940 Fig. 9-1



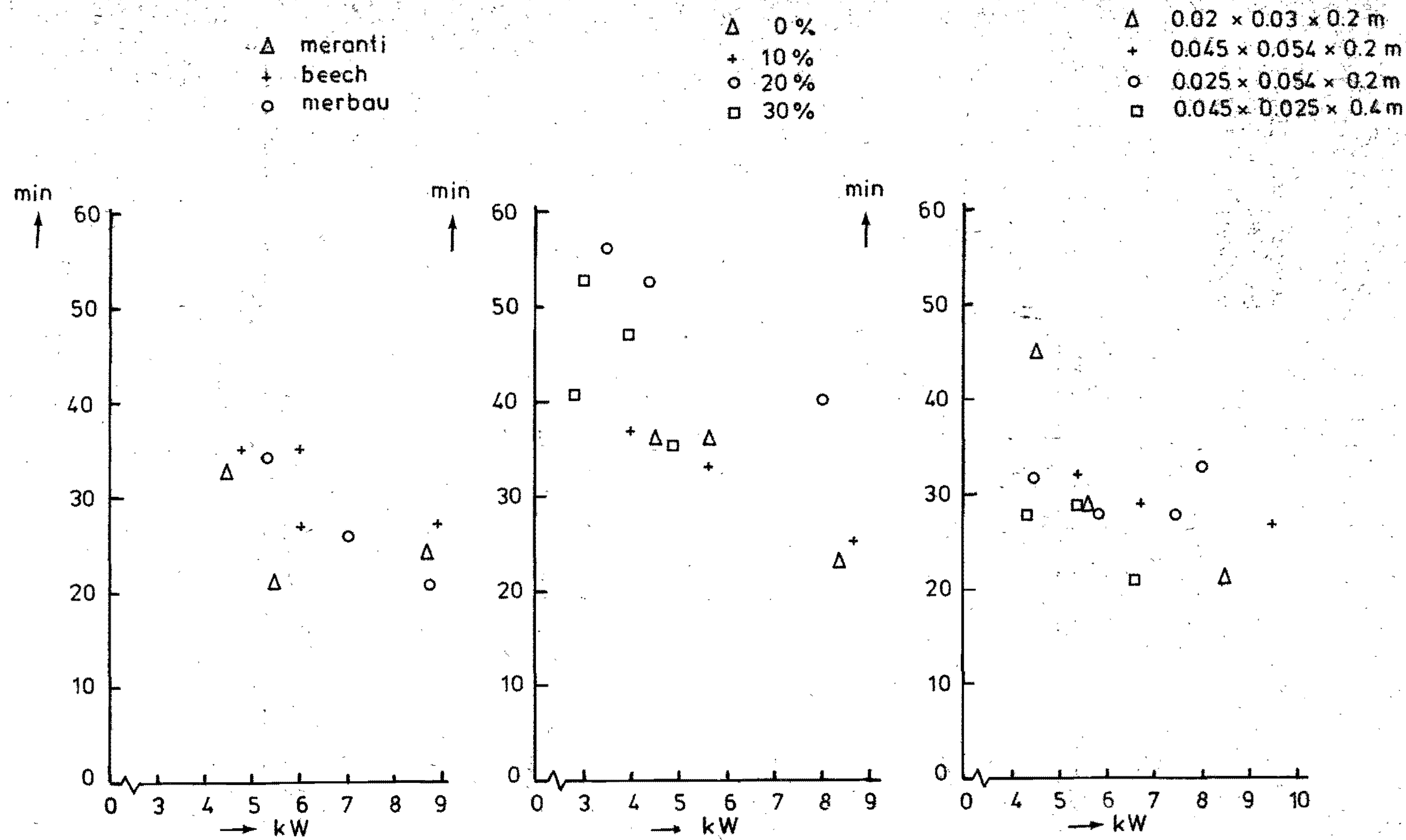
Efficiency as a function of the heat output of the Nouna woodstove for various moisture contents	MT-TNO 84940 Fig. 9.2



Efficiency as a function of the heat output of the Nouna woodstove for various wood sizes

MT-TNO
84940
Fig. 9.3

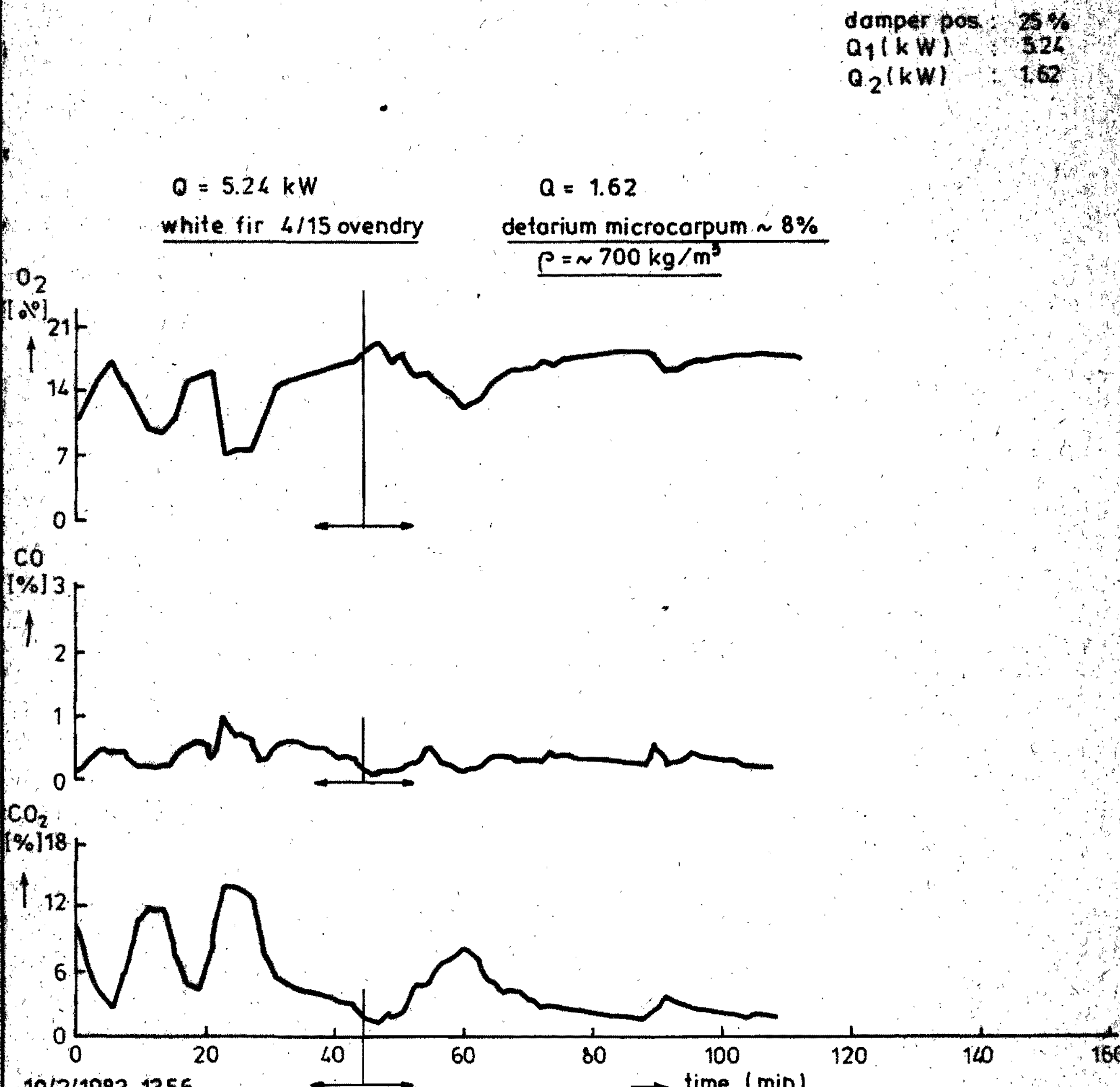
Boiling time as a function of the heat output for different wood properties



Boiling time as a function of the heat output for:

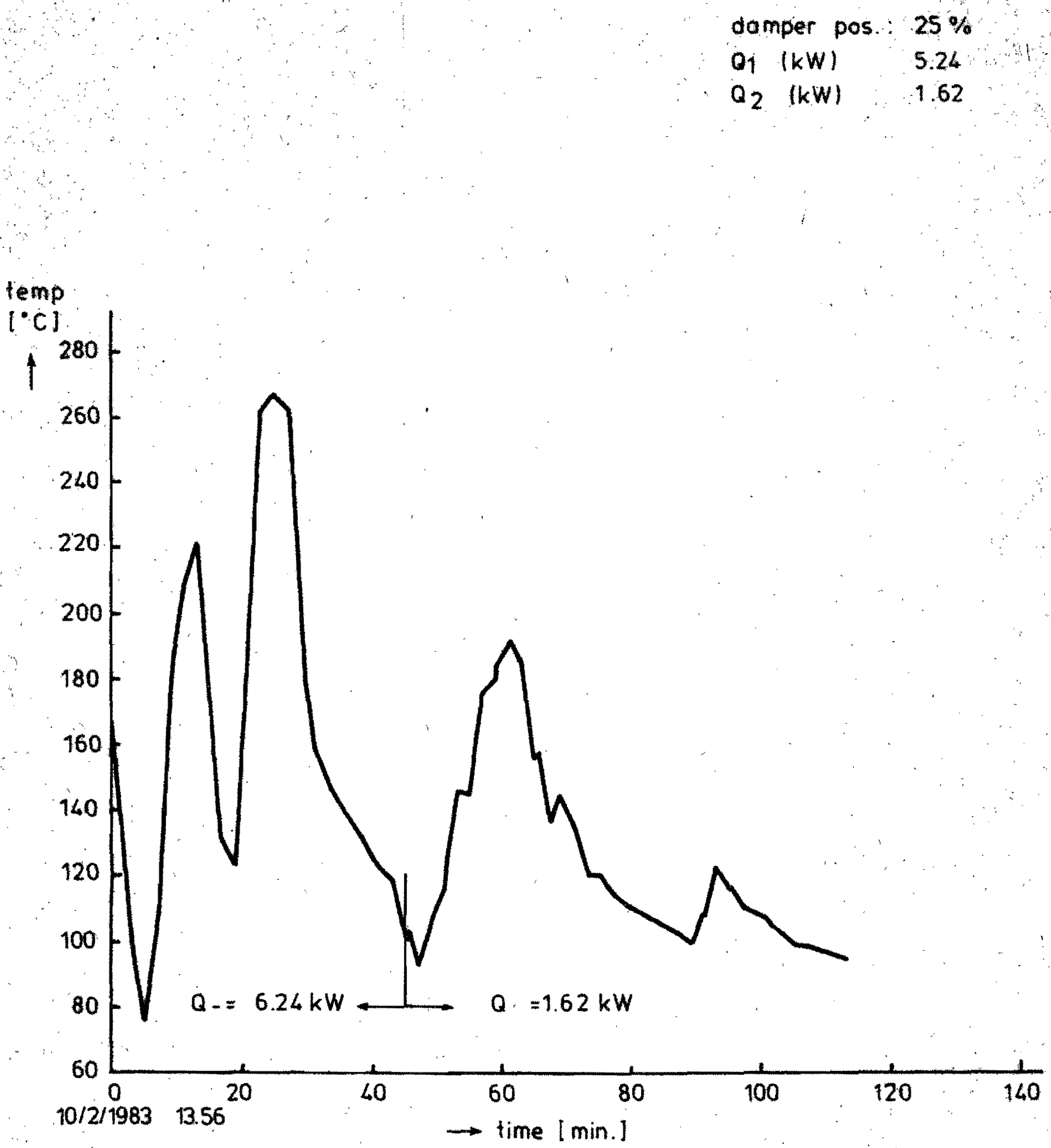
various kinds of wood — various moisture contents — various wood sizes

MT-TNO 84940 Fig. 9.4



The flue gas concentration as a function of time

MT-TNO
84940
Fig. 9·5



The flue gas temperature as a function of time

MT_TNO
84940
Fig. 9-6