

SERC

Centrum för solenergiforskning
Solar Energy Research Center

Högskolan Dalarna
SE 781 88 Borlänge
tel: +46 23 778000
www.serc.se



HÖGSKOLAN
Dalarna

Methodology for identifying parameters for the TRNSYS mod- el Type 210 -wood pellet stoves and boilers

Tomas Persson, Frank Fiedler and Svante Nordlander

Version 17.04.2012

ISSN 1401 - 7555
ISRN DU-SERC- -92- -SE
April 2006

Acknowledgements

This work has been financed by Formas, STEM, Nordic Energy Research and Högskolan Dalarna. We also would like to thank the different companies for the opportunity to measure their boilers and stoves to be able to verify the model towards real products.

Abstract

This report describes a method how to perform measurements on boilers and stoves and how to identify parameters from the measurements for the boiler/stove-model TRNSYS Type 210. The model can be used for detailed annual system simulations using TRNSYS. Experience from measurements on three different pellet stoves and four boilers were used to develop this methodology. Recommendations for the setup of measurements are given and the required combustion theory for the data evaluation and data preparation are given. The data evaluation showed that the uncertainties are quite large for the measured flue gas flow rate and for boilers and stoves with high fraction of energy going to the water jacket also the calculated heat rate to the room may have large uncertainties. A methodology for the parameter identification process and identified parameters for two different stoves and three boilers are given. Finally the identified models are compared with measured data showing that the model generally agreed well with measured data during both stationary and dynamic conditions.

Keywords: Wood pellets; boiler; stove; combustion calculations; TRNSYS; simulation model, measurements; validation.

Contents

1	INTRODUCTION	4
2	METHOD.....	4
3	THE SIMULATION MODEL.....	4
4	MEASUREMENTS	5
4.1	Set up of the measurements	5
5	COMBUSTION CALCULATIONS	7
5.1	Combustion power	7
5.2	Chemical composition for air.....	7
5.3	Chemical compositions of soft wood pellet.....	8
5.4	Combustion reactions.....	9
5.5	Methodology for combustion calculations.....	12
5.6	Calculation of air factor from measurements of CO ₂	14
5.7	Calculation of air factor from measurements of O ₂	15
5.8	Heat rate to the water circuit	16
5.9	Heat rate to the flue gasses.....	16
5.10	Density of the flue gas	18
5.11	Density of air.....	20
5.12	Heat capacity of the flue gas	21
5.13	Heat capacity of air	23
5.14	Heat rate to the ambient and full energy balance of the boiler	23
5.15	Efficiency of boilers and stoves	24
5.16	Emission calculations.....	24
5.17	Conversion to other units	26
6	PARAMETER IDENTIFICATION	27
6.1	Procedure	27
6.2	Parameters for the Thermia Bionet boiler.....	29
6.2.1	Minimal and maximal combustion power	29
6.2.2	UA value identification.....	29
6.2.3	Thermal masses.....	30
6.2.4	Air factor.....	31
6.2.5	CO factor.....	32
6.2.6	CO emission for start/stop.....	32
6.2.7	Duration of the two start phases.....	33
6.2.8	Fan operation and mass flow after stop	34
6.2.9	Leakage flow rate.....	34
6.2.10	Electricity consumption	35

6.3	Parameters for the Rebus boiler	35
6.3.1	Minimal and maximal combustion power	35
6.3.2	UA value identification	36
6.3.3	Thermal masses	37
6.3.4	Air factor	38
6.3.5	CO factor	39
6.3.6	CO emission for start/stop	39
6.3.7	Duration of the two start phases	40
6.3.8	Fan operation and mass flow after stop	40
6.3.9	Leakage flow rate	40
6.3.10	Electricity consumption	41
6.4	Characteristics of the models	41
7	VERIFICATION AND VALIDATION	43
	NOMENCLATURE	46
	REFERENCES	51
	APPENDIX 1. Identified parameters for the boilers and stoves	53

1 Introduction

This report describes a method how to perform measurements on boilers and stoves and how to identify parameters from the measurements for the boiler/stove-model Type 210 (Nordlander, 2003). The model can be used for detailed annual system simulations using TRNSYS (Klein *et al.*, 2000).

2 Method

This work is based on measurements on three pellet stoves; one traditional with only convection/radiation to the room and two water mantled (jacketed) stoves and four different boilers. One boiler with an integrated solar store of 0.63 m³ and heat exchangers for hot water preparation, one boiler with integrated DHW (hot water) preparation and a relatively large volume and two boilers without DHW preparation and a smaller water volume. Parameters were identified for three boilers and two stoves. In addition preliminary parameters were identified for the boiler with integrated hot water preparation.

From the experiences of the measurements and parameter identification process this methodology was developed. It contains detailed information about the setup of the measurements and necessary measurement sequences, the data evaluation, necessary combustion calculations and the parameter identification process and the verification and validation.

3 The simulation model

The simulation model is described by Nordlander (2003), so only a general description is given here. The model is illustrated in Fig. 3.1. Fuel and combustion air enter the stove and combust to form a hot flue gas. Energy is transferred from the gas to a first thermal mass, m_1 , representing the part of the stove that transfers heat to the ambient air. Having passed m_1 the gas has temperature T_{g1} and heat is transferred to m_2 , which represents the gas-liquid heat exchanger in the stove. Finally the gas leaves the stove at temperature T_{g2} . Heat transfer between the room and m_1 is governed by the coefficient UA_{ma} and heat transfer between m_2 and the liquid flow stream is governed by UA_{mliq} . Heat transfer between m_1 and m_2 is governed by UA_{mm} , etc.

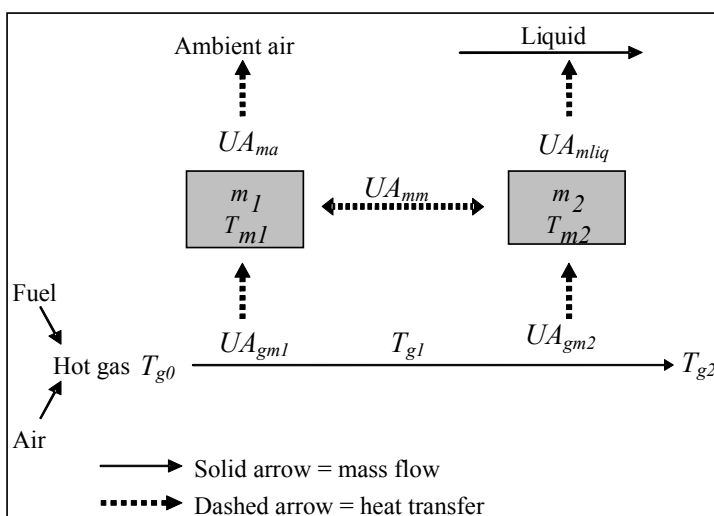


Fig. 3.1. Schematic structure of the model Type 210 (Nordlander, 2003).

4 Measurements

4.1 Set up of the measurements

Recommended set up of the measurement equipment and the sensors are illustrated in Fig. 4.1. The weight of the pellets store is continuously measured by WS1 and if the pellets store is integrated in the boiler/burner, the whole unit can be placed on the scale as in Fig. 4.1. Flexible connections of silicone for the chimney must then be used to ensure that temperature expansion don't affect the weight measurements. Also the flexible pipes and cables must be carefully connected not to influence the scale due to temperature or pressure. The value of the scale must be corrected for the density difference due to the varying temperature of the water volume in the boiler.

The gas analyser GA1 is measuring the volume components on dry gas, as a cooling unit is built in the machine to condensate the water in the flue gas. The pressure meter PS1 measure the dynamic pressure in the chimney from the Prandtl tube. Water flow rate through the boiler is measured by a flow sensor VF1 and the inlet and outlet temperatures are measured as close as possible to the boiler/stove. A complete list of the different variables that should be measured is listed in Table 4.1.

The heat loss measurements at no combustion is performed by circulating the water through the boiler and the electric heater ELAUX and measure the electricity consumption to the heater and the pump. The heat losses for the heating circuit is also without the boiler connected in the circuit.

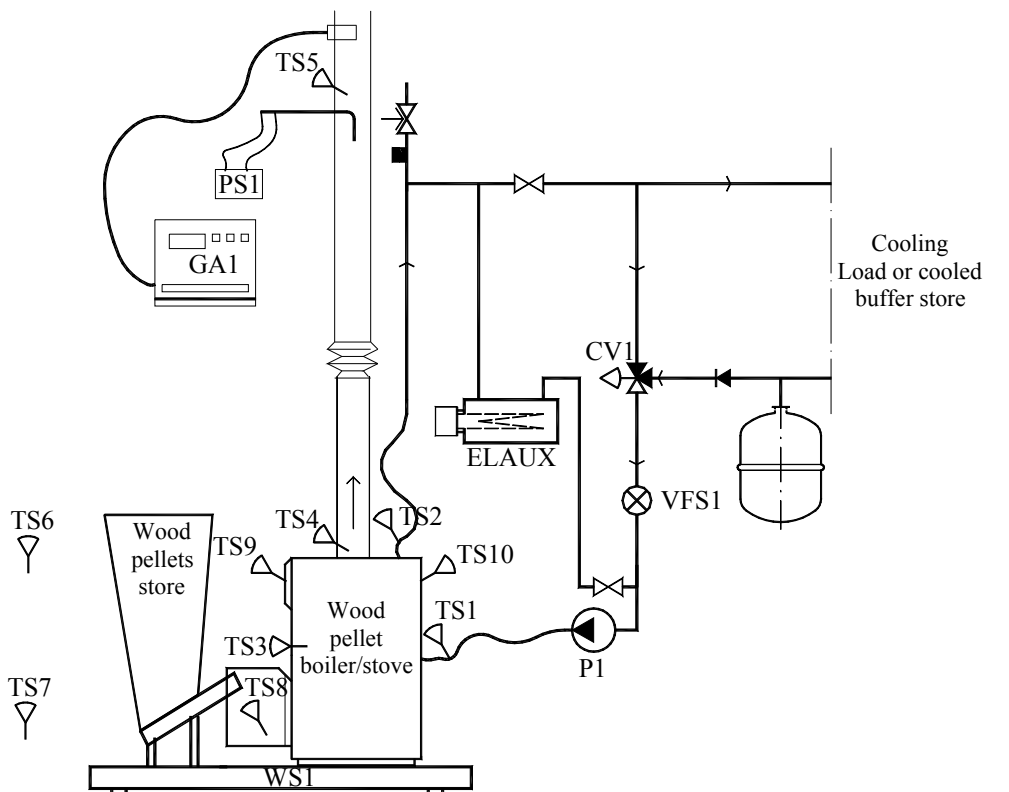


Fig. 4.1. Schematic of the test apparatus for the boilers and stoves.

Table 4.1

The different parameters that are measured during the tests of boilers and stoves.

VFS1	Flow rate through the boiler	m ³ /s
TS1	Water inlet temperature to boiler/stove	°C
TS2	Water outlet temperature from boiler/stove	°C
TS3	Boiler/stove water temperature	°C
TS4	Flue gas temperature at outlet of boiler/stove	°C
TS5	Flue gas temperature at position of flow sensor	°C
TS6, TS7	Room air temperature in different heights	°C
TS8	Temperature of combustion air	°C
TS9	Surface temperature of hot surfaces on stove/boiler	°C
TS10	Temperature of hot air outlet of stove	°C
GA1	Oxygen concentration in flue gas	%
GA1	Carbon dioxide concentration in flue gas	%
GA1	Carbon monoxide concentration in flue gas	PPM
GA1	NO _x concentration in flue gas	PPM
PS1	Dynamic pressure in chimney	Pa
WS1	Weight of pellet store or boiler/stove (integrated pellets store)	kg
-	Voltage to the screw motor or number of rotations	V, rpm
-	Voltage to the fan motor or number of rotations	V, rpm

5 Combustion calculations

The parameter identification process for Type 210 requires input data which can be taken directly from the measurement or need to be calculated from the measurement data.

The following input data are necessary:

1. The maximal combustion power of the boiler/stove
2. The heating value of the fuel
3. The boiler inlet temperature
4. The temperature of the combustion air
5. The air factor lambda
6. The air temperature in the boiler room
7. The liquid mass flow through the boiler
8. The combustion (fuel) power
9. The heat rate to the liquid
10. And the heat rate to the flue gas

The values need to be determined for steady state conditions and for several sequences over the whole combustion range of the boiler/stove.

The following chapter describe how to obtain the values 2, 5, 8, 9 and 10. Especially the heat rate to the flue gas requires extensive calculations to determine the flue gas composition and flue gas rate.

5.1 Combustion power

The combustion power is calculated from the weight measurements and the lower heating value of the pellet. To minimize the fluctuations the pellet consumption were smoothed.

$$\dot{Q}_{comb} = LHW \cdot \dot{m}_{pell} \quad \text{Equ. 5.1}$$

Where LHW is the lower heating value of the pellet and \dot{m}_{pell} is the mass flow rate of consumed pellet. The lower heating value should be defined by a laboratory for the particular pellet that is used. The pellet should be stored in a tight container or plastic bag until it is used, so that the humidity of the pellet does not change.

5.2 Chemical composition for air

Wester (2002) gives the molar composition of dry air in to Table 5.1. The values in Table 5.1 show that the number of moles of N₂, is 3.727 times the amount of O₂ and the number of moles of air is 4,773 times the amount of O₂ assuming that the molar concentration and the volume concentration is the same.

The molecular weight of the air mixture can be calculated from

$$MW_{mix} = \sum_{i=1}^n \chi_i \cdot MW_i \quad \text{Equ. 5.2}$$

and is 28.964 kg/kmol with the values of Table 5.1. The number of moles of a substance can be calculated from

$$N_i = \frac{m_i}{MW_i} \quad \text{Equ. 5.3}$$

The water content of the air is defined as kg of water per kg of dry air (Alvarez, 1990):

$$x_{H_2O,air} = \frac{m_{H_2O}}{m_{air,dry}} \quad \text{Equ. 5.4}$$

Table 5.1

Molar fraction and molecular weight of the compositions in dry air. The values are taken from Wester (2002) and are given as volume fraction, however the volume fraction and molar fraction can be considered as equal if the gasses are assumed to behave as ideal gasses. For the pressures and temperatures related to the equations here only CO₂ cannot be considered as an ideal gas (Wester, 2002), however, the mole fraction of CO₂ is so low that the influence on the final result can be neglected.

		Mole fraction in dry air χ [kmol/kmol]	Mole fraction compared to O ₂ [kmol/kmol]	MW [kg/kmol]	$\chi \cdot MW$ [kg/kmol]	Mass fraction Y [kg/kg]
Nitrogen	N ₂	0.7809	3.7274	28.013	21.875	0.7553
Oxygen	O ₂	0.2095	1.0000	31.999	6.704	0.2315
Argon	Ar	0.0093	0.0444	39.948	0.372	0.0128
Carbon dioxide	CO ₂	0.0003	0.0014	44.011	0.013	0.0005
Total		1.0000			28.964	1.0000

5.3 Chemical compositions of soft wood pellet

The wood pellet that was used during the measurements was sent to a laboratory to determine the chemical composition of the pellet and the heating value. The values in Table 5.2 are an example to describe a method for the combustion calculations used for parametric identification of wood pellet boilers and pellet stoves. The moisture content of the pellet was between 5.8 and 6.0% (6.0% for the pellet directly from the package). The moisture content was defined per kg of moisture pellet. The LHW was defined to be 17.986 MJ/kg for the pellet with 5.8% moisture content and 17.942 MJ/kg for the pellet with 6.0% moisture content.

As the moisture content of the fuel $m_{H_2O,fuel}$ is given per kg of humid wood pellet, thus the following correlation is valid.

$$m_{H_2O,fuel} = Y_{H_2O,fuel} \cdot m_{fuel,h} \quad \text{Equ. 5.5}$$

or

$$m_{H_2O,fuel} = Y_{H_2O,fuel} \cdot (m_{fuel,dry} + m_{H_2O,fuel}) \quad \text{Equ. 5.6}$$

Solving for $m_{H_2O, fuel}$ gives the following expression

$$m_{H_2O, fuel} = \frac{Y_{H_2O, fuel} \cdot m_{fuel, dry}}{1 - Y_{H_2O, fuel}} \quad \text{Equ. 5.7}$$

Table 5.2

The analysed compositions (mass fraction) of the dry wood pellet from (reference).

		Y [%]
Carbon	C	51.3
Hydrogen	H ₂	6.2
Nitrogen	N ₂	< 0.1
Oxygen	O ₂	42.2
Ashes		0.25

5.4 Combustion reactions

For a fuel containing C (Carbon) H₂ (Hydrogen), N₂ (Nitrogen), O₂ (Oxygen) and A, (ashes) the chemical reactions for the combustion (oxidation) can be written



Here we assume that no CO Carbon monoxide or NO_x is produced. Usually the concentration of NO_x and CO in the flue gas is far below 1000 ppm (1%) for wood pellet boilers (Äfab/Konsumentverket, 2005). so the influence on the energy efficiency is small.

From the chemical reactions in Equ. 5.8 and Equ. 5.9 an expression for the required amount of oxygen for stoichiometric combustion can be derived. The oxygen in the fuel is used during the combustion and the oxygen demand is therefore reduced by the oxygen content in the fuel. Stoichiometric combustion is complete combustion with minimum required amount of oxygen. The ashes are not taking part in the reactions

$$N_{O_2, stoic} = \frac{m_{C, fuel}}{MW_C} + \frac{m_{H_2, fuel}}{2 \cdot MW_{H_2}} - \frac{m_{O_2, fuel}}{MW_{O_2}} \quad \text{Equ. 5.10}$$

From Table 5.1 it can be calculated that the total amount (number of moles) of air will be 4.773 times the amount of oxygen for stoichiometric combustion.

$$N_{air, stoic} = \left(\frac{m_{C, fuel}}{MW_C} + \frac{m_{H_2, fuel}}{2 \cdot MW_{H_2}} - \frac{m_{O_2, fuel}}{MW_{O_2}} \right) \cdot 4.773 \quad \text{Equ. 5.11}$$

The number of moles of the different compositions in the combustion air can also be calculated separately in relation to the stoichiometric O₂ demand as

$$N_{CO_2,stoic,air} = 0.0014 \cdot N_{O_2,stoic} \quad \text{Equ. 5.12}$$

$$N_{N_2,stoic,air} = 3.7274 \cdot N_{O_2,stoic} \quad \text{Equ. 5.13}$$

$$N_{Ar,stoic,air} = 0.0444 \cdot N_{O_2,stoic} \quad \text{Equ. 5.14}$$

where the coefficients come from Table 5.1.

$$N_{air,stoic} = N_{O_2,stoic} + N_{CO_2,stoic,air} + N_{N_2,stoic,air} + N_{Ar,stoic,air} \quad \text{Equ. 5.15}$$

The total amount of dry flue gasses is calculated from the sum of CO₂, N₂ and Ar in the flue gases.

$$N_{flue,stoic} = N_{CO_2,stoic} + N_{N_2,stoic} + N_{Ar,stoic} \quad \text{Equ. 5.16}$$

The number of moles of N₂ in the flue gas at stoichiometric combustion $N_{N_2,stoic}$ can be assumed as:

$$N_{N_2,stoic} = N_{N_2,stoic,air} + N_{N_2,fuel} \quad \text{Equ. 5.17}$$

The molar concentration of species i in a mixture of species can be calculated by

$$\chi_i = \frac{N_i}{N_{tot}} \quad \text{Equ. 5.18}$$

and mass fractions can be calculated by

$$Y_i = \frac{m_i}{m_{tot}} = \frac{\chi_i \cdot MW_i}{MW_{mix}} \quad \text{Equ. 5.19}$$

which is a general form of Equ. 5.5.

The air factor λ is defined as the rate between real dry air consumption and the theoretical amount of dry air required to achieve stoichiometric combustion:

$$\lambda = \frac{N_{air}}{N_{air,stoic}} \quad \text{Equ. 5.20}$$

This relation can be used to calculate the amount of excess air in the combustion, which is the extra air that is not used in the combustion. The dry amount of excess air can be calculated

$$N_{air,exc} = N_{air} - N_{air,stoic} = (\lambda - 1) \cdot N_{air,stoic} \quad \text{Equ. 5.21}$$

From the knowledge of the consumed wood pellet, the amount of stoichiometric air and excess air the total mass of flue gas can be calculated

$$m_g = m_{fuel} + m_{H_2O, fuel} - m_{ashes} + m_{air, stoic} + m_{H_2O, air, stoic} + m_{air, exc} + m_{H_2O, air, exc} \quad \text{Equ. 5.22}$$

The total mass can also be calculated from the components in the flue gas as

$$m_g = N_{H_2O} \cdot MW_{H_2O} + N_{CO_2} \cdot MW_{CO_2} + N_{N_2} \cdot MW_{N_2} + N_{Ar} \cdot MW_{Ar} + N_{O_2} \cdot MW_{O_2} \quad \text{Equ. 5.23}$$

From the chemical reactions above and from the stoichiometric air consumption $N_{O_2, stoic}$ the number of moles of the carbon dioxide at stoichiometric conditions can be calculated. The non-burning components in the fuel: nitrogen and ashes is assumed to have negligible impact on the flue gases. Also the relatively low production of CO and NO_x is ignored (Äfab/Konsumentverket, 2005).

$$N_{CO_2, stoic} = \frac{m_{C, fuel}}{MW_C} + \frac{\chi_{CO_2, air}}{\chi_{O_2, air}} \cdot N_{O_2, stoic} \quad \text{Equ. 5.24}$$

Where $\chi_{CO_2, air}$ is the mole fraction of CO₂ in the dry air and $\chi_{O_2, air}$ is the mole fraction of O₂ in the dry air. The first term in Equ. 5.24 is the amount of carbon dioxide that is produced from the carbon in the fuel according to Equ. 5.8. The second term is the amount of carbon dioxide in the air (identical with Equ. 5.12) that is used for stoichiometric combustion. It relates the concentration of carbon dioxide in the air to the concentration of oxygen in the air and by multiplying this ratio with the oxygen consumption at stoichiometric conditions, the amount of carbon dioxide in the air is calculated.

The concentration of CO₂ at real combustion also includes the amount of CO₂ in the extra air, though this is almost zero (Table 5.1) and therefore almost the same as the concentration of CO₂ at stoichiometric combustion.

$$N_{CO_2} = \frac{m_{C, fuel}}{MW_C} + \frac{\chi_{CO_2, air} \cdot N_{O_2, stoic}}{\chi_{O_2, air}} + N_{air, exc} \cdot \chi_{CO_2, air} \approx N_{CO_2, stoic} \quad \text{Equ. 5.25}$$

The third term is the amount of carbon dioxide that is coming from the extra amount of air and because of that the concentration of carbon dioxide in air is nearly zero, $N_{CO_2} \approx N_{CO_2, stoic}$.

$$N_{N_2} = \frac{m_{N_2, fuel}}{MW_{N_2}} + \frac{\chi_{N_2, air} \cdot N_{O_2, stoic}}{\chi_{O_2, air}} + N_{air, exc} \cdot \chi_{N_2, air} \quad \text{Equ. 5.26}$$

$$N_{Ar} = \frac{\chi_{Ar, air} \cdot N_{O_2, stoic}}{\chi_{O_2, air}} + N_{air, exc} \cdot \chi_{Ar, air} \quad \text{Equ. 5.27}$$

$$N_{O_2} = N_{air, exc} \cdot \chi_{O_2, air} \quad \text{Equ. 5.28}$$

The water in the flue gasses comes from the oxidation of hydrogen in the fuel, from the humidity in the fuel and from the humidity in the combustion air and can be expressed.

$$\begin{aligned}
N_{H_2O} &= \frac{m_{H_2,fuel}}{MW_{H_2}} + \frac{m_{H_2O,fuel}}{MW_{H_2O}} + \frac{x_{H_2O,air} \cdot N_{air,stoic} \cdot MW_{air}}{MW_{H_2O}} + \frac{x_{H_2O,air} \cdot N_{air,exc} \cdot MW_{air}}{MW_{H_2O}} = \\
&= \frac{m_{H_2,fuel}}{MW_{H_2}} + \frac{m_{H_2O,fuel}}{MW_{H_2O}} + \frac{x_{H_2O,air} \cdot N_{air} \cdot MW_{air}}{MW_{H_2O}}
\end{aligned}
\tag{Equ. 5.29}$$

The air/fuel ratio at stoichiometric combustion conditions is a parameter required in the TRNSYS model of the boiler and it can be calculated as

$$A_f = \frac{m_{air,stoic} + m_{H_2O,air,stoic}}{m_{fuel} + m_{H_2O,fuel}}
\tag{Equ. 5.30}$$

5.5 Methodology for combustion calculations

The main reason for performing the combustion calculations is to evaluate the molar fractions and mass fractions of the different compositions in the flue gas and in that way be able to calculate the mass flow rate of the flue gas as well as the heat capacity and the density of the flue gas. It will be seen that the major parameters affecting the properties is the temperature and the air factor. The moisture content of the wood pellet and the moisture content of the air are considered to be constant.

An example of a combustion calculation performed in a spread sheet is summarized in Table 5.3. The calculation is performed for 1 kg of dry wood pellet. The mass fractions of the different compositions in the dry wood pellet from Table 5.2 is given in C3 to C7, and the molar weights of the species are given in B3 to B6. The number of moles of the different species are calculated in D3 to D6 using Equ. 5.3. The number of moles of O₂ required to combust the fuel is then calculated in E8 by Equ. 5.10, which is a sum of the values in E3 to E6. Finally the number of moles of H₂O, CO₂ and N₂ from the fuel and the oxygen is calculated in F4, G3 and H5 respectively.

The fuel moisture ratio is given in C9 and the additional moisture content for 1 kg dry fuel is calculated in C10 using Equ. 5.7. The molecule weight of water is given in B10 and the number of moles of water is calculated in D10 by Equ. 5.3.

The number of moles of CO₂, N₂ and Ar and in the air for stoichiometric combustion is calculated in G12, H12 and I12, by Equ. 5.12, Equ. 5.13 and Equ. 5.14. The total number of moles of N₂, Ar and CO₂ is then summarized in E12 and the stoichiometric dry air demand in E13 is calculated by summing the number of moles of oxygen in E8 and the number of moles of N₂, Ar and CO₂ in E12 using Equ. 5.15. The mass of the stoichiometric air demand is calculated in E14 using Equ. 5.3. The molecule weight of the air is calculated in Table 5.1.

The air moisture ratio is given in C15. The number of kg of humidity in the air is calculated in E16 by Equ. 5.4 and the number of moles of humidity is calculated in E17 by Equ. 5.3. The value of E17 is also inserted in F17. The Stoichiometric humid air demand in E18 is calculated in E18 by summing up the stoichiometric dry air demand in E13 with the Humidity in the air in E17. The total amount of CO₂, N₂ and Ar in the flue gas is summed up in G19, H19 and I19 (full expression for CO₂ is shown in Equ. 5.24). And the total amount of stoichiometric dry flue gas is calculated in E19 by using Equ. 5.16 (summing G19, H19 and I19). Finally the molar concentration of CO₂, N₂ and

Ar in the dry flue gas at stoichiometric conditions can be calculated in G20, H20 and I20 by using Equ. 5.18.

On line 21 the number of moles of humidity in the air and the humidity produced at the combustion is added, but not the humidity in the fuel.

Table 5.3

Calculation of molar concentrations and mass concentrations of the compositions in the flue gas both for dry flue gas at stoichiometric combustion and real combustion with humidity in fuel and air. Explanations of the calculation method are given in section 5.5.

	A	B	C	D	E	F	G	H	I	J	K
1	Ämne	MW	m	N	N _{O2}	Flue gas (kmol)					
2		(kg/kmol)	(kg)	(kmol)	(kmol)	H ₂ O	CO ₂	N ₂	Ar	O ₂	
3	C	12.01115	0.5130	0.04271	0.04271	0.04271					
4	H ₂	2.016	0.0620	0.03075	0.01538	0.03075					
5	N ₂	28.013	0.0005	0.00002	0.00000	0.00002					
6	O ₂	31.999	0.4220	0.01319	-0.01319						
7	Aches		0.0025								
8	Total dry=		1.0000 kg		0.04490						
9	Fuel moisture ratio=		0.059 kg/kg total								
10	Moisture	18.016	0.0627	0.00348		***					
11	Totalt fuel=		1.0627 kg								
12	N ₂ , Ar och CO ₂ i luft = 3.773 * O ₂ =				0.16942	0.00006	0.16736	0.00199			kmol
13	N _{air,stoic} = Stoichiometric dry air =				0.21432						kmol
14	Stoichiometric dry air =				6.20744						kg
15	Air moisture ratio=		0.004 kg/kg dry								
16	Humidity in air =				0.02483						kg
17	Humidity in air =				0.00138	0.00138					kmol
18	Stoichiometric humid air demand =				0.21570						kmol
19	N _{flue,stoic} = stoichiometric dry flue gas =				0.21215	0.04277	0.16738	0.00199			kmol
20	Concentration in dry gas, stoichiometric =				1.00000	0.20163	0.7890	0.0094			kmol/kmol
21	stoichiometric flue gas, dry fuel, humid air =				0.24428	0.03213	0.04277	0.16738	0.00199		kmol
22	air factor =		3.0								
23	Dry excess air =				0.42863	0.00013	0.33472	0.00399	0.08980		kmol
24	Dry excess air =				12.4149						kg
25	Humidity in excess air =				0.04966						kg
26	Humidity in excess air =				0.00276	0.00276					kmol
27	Amount of humid excess air =				0.43139						kmol
28	Total dry flue gases =				0.64078	0.04290	0.50210	0.00598	0.08980		kmol
29	Humidity from fuel =				0.00348						kmol
30	Total humid flue gases =				0.67915	0.03837	0.04290	0.50210	0.00598	0.08980	kmol
31	Amount of flue gas =		19.7570 kg			Molar concentration χ [kmol/kmol]					
32	Molar concentration in dry gas = χ =				1.0000	0.00000	0.06695	0.78357	0.00933	0.14014	kmol/kmol
33	Molar concentration in humid gas = χ =				1.0000	0.05650	0.06317	0.73931	0.00880	0.13222	kmol/kmol
34						Mass concentration Y [kg/kg]					
35	MW =					18.016	44.011	28.013	39.948	31.999	kg/kmol
36	χMW =				29.091	1.01782	2.78027	20.7102	0.35172	4.23099	kg/kmol
37	Mass concentration in humid gas = Y =				1.0000	0.03499	0.09557	0.71191	0.01209	0.14544	kg/kg
38	Air/fuel ratio =		5.8646 kg humid air per kg humid fuel								

In C22 the air factor is given and the amount of dry excess air is calculated in E23 using Equ. 5.21. The amount of the different components in this air is calculated in G23 to J23 using Equ. 5.18. The mole fractions of the different components in air are given in Table 5.1. The number of kg of excess air is calculated in E24 using Equ. 5.3. The number of kg of humidity in the excess air is calculated in E25 using Equ. 5.4 and the number of moles of humidity in the excess air is calculated in E26 using Equ. 5.3. The value of E26 is also inserted in F26. The amount of humid excess air is calculated in E27 by summing up the dry excess air in E23 with the Humidity in the excess air in E26.

The total amount of the compositions in the flue gas except from the water is summed up in line 28 (Equ. 5.26, Equ. 5.27 and Equ. 5.28 give give the whole calculation procedure. The number of moles of humidity in the fuel calculated in D10 is now inserted in F29 and the total amount of humid flue gasses are summed up in line 30. The water in the flue gas now comes from the combustion reactions, from the humidity in the fuel, from the humidity in the stoichiometric air demand and from the humidity in the excess air and can directly be calculated using Equ. 5.29.

In line 32 the molar concentrations of the components in the dry flue gas are calculated using Equ. 5.18. In line 33 the molar concentrations of the components in the humid flue gas are calculated using Equ. 5.18.

In line 35 the molecule weight of the different components in the flue gas are given and the mass fractions are multiplied with the molecular weights in line 36. The molecular weight of the mixture is calculated in E36 using Equ. 5.2. The mass fractions of the components are then calculated in line 37 using Equ. 5.19.

The theoretical air/fuel ratio at stoichiometric combustion is a parameter in the TRNSYS model and calculated in line 37 using Equ. 5.30.

Finally also the total amount of flue gas per kg fuel can be calculated in C31 using Equ. 5.23 or Equ. 5.22 by summing the values in C11, E14, E16, E24 and E25 and by subtracting the amount of ashes in C7.

5.6 Calculation of air factor from measurements of CO₂

Most often the emission measurements are on the concentration for dry gasses, thus there is a dehumidifier in the instrument. The following derivation is for measurements of molar concentration of CO₂ on dry gasses. The number of moles of CO₂ in the dry flue gas can be calculated from

$$N_{CO_2} = \chi_{CO_2} \cdot N_{flue} \quad \text{Equ. 5.31}$$

Where χ_{CO_2} is the molar concentration of CO₂ in the dry flue gas and N_{flue} is the total number of moles of dry flue gas. For the stoichiometric case the number of moles of CO₂ in the flue gas is the same as for real combustion (assuming that the concentration of CO₂ in air = 0%)

$$N_{CO_2,stoic} = N_{CO_2} = \chi_{CO_2,stoic} \cdot N_{flue,stoic} \quad \text{Equ. 5.32}$$

The total amount of dry flue gas is the sum of the amount of dry flue gas at stoichiometric combustion and the amount of the dry extra air ($\lambda-1$) that is used and it can be expressed

$$N_{flue} = N_{flue,stoic} + (\lambda - 1) \cdot N_{air,stoic} \quad \text{Equ. 5.33}$$

Where $N_{flue,stoic}$ is the number of moles of dry flue gas per kg of dry fuel at stoichiometric combustion. Combining Equ. 5.31, Equ. 5.32 and Equ. 5.33 to express the number of moles of CO_2 gives the following expression

$$N_{CO_2} = \chi_{CO_2,stoic} \cdot N_{flue,stoic} = \chi_{CO_2} \cdot (N_{flue,stoic} + (\lambda - 1) \cdot N_{air,stoic}) \quad \text{Equ. 5.34}$$

Solving for λ gives

$$\lambda = 1 + \frac{N_{flue,stoic}}{N_{air,stoic}} \left(\frac{\chi_{CO_2,stoic}}{\chi_{CO_2}} - 1 \right) \quad \text{Equ. 5.35}$$

If $N_{flue,stoic} \approx N_{air,stoic}$, the following simplification is possible

$$\lambda \approx \left(\frac{\chi_{CO_2,stoic}}{\chi_{CO_2}} \right) \quad \text{Equ. 5.36}$$

The value of $\chi_{CO_2,stoic}$ can be calculated from the knowledge of the compositions of the different substances in the wood pellet and in the air and have been calculated to 0.20163 in Table 5.3. The value of $N_{flue,stoic}$ is 0.21215 and $N_{air,stoic}$ is 0.21432 according to Table 5.3.

5.7 Calculation of air factor from measurements of O_2

The air factor can also be calculated from the measured concentration of O_2 in the exhaust gasses. The following is a derivation for measurements on dry gasses. Assuming that there is 20.95% O_2 in the air the concentration of O_2 in the dry flue gas can be expressed as

$$\chi_{O_2} = \frac{N_{O_2}}{N_{flue}} = \frac{0.2095 \cdot (\lambda - 1) \cdot N_{air,stoic}}{N_{flue,stoic} + (\lambda - 1) \cdot N_{air,stoic}} \quad \text{Equ. 5.37}$$

Solving for λ gives

$$\lambda = 1 + \frac{N_{flue,stoic}}{N_{air,stoic}} \cdot \frac{\chi_{O_2}}{0.2095 - \chi_{O_2}} \quad \text{Equ. 5.38}$$

From Table 5.3 it can be seen that $N_{flue,stoic} = 0.21215 \approx N_{air,stoic} = 0.21432$ making the following simplification possible

$$\lambda \approx 1 + \frac{\chi_{O_2}}{0.2095 - \chi_{O_2}} = \frac{0.2095}{0.2095 - \chi_{O_2}} \quad \text{Equ. 5.39}$$

The average of the values calculated by Equ. 5.35 and Equ. 5.38 was used during the parameter identification process.

5.8 Heat rate to the water circuit

The heat rate to the water circuit \dot{Q}_{wat} can be calculated as

$$\dot{Q}_{wat} = \dot{V}_{wat} \cdot \rho_{wat} \cdot (c_{p,wat,out} \cdot (t_{out} + 273.15) - c_{p,wat,in} \cdot (t_{in} + 273.15)) \quad \text{Equ. 5.40}$$

Where \dot{V}_{wat} is the measured volume flow rate of water, T_{bout} and T_{bin} is the outlet and inlet water temperature of the boiler respectively and ρ_{wat} is the density of the water and $c_{p,wat}$ is the heat capacity of the water evaluated at the average water temperature. Polynomial functions have been derived from values from Incropera and DeWitt (1996) for a temperature range from 0°C to 100°C in Fig. 5.1

$$\rho_{wat} = 0.00001732238 \cdot t^3 - 0.006092642 \cdot t^2 + 0.01617533 \cdot t + 1000.201 \quad (0^\circ\text{C} < t < 100^\circ\text{C}) \quad \text{Equ. 5.41}$$

where t is the temperature of the water passing through the flow sensor. The hat capacity can be calculated from:

$$c_{p,wat} = 0.000003275970 \cdot t^4 - 0.0008014144 \cdot t^3 + 0.07725127 \cdot t^2 - 2.96427132 \cdot t + 4215.984 \quad (0^\circ\text{C} < t < 100^\circ\text{C}) \quad \text{Equ. 5.42}$$

where t is the inlet or outlet temperature

The functions above give the agreement with data from Incropera and De Witt according to Fig. 5.1

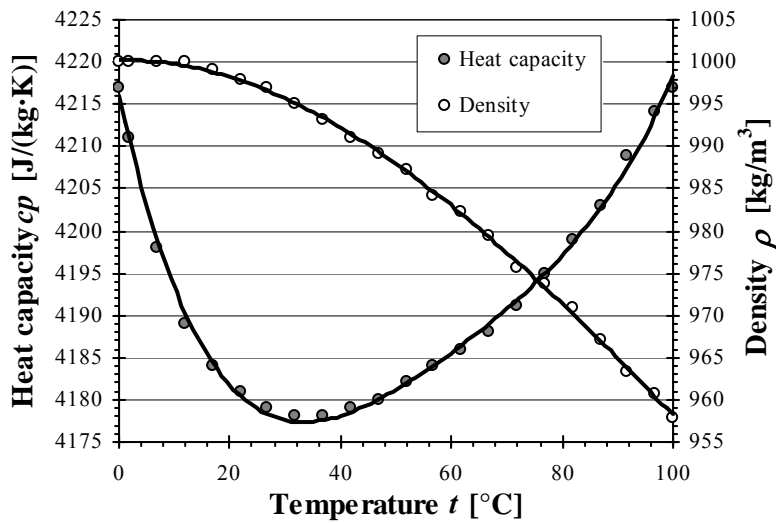


Fig. 5.1. Values of density and heat capacity of water from Incropera and DeWitt (1996) displayed as dots in comparison with the polynomial functions Equ. 5.41 and Equ. 5.42 displayed as lines.

5.9 Heat rate to the flue gasses

According to Wester (2002) the flue gas losses can be calculated by (assuming no condensation)

$$\dot{Q}_{flu} = \dot{m}_g \cdot (c_{pg,out} \cdot T_{gout} - c_{pg,in} \cdot T_{gin}) \quad \text{Equ. 5.43}$$

Where T_{gin} is the incoming combustion air temperature \approx ambient air temperature T_{∞} . The heat capacity should be evaluated at the average temperature of T_{gout} and T_{gin} or better the average heat capacity over the entire temperature range. The mass flow in the exhaust gasses \dot{m}_{flu} and thereby the energy to the flue gases can be calculated from

$$\dot{m}_g = \rho_g \cdot A \cdot \bar{v} \quad \text{Equ. 5.44}$$

Where \bar{v} is the average gas velocity and A is the chimney area. The average gas velocity can be calculated from the dynamic pressure p_d measured in the middle of the chimney by the Prandtl tube using the definition of dynamic pressure p_d and a correction factor for the velocity profile in the chimney which was identified to $v_{corr} \approx 0.85$ by measurements in different points.

$$\bar{v} = v_{corr} \cdot \sqrt{\frac{2 \cdot P_d}{\rho_g}} \quad \text{Equ. 5.45}$$

The upper two equations lead to the following equation for the mass flow rate

$$\dot{m}_g = A \cdot v_{corr} \cdot \sqrt{2 \cdot \rho_g \cdot P_d} \quad \text{Equ. 5.46}$$

Using the combustion calculations there is also a second way to calculate \dot{m}_g

$$\dot{m}_g = \dot{m}_{fuel,h} \cdot \frac{\dot{m}_g}{\dot{m}_{fuel,h}} \quad \text{Equ. 5.47}$$

where \dot{m}_g can be calculated from Equ. 5.22. The ratio \dot{m}_g/\dot{m}_{fuel} depends mainly on the air factor λ and is plotted in Fig. 5.2 for the actual type of pellet in this work. A regression gives the following polynomial function

$$\frac{\dot{m}_g}{\dot{m}_{fuel,h}} \approx -3.55271 \cdot 10^{-15} \cdot \lambda^2 + 5.86457 \cdot \lambda + 0.997647 \quad (1 < \lambda < 10) \quad \text{Equ. 5.48}$$

If the consumed pellet is measured in that way that the ashes is not leaving the weight stack what is measured is $\dot{m}_{fuel,h} - \dot{m}_{ashes}$ and the ratio in Equ 5.47 should be $\dot{m}_g/(\dot{m}_{fuel,h} - \dot{m}_{ashes})$. The polynomial function becomes

$$\frac{\dot{m}_g}{\dot{m}_{fuel,h} - \dot{m}_{ash}} \approx -5.32907 \cdot 10^{-15} \cdot \lambda^2 + 5.87840 \cdot \lambda + 1.00000 \quad (1 < \lambda < 10) \quad \text{Equ. 5.49}$$

There is also Siegerts equation that can be used for calculating the flue gas losses (Wester, 2002):

$$\dot{Q}_{flu} = \dot{Q}_{comb} \cdot \left(k \cdot \frac{T_{gout} - T_{\infty}}{\chi_{CO_2}} \right) \quad \text{Equ. 5.50}$$

where the coefficient k depends on the fuel type, fuel moisture content and the carbon dioxide concentration. Extrapolating the fuel moisture rate to $\approx 6\%$ from Wester (2002) and takes into account that χ_{CO_2} are given as molar fraction or volume fraction and not % give:

$$k = 0.000085 \cdot \chi_{CO_2} + 0.000068$$

Equ. 5.51

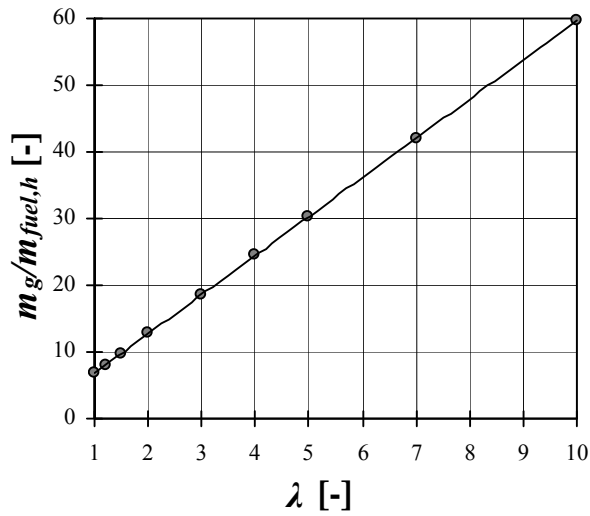


Fig. 5.2. The ratio m_g/m_{fuel} as a function of λ . The dots show the results calculated by Equ. 5.22 and the line show the regression with Equ. 5.48.

The combustion power is calculated from the mass flow of combusted pellet and the LHW as

$$\dot{Q}_{comb} = \dot{m}_{fuel,h} \cdot LHW \quad \text{Equ. 5.52}$$

If the consumed pellet is measured in that way that the ash is not leaving the weight stack the measured part will be $\dot{m}_{fuel,h} - \dot{m}_{ash}$ and the combustion power can be calculated from

$$\dot{Q}_{comb} = \left[(\dot{m}_{fuel,h} - \dot{m}_{ash}) + \dot{m}_{ash} \right] \cdot LHW \quad \text{Equ. 5.53}$$

however \dot{m}_{ash} is unknown, but can be calculated

$$\dot{m}_{ash} = \frac{\dot{m}_{ash}}{\dot{m}_{fuel,h}} \cdot \dot{m}_{fuel,h} = \frac{\dot{m}_{ash}}{\dot{m}_{fuel,h} - \dot{m}_{ash}} \cdot (\dot{m}_{fuel,h} - \dot{m}_{ash}) \quad \text{Equ. 5.54}$$

thus the combustion power is

$$\dot{Q}_{comb} = \left[(\dot{m}_{fuel,h} - \dot{m}_{ash}) + \frac{\dot{m}_{ash}}{\dot{m}_{fuel,h} - \dot{m}_{ash}} \cdot (\dot{m}_{fuel,h} - \dot{m}_{ash}) \right] \cdot LHW \quad \text{Equ. 5.55}$$

The mass of the ashes, \dot{m}_{ash} can be obtained from Table 5.3 to 0.0025 kg and the mass of the fuel, $\dot{m}_{fuel,h}$ to 1.0627 kg.

5.10 Density of the flue gas

The density of the flue gasses (ρ_g) can be estimated from the general gas law if the flue gasses are assumed to behave as ideal gases.

$$\rho_i = \frac{P \cdot MW_i}{R_u \cdot T} \quad \text{Equ. 5.56}$$

Where the general gas constant R_u is 8314.51 J/(kmol·K). Nitrogen (N_2) and oxygen (O_2) can be considered as ideal gases, however vapor (H_2O) and carbon dioxide (CO_2) are not ideal gases and could therefore cause problems in the accuracy when using the general gas law (Wester, 1991). The following correlation is therefore suggested

$$\rho_i = \frac{P \cdot MW_i}{z_i \cdot R_u \cdot T} \quad \text{Equ. 5.57}$$

Where z is the compressibility factor, which is dependent on the type of gas, the temperature and the pressure (Alvares, 1990). For the temperatures and pressures that is relevant here the compressibility factor can be assumed to be constant and for water vapour $z = 0.945$ and for carbon dioxide $z = 0.993$ (Vester, 2002). For the other gases actual in this work the compressibility factor z is close to unity. The average density of the gas mixture can then be calculated from the molar concentration

$$\rho_g = \sum_{i=1}^n \chi_i \cdot \rho_i \quad \text{Equ. 5.58}$$

The full expression of the density for the flue gases is

$$\rho_g = \frac{P}{R_u \cdot T_g} \cdot \left(\frac{\chi_{H_2O,wf} \cdot MW_{H_2O}}{z_{H_2O}} + \frac{\chi_{CO_2,wf} \cdot MW_{CO_2}}{z_{CO_2}} + \frac{\chi_{N_2,wf} \cdot MW_{N_2}}{z_{N_2}} + \frac{\chi_{Ar,wf} \cdot MW_{Ar}}{z_{Ar}} + \frac{\chi_{O_2,wf} \cdot MW_{O_2}}{z_{O_2}} \right) \quad \text{Equ. 5.59}$$

The density will vary depending on the air factor, the moisture content in the air, etc, however the variations are relatively small. With the composition of the wood pellet that is used here (Table 5.2 and moisture rate of 0.059 kg/kg), an assumed air factor of 2.0, an air moisture ratio of 0.004 kg/kg and, a pressure of 101325 Pa, the equation can be reduced to:

$$\rho_g \approx \frac{K}{T_g} \quad \text{Equ. 5.60}$$

Where $K \approx 357.7$ for an air factor of 2.0. This is a value calculated for this particular pellet fuel specified in Table 5.2, moisture content specified in Table 5.3 and for the assumed moisture content in the air. Plotting the value of K depending on air factor and making a polynomial approximation for K give the results as in Fig. 5.3. For the calculation of density for dry flue gases K_{dry} can be calculated in a similar way, just ignoring the water content

$$\rho_{g,dry} \approx \frac{K_{dry}}{T_g} \quad \text{Equ. 5.61}$$

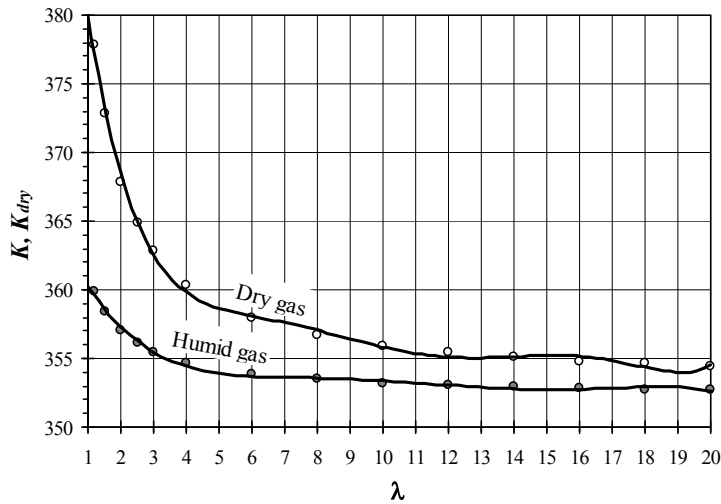


Fig. 5.3. The value of K in as a function of the air factor. The dots are calculated using Equ. 5.59 and Equ. 5.60 and the curves are calculated with the polynoms in Equ. 5.62 and Equ. 5.63.

Making a linear regression of the influence for the air factor λ between 1.2 and 20 (Fig. 5.3) give the following correlation for K and K_{dry} . The moisture content of the air is assumed to be constant 0.004 kg/kg. The moisture content of the wood pellet is assumed to be 0.059 kg/kg.

$$K \approx -0.0000714083 \cdot \lambda^5 + 0.00430408 \cdot \lambda^4 - 0.0983040 \cdot \lambda^3 + 1.06073 \cdot \lambda^2 - 5.50594 \cdot \lambda + 364.771 \quad (1 < \lambda < 20) \quad \text{Equ. 5.62}$$

$$K_{dry} \approx 0.0000338899 \cdot \lambda^6 - 0.00237913 \cdot \lambda^5 + 0.0661973 \cdot \lambda^4 - 0.929601 \cdot \lambda^3 + 6.94904 \cdot \lambda^2 - 26.8175 \cdot \lambda + 400.74 \quad (1 < \lambda < 20) \quad \text{Equ. 5.63}$$

5.11 Density of air

The density of dry air at atmospheric pressure making a regression from data in Incropera and DeWitt (1996) for temperatures between 300 and 500 K give the following expression.

$$\rho_{air} = 0.00000584000 \cdot T^2 - 0.00697400 \cdot T + 2.72558 \quad (300\text{K} < T < 500\text{K}) \quad \text{Equ. 5.64}$$

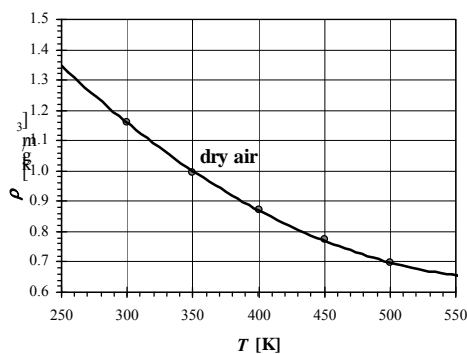


Fig. 5.4. Values of density of dry air from Incropera and DeWitt (1996) displayed as dots in comparison with the polynomial function in Equ. 5.64 displayed as a curve

5.12 Heat capacity of the flue gas

According to Wester (2002) the heat capacity of the flue gases can be estimated by weighting the heat capacity for the different species by the mass fractions

$$c_{p,g} = Y_{H_2O} \cdot c_{p,H_2O} + Y_{CO_2} \cdot c_{p,CO_2} + Y_{N_2} \cdot c_{p,N_2} + Y_{Ar} \cdot c_{p,Ar} + Y_{O_2} \cdot c_{p,O_2} \quad \text{Equ. 5.65}$$

The mass fractions for the different species are calculated in Table 5.3. To be able to take into account the influence from varying air factors, the mass fractions of the different species have to be expressed as a function of the air factor.

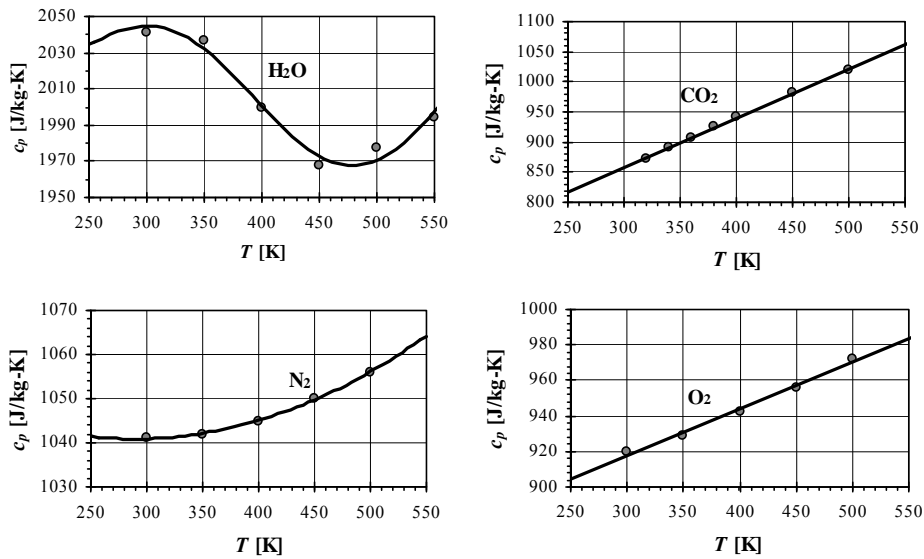


Fig. 5.5. Heat capacity for the different species in the flue gas as a function of temperature. Data for water vapour at atmospheric pressure have been taken from Granryd (1995). The other values have been taken from Incropera and DeWitt (1996) for atmospheric pressure. Data from Granryd (1995) also show that the heat capacity for Argon is almost independent from the temperature and a value of 521 J/(kg·K) can be used over the entire temperature range.

Influence from the temperature on the different c_p - values is shown in Fig. 5.5.

And the regressions for temperatures between 300 and 500 give the following expressions for the heat capacity

$$c_{p,H_2O} = -4.81630 \cdot 10^{-10} \cdot T^5 + 9.63419 \cdot 10^{-7} \cdot T^4 - 0.000735482 \cdot T^3 + 0.266607 \cdot T^2 - 45.9189 \cdot T + 5050.6 \quad \text{Equ. 5.66}$$

$$c_{p,CO_2} = 0.81615 \cdot T + 613.65 \quad (300 \text{ K} < T < 500 \text{ K}) \quad \text{Equ. 5.67}$$

$$c_{p,N_2} = 0.0003429 \cdot T^2 - 0.19829 \cdot T + 1069.5 \quad (300 \text{ K} < T < 500 \text{ K}) \quad \text{Equ. 5.68}$$

$$c_{p,Ar} = 521 \quad (300 \text{ K} < T < 500 \text{ K}) \quad \text{Equ. 5.69}$$

$$c_{p,O_2} = 0.2620 \cdot T + 839.0 \quad (300 \text{ K} < T < 500 \text{ K}) \quad \text{Equ. 5.70}$$

Plotting the results from using Equ. 5.65 To Equ. 5.70 and using mass fractions (Y) from six different air factors give the points in Fig. 5.6. The lines show the calculated values using Equ. 5.70, which is a regression of both T and λ . The curves are quite linear, thus evaluating c_p at the average temperature between flue gas and ambient is a good assumption for the average c_p in the temperature range.

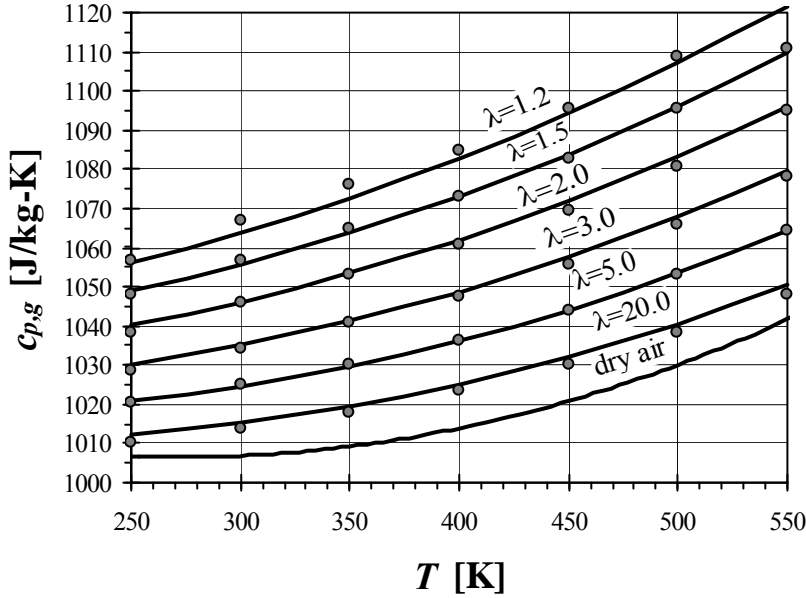


Fig. 5.6. Heat capacity of the flue gas depending on temperature and air factor. The dots show calculated values using Equ. 5.66 and the data from Table 5.3 and Equ. 5.66 to Equ. 5.70. The curves show the result calculated by Equ. 5.71. The line for dry air is calculated from Equ. 5.76. These values are calculated for this particular pellet fuel specified in Table 5.2, moisture content specified in Table 5.3 and for the assumed moisture content in the air.

$$c_{p,g} = (0.01382 \cdot T + 3.188) \cdot (\ln \lambda)^2 - (0.07622 \cdot T + 17.693) \cdot \ln \lambda + 0.0002785 \cdot T^2 + 0.008727 \cdot T + 1043.1 \quad (1.2 < \lambda < 20) \text{ Equ. 5.71}$$

To be able to find a suitable correlation like Equ. 5.71 the following procedure can be followed. Express the heat capacity as a function of the air factor in an equation of the following type

$$c_{p,g} = K_1 \cdot (\ln \lambda)^2 + K_2 \cdot \ln \lambda + K_3 \quad \text{Equ. 5.72}$$

where K_1 , K_2 and K_3 are constants that can be identified for each temperature using the "Solver" in Microsoft Excel trying to minimize the absolute difference between heat capacity calculated by Equ. 5.65 and Equ. 5.72. Then plot the constants towards the temperature and use a linear regression to express K_1 and K_2 and a polynomial of the second order to express K_3 . For this case then the constants are found to be

$$K_1 = 0.01382 \cdot T + 3.188 \quad \text{Equ. 5.73}$$

$$K_2 = -(0.07622 \cdot T + 17.693) \quad \text{Equ. 5.74}$$

$$K_3 = 0.0002785 \cdot T^2 + 0.008727 \cdot T + 1043.1 \quad \text{Equ. 5.75}$$

5.13 Heat capacity of air

The heat capacity of dry air at atmospheric pressure from Incropera and DeWitt (1996) are shown in Fig. 5.7. The data give the following expression when making a regression.

$$c_{p,air} = 0.00045714 \cdot T^2 - 0.24971 \cdot T + 1040.7 \quad (300 \text{ K} < T < 500 \text{ K}) \quad \text{Equ. 5.76}$$

The heat capacity of the air can also be calculated using the same methodology as for the combustion reactions (Equ. 5.65) and with the properties of dry air from Table 5.1 and Equ. 5.66 to Equ. 5.70 almost the same results as with Equ. 5.76 is obtained. Including a moisture content of 0.0004 kg water per kg dry air have any significant influence on the heat capacity.

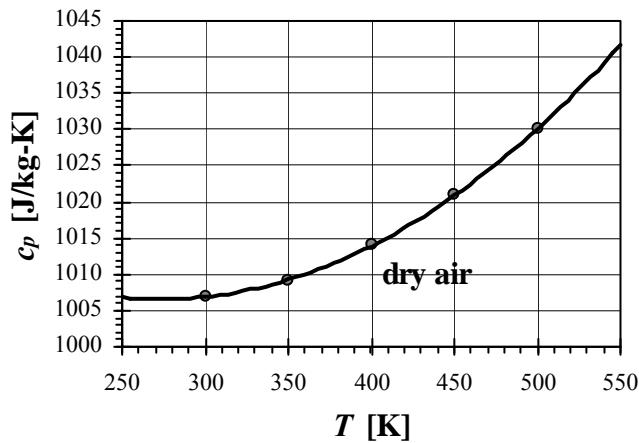


Fig. 5.7. Heat capacity of dry air from Incropera and DeWitt (1996) displayed as dots in comparison with the polynomial functions (Equ. 5.76) displayed as lines.

5.14 Heat rate to the ambient and full energy balance of the boiler

With the setup of the measurements as described in this work it is impossible to directly measure heat rate to the ambient (the room). It can only be calculated from the energy balance of the boiler. The energy balance of the boiler for stationary conditions are:

$$Q_{pell} = Q_{amb} + Q_{wat} + Q_{flu} \quad \text{Equ. 5.77}$$

Or in terms of heat rate

$$\dot{Q}_{pell} = \dot{Q}_{amb} + \dot{Q}_{wat} + \dot{Q}_{flu} \quad \text{Equ. 5.78}$$

If the temperature of the boiler is changed during the measurements also a term of internal energy change must be included.

$$\dot{Q}_{pell} = \dot{Q}_{amb} + \dot{Q}_{wat} + \dot{Q}_{flu} + \dot{Q}_{int} \quad \text{Equ. 5.79}$$

And it can be calculated as

$$\dot{Q}_{int} = \frac{m_{wat} \cdot (c_{p,wat} \cdot T_{b2} - c_{p,wat} \cdot T_{b1}) + m_{steel} \cdot c_{p,steel} \cdot (T_{b2} - T_{b1})}{t_2 - t_1} \quad \text{Equ. 5.80}$$

When identifying the parameters for the UA-values using a spreadsheet with the boiler model included the average heat rates of combustion, to the water, to the flue gas and to the ambient should be given. If the boiler water temperature have changed during the measurement sequence a correction due to the internal energy change must be made. As the change of the temperature of the water in the boiler is energy mainly up taken by the water and the steel it is suggested that the heat rate to the water includes both the terms $\dot{Q}_{wat} + \dot{Q}_{int}$.

5.15 Efficiency of boilers and stoves

The efficiency for a boiler can be calculated from

$$\eta_{boil} = \frac{Q_{wat} + Q_{int}}{Q_{pell}} \quad \text{Equ. 5.81}$$

where Q_{wat} is the energy to the water and Q_{pell} is the energy content of the consumed pellet. In a stove also the heat rate to the ambient air is considered to be useful heat and the efficiency can be calculated as

$$\eta_{stove} = \frac{Q_{wat} + Q_{amb} + Q_{int}}{Q_{pell}} \quad \text{Equ. 5.82}$$

5.16 Emission calculations

The emissions are measured as volume concentrations in dry gases in PPM (parts per million). Higher air factor will give lower values in PPM, though the emissions are constant which makes it necessary to relate the emissions to the combusted pellet in [kg/(J fuel)] or similar to be useful as a parameter in the simulation model. The combustion calculation in Table 5.3 give the number of moles of dry flue gases as a function of the air factor, where the number of moles of CO can be calculated from the following equation during stationary combustion conditions.

$$N_{CO} = \chi_{CO} \cdot N_{flue} \quad \text{Equ. 5.83}$$

Where N_{flue} is the number of moles of dry flue gas given from the combustion calculations in Table 5.3 or by Equ. 5.33. N_{flue} is dependent on the air factor and for the actual fuel in this study a linear regression give the following correlation with λ .

$$N_{flue} = 0.214317 \cdot \lambda - 0.00217122 \quad \text{Equ. 5.84}$$

Multiplying with the molecular weight of CO (=28.012 kg/kmol) gives the mass of CO and by dividing with the heating value of the wood pellet and the consumed wood pellet give the CO emissions per J fuel.

$$\frac{m_{CO}}{Q_{comb}} = \frac{\chi_{CO} \cdot N_{flue} \cdot MW_{CO}}{LHW \cdot m_{pell}} \quad \text{Equ. 5.85}$$

When N_{flue} is calculated from the combustion calculations in Table 5.3 it assumes combustion of 1.0627 kg humid pellet, which must be correlated to the actual amount of combusted pellet

$$\frac{m_{CO}}{Q_{comb}} = \frac{\chi_{CO} \cdot \frac{m_{pell}}{1.0627} \cdot N_{flue} \cdot MW_{CO}}{LHW \cdot m_{pell}} = \frac{\chi_{CO} \cdot N_{flue} \cdot MW_{CO}}{LHW \cdot 1.0627} \quad \text{Equ. 5.86}$$

Assuming that $\chi_{CO} = \omega_{CO}$ will make it possible to use the above equation for calculating the CO emissions. During the start and stop sequences the CO emissions cannot be calculated from the knowledge of the combustion conditions, as they are not known. Here the CO must be calculated in absolute terms based on the measured flow rate of the flue gas. For measurements of one time step the CO-emissions are

$$m_{CO} = \dot{V}_{CO} \cdot \rho_{CO} \cdot t_s \quad \text{Equ. 5.87}$$

The mass of CO-emissions for at time step will be calculated based on the following equation.

$$m_{CO} = \omega_{CO} \cdot \dot{V}_{g,dry,Tref} \cdot \rho_{CO,Tref} \cdot t_s \quad \text{Equ. 5.88}$$

Where the volume flow rate of the dry flue gas can be estimated the following way

$$\dot{V}_{g,dry,20^\circ C} = \frac{\dot{m}_g - \dot{m}_{H_2O,g}}{\rho_{g,dry,Tref}} \approx \frac{\dot{m}_g - \dot{m}_g \cdot Y_{H_2O,g}}{\rho_{g,dry,Tref}} \approx \frac{\dot{m}_g - \dot{m}_g \cdot x_{H_2O,air}}{\rho_{g,dry,Tref}} \quad \text{Equ. 5.89}$$

Where \dot{m}_g can be calculated using Equ. 5.46 or Equ 5.47. The reference temperature should theoretically be the temperature of the flue gas inside the instrument, when the concentration is measured, however due to calibration issues it may be better to relate to the reference temperature of the calibration gas, which is usually 15°C.

The combustion calculations in Table 5.3 can be used to estimate the mass flow of water in the flue gas depending on the air factor. A correlation can be found in Equ. 5.90. If it is assumed to be mainly air in the flue gas the assumed moisture ratio in the air can be used. Observe that the simplified expression using the moisture content of air ($x_{H_2O,air}$) does not follow the definition in Equ. 5.4, however the influence on the accuracy is small compared to the uncertainty in the flow rate measurements.

A regression of the mass fraction of water in the flue gas as a function of λ from the combustion calculation in Table 5.3 are shown in Fig. 5.8 and give the following correlation.

$$Y_{H_2O,g} = 0.081009 \cdot \lambda^{-0.746091} \quad (1.2 < \lambda < 30) \quad \text{Equ. 5.90}$$

During ignition and cooling down phase the air factors are large and the combustion calculations cannot be used. For these operation times the humidity of air should be used.

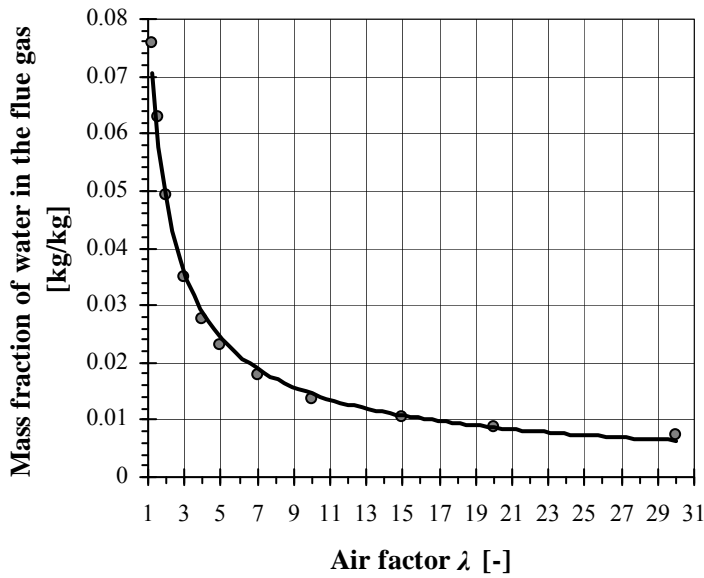


Fig. 5.8. The mass fraction of water in the flue gas $Y_{H_2O,g}$ as a function of the air factor λ from the combustion calculation in Table 5.3 displayed as dots and the values of Equ. 5.90 as a curve.

5.17 Conversion to other units

Concentrations of the species have previously been expressed only in molar fraction, while measurement instruments usually give the readings in volume fraction. To convert from volume fraction to molar fraction, the following equation can be used (Wester 2002), where values of z for different species are given in section 5.10:

$$\chi_i = \frac{\omega_i \cdot z_i}{\sum_{i=1}^n (\omega_i \cdot z_i)} \quad \text{Equ. 5.91}$$

6 Parameter identification

6.1 Procedure

The following procedure can be recommended in order to identify parameters and verify the Type 210 from measurements on a boiler or a water jacket stove.

- 1) The lower heating value (parameter 3) of a test sample is determined by an authorised test institute. From the combustion calculations (Table 5.3) parameters 2 can be identified.
- 2) From the measurements of stationary operation data for max, min and operation power, the UA values (parameter 21 to 30) can be identified using a separate excel sheet with the theory of the model implemented as a Visual Basic program. To minimise the error many sequences shall be used, but with an even distribution of heat rate. The heat rate to the water circuit, the heat rate to the flue gases and the heat rate to the ambient air is compared in the identification process. To avoid the deviation between the temperature in mass 2 and the measured water temperature, the UA value between the water and mass 2 (parameter 27) shall be very large. The parameters defining the heat losses at no combustion UA_{m1-Amb} and UA_{m1-m2} (parameters 25 and 29) shall be correlated to each other by the following equation

$$UA_{m1-Amb} = \frac{1}{\frac{1}{UA_{loss}} - \frac{1}{UA_{m1-m2}}} \quad \text{Equ 6.1}$$

where the heat loss coefficient is calculated from a test where the boiler is heated by electricity with the chimney blocked to avoid all chimney losses. The following equation can be used to calculate the heat loss coefficient from heat loss measurements for the boiler.

$$UA_{loss} = \frac{Q_{el,h+b}}{\left(\overline{T_{boil,h+b}} - \overline{T_{amb,h+b}}\right) \cdot t} - \frac{Q_{el,h}}{\left(\overline{T_{boil,h}} - \overline{T_{amb,h}}\right) \cdot t} \quad \text{Equ 6.2}$$

The first term is total heat loss coefficient from the boiler and the electrical heating circuit and the second term the heat loss coefficient for only the heater.

- 3) From the stationary combustion measurements also the maximum combustion power and the minimum combustion power (parameter 7 and 8), the air factor (inputs 2 and 3) and the CO-emission parameters (parameters 31 and 32) can be identified from these data. From the same measurements also data of electricity consumption should be taken and the parameters 11 and 12 can be identified.
- 4) From a start and a stop sequence where the operation of the fan and the pellet feeder are measured the parameters 13, 14 and 15, can be identified. By using the flow rates measured by the Prandtl tube, the flue gas mass flow rate during the stop phase and during standby (parameters 17 and 18) can be identified. The sequence might be different depending if the boiler is warm or cold and if the boiler is operating with stand by combustion or not. The average values of several measurement sequences should be used.

- 5) The leak flow rate is set by parameter 18 and is calculated the following way internally in the model according to Nordlander (2003) which means that it varies with the outgoing gas temperature and the outdoor temperature.

$$\dot{m}_{air} = \dot{m}_{air50} \cdot \sqrt{\frac{T_g - T_{outd}}{50}} \quad \text{Equ 6.3}$$

In practise the leak flow rate will not only vary depending on the outdoor temperature and the outgoing gas temperature but also with the chimney height and diameter, the thermal mass and the insulation standard of the chimney, if there is a valve that maximise the under pressure in the chimney and the actual pressure in the boiler room relative the outdoor pressure.

It is good if the total heat loss coefficient of the boiler have been measured at different under pressures in the combustion chamber like in Fig. 6.1. If the measurements have been performed at a specific under pressure in the combustion chamber and the measurements shall be used for the validation sequence an artificial outdoor temperature must be calculated to give the right leak flow rate at no combustion. The following equation, derived from Equ. 6.3 and shall be used.

$$T_{outd} = T_g - 50 \cdot \left(\frac{\dot{m}_{air}}{\dot{m}_{air50}} \right)^2 \quad \text{Equ 6.4}$$

The coefficient \dot{m}_{air50} is the leakage flow rate of air at 50K difference between the gas leaving the boiler and the outdoor temperature (Nordlander 2003). Assuming this temperature difference for an outdoor temperature of 0°C and a 5 m high chimney and ignoring friction losses, that the gas is cooled down along the chimney and that the gas might contain other substances near a combustion period it cause an under pressure of

$$p = (\rho_{air,50^\circ C} - \rho_{air,0^\circ C}) \cdot g \cdot H \quad \text{Equ 6.5}$$

Evaluating the densities using Equ. 5.64 and assuming a chimney height of 5 m give the actual under pressure from Equ 6.5. The heat loss coefficient for the self draught $UA_{los,flu}$ can be derived from Fig. 6.1, where the heat losses have been measured depending on the under pressure in the chimney. The mass flow rate at this particular condition can then be calculated as

$$\dot{m}_{air} = \frac{UA_{los,flu}}{c_{p,air,25^\circ C}} \quad \text{Equ 6.6}$$

- 6) Electricity consumption parameters (Parameters 10, 11 and 12) shall be identified from electric consumption measurements of the boiler. An average power consumption during the start sequence shall be measured and for the operation electricity consumption should be measured at different combustion powers and at standby. The model assumes a linear dependence on combustion power.
- 7) From a cooling off sequence where the chimney is blocked to avoid flue gas losses the thermal masses of the boiler (parameters 5 and 6) can be adjusted during simulations of the cooling of sequence. The masses shall be adjusted so that the simulated temperatures of the masses correspond well with the measured boiler water temperature during the whole cool-

ing down period. Particularly for a stove with large amount of heat rate to the room it is more important to compare the measured temperature at the convection air outlet and the glass surface with the simulated heat rate to the room both at the start up period and the cooling down period. Increasing the thermal mass 1 will have a delaying effect on the heat rate to the ambient.

- 8) The time constant for the after glow phase can be identified by simulating a measured stop sequence and compare the measured boiler water temperature (or air outlet temperature and glass temperature for a stove) with the temperature of the thermal masses.
- 9) Verification of the model should be performed to show that the model can make predictions during real operation conditions. For this a sequence including real operation conditions should be used. The measurement data should be prepared in that way that combustion power ambient air temperature inlet water temperature and inlet water flow rate are used as input. If a real chimney is used during the measurements the outdoor temperature is given as input and if a fan with a constant draught is used a fictive outdoor temperature is calculated using Equ 6.4.

6.2 Parameters for the Thermia Bionet boiler

In this section the parameter identification of the Bionet boiler serves as an example of the methodology. The parameters are given in Appendix 1 together with the identified parameters for the other stoves and boilers.

The boiler can be operated at three different combustion steps and adjustment of the fuel and air settings were carefully performed at an constant under pressure of 11 Pa in the combustion chamber. The lowest possible air factor was adjusted trying to maintain low emissions of CO and NO_x. The measured air factors are given in Fig. 6.3.

Identified parameters for the different boilers that have been tested are collected in Appendix 1. The table also shows all inputs to the model, where the air factor for some boilers are modelled externally. The parameters are collected from the following references: Persson (2003), Persson (2004), Nordlander (2004a) and Nordlander (2004b). In Persson (2003) and Persson (2004) an old version of the boiler model was used, but the parameters are converted to the relevant parameters for type 210 in Appendix 1. For the model of Biosolpannan that is combined with the type 140 model (Drück, 2000), also these parameters are given in Appendix 1.

6.2.1 Minimal and maximal combustion power

The maximum and minimum combustion power was calculated from an average of pellet consumption during stationary operation on highest respective lowest combustion power. The values was identified to 45 076 kJ/hr and 16 206 kJ/hr respectively.

6.2.2 UA value identification

Microsoft Excel was used for automatic generation of the UA values, however two restrictions were used. The UA value between the water and mass 2 was fixed to 72 000 kJ/(hr·K) to keep the outlet water temperature close to the temperature of mass 2. The UA value between the mass 1 and the ambient air was described using equ. 6.1, where UA_{loss} are 4.24 W/K from Fig. 6.1.

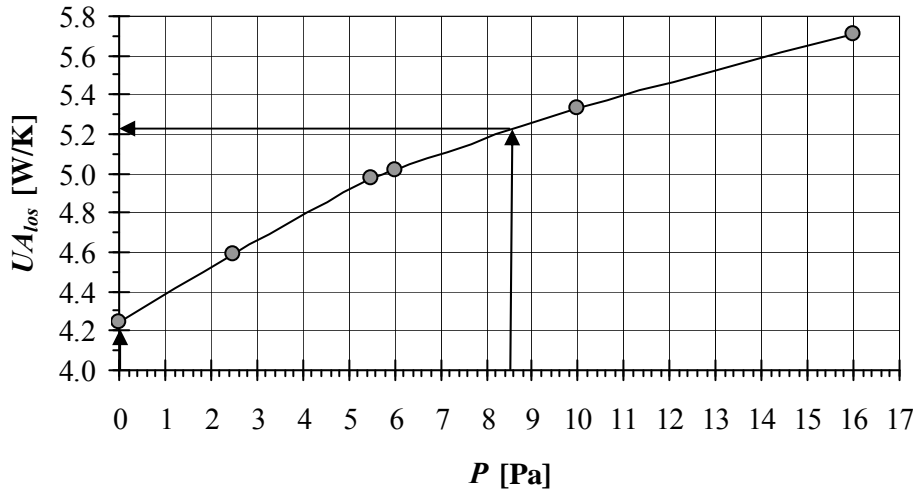


Fig. 6.1. Heat loss coefficient as a function of the under pressure P in the combustion chamber for the Thermia Bionet boiler.

6.2.3 Thermal masses

The thermal masses (parameters 5 and 6) and the time constant for the after glow phase (parameter 16) should be identified. A measured cooling down period with blocked chimney and a start and stop sequence where the water circulation stops at the same time as the fuel feeding stops is used for that purpose (Fig. 6.2 part (a)). The temperature of the thermal mass 2 that models the heated water and the steel in the boiler is adjusted so that the simulated temperature of mass 2 agree with the measured boiler water temperature. In the mathematical model there are no thermal mass of the water, thus the temperature difference between the thermal mass 2 and the water is dependant on the heat rate and the UA-value (parameters 23 and 24). For this reason it can be advantages if parameter 23 has a large value. One way to get a proper water temperature is to combine the model with a mass of water like type 140 (storage tank model). This can be done either by just model the water using type 140 and move the thermal mass from mass 2 to the storage tank model or by using the boiler model only to produce hot flue gas that goes through one heat exchanger of the boiler model.

In Fig. 6.2 part (b), thermal mass 1 and the time constant for the after glow phase are identified. Parameter 5 are adjusted until the simulated heat rate to the water circuit and to the ambient follows the measured data and parameter 16 is adjusted until the temperature increase of mass 2 after the stop agree with the measured water temperature.

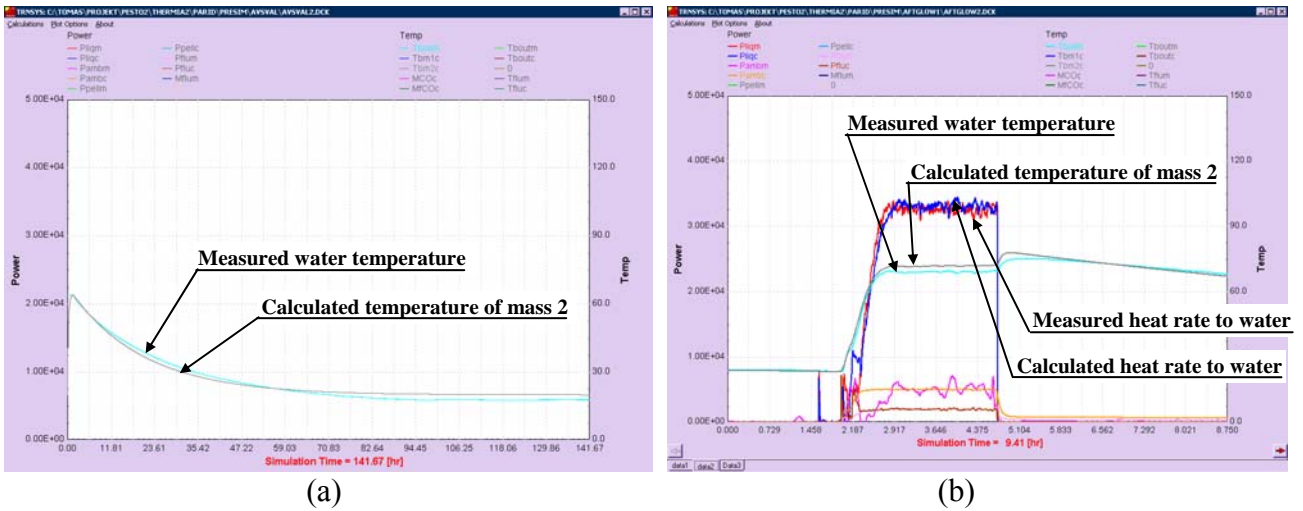


Fig. 6.2. (a) shows simulated temperature of the thermal mass 2 in comparison with measured boiler water temperature for a cooling of sequence with blocked chimney. (b) shows a start and a stop sequence with simulated heat rate to the water and temperature of mass 2 in comparison with the measured heat rate to the water and boiler water temperature.

6.2.4 Air factor

The measured air factor (calculated from and average of Equ. 5.35 and Equ. 5.38) is plotted as a function of the combustion power in Fig. 6.3 and a regression is performed giving the following correlation

$$\lambda = 413.365 \cdot \dot{Q}_{Pell}^{-0.5208664} \quad [-] \quad \text{Equ. 6.7}$$

where \dot{Q}_{Pell} shall be given in [kJ/hr]. A linear correlation as is suggested by the model is not considered to be accurate enough for this type of boiler, thus the air factor must be modelled externally using Equ 6.7.

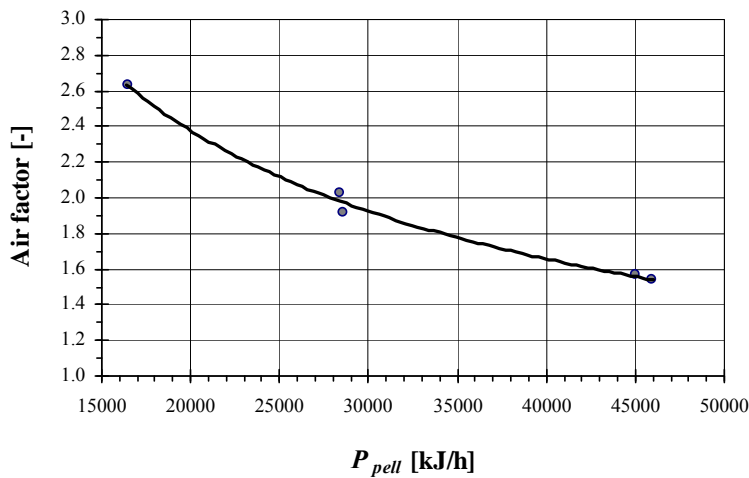


Fig. 6.3. Measured (as dots) and modelled (as a curve) air factor as a function of combustion power for the Thermia Bionet boiler.

6.2.5 CO factor

The CO emissions during stationary operation are plotted in Fig. 6.4 expressed both in [g/MJ] and in [g/h] and the regressions are the following:

$$m_{CO} = 31056043 \cdot \dot{Q}_{P_{ell}}^{-1.917884} \text{ [g/MJ fuel]} \quad \text{Equ. 6.8}$$

$$m_{CO} = 31056.043 \cdot \dot{Q}_{P_{ell}}^{-0.1917884} \text{ [g/hr]} \quad \text{Equ. 6.9}$$

where $\dot{Q}_{P_{ell}}$ shall be given in [kJ/hr]. The model assumes a linear correlation for the emissions, however better agreement with the measured data will be achieved using an exponential function. Thus the emissions are modelled externally using the exponential function in Equ 6.9. The CO-emissions during the start sequence was identified to be 2.2 g/start.

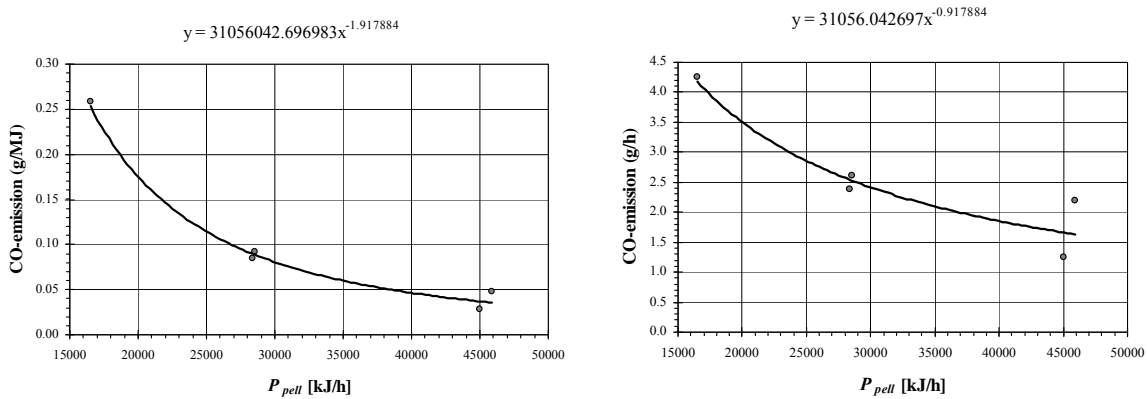


Fig. 6.4. Measured (as dots) and modelled (as a curve) CO emissions as a function of combustion power for the Thermia Bionet boiler.

6.2.6 CO emission for start/stop

During the stop sequence the CO-emissions arise through a long time span according to Fig. 6.5. Thus the emissions must be stopped if the burner starts again during this time span. The stop emissions are therefore modelled externally according to Fig. 6.5. The mass flow of the emissions are modelled as 0.2 h average values for the first three hours and with one hour average values for the last three hours.

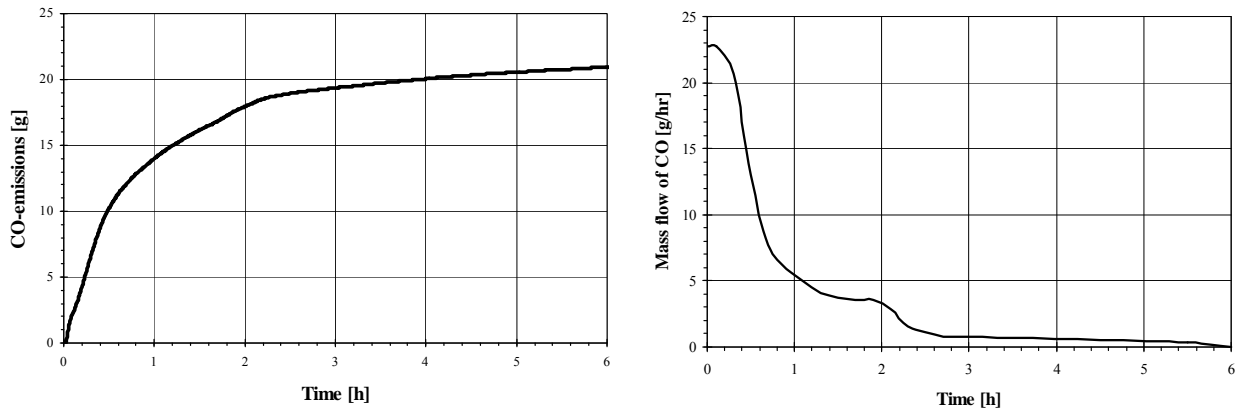


Fig. 6.5. Accumulated CO-emissions and mass flow of CO-emissions during the stop sequence. Average value of two independent stop sequences.

6.2.7 Duration of the two start phases

From measurements of start sequences in Fig. 6.6 (starting by making a new ignition) the duration of the start phases can be identified. The start time of the pellet feeder is found from the voltage measurement of the screw motor. Phase one is assumed to start when the pellet feeder starts and phase 2 when the ignition occurs. There is a period with low fan operation and no pellet feeding during phase 2 before the thermostat is allowed to control the burner.

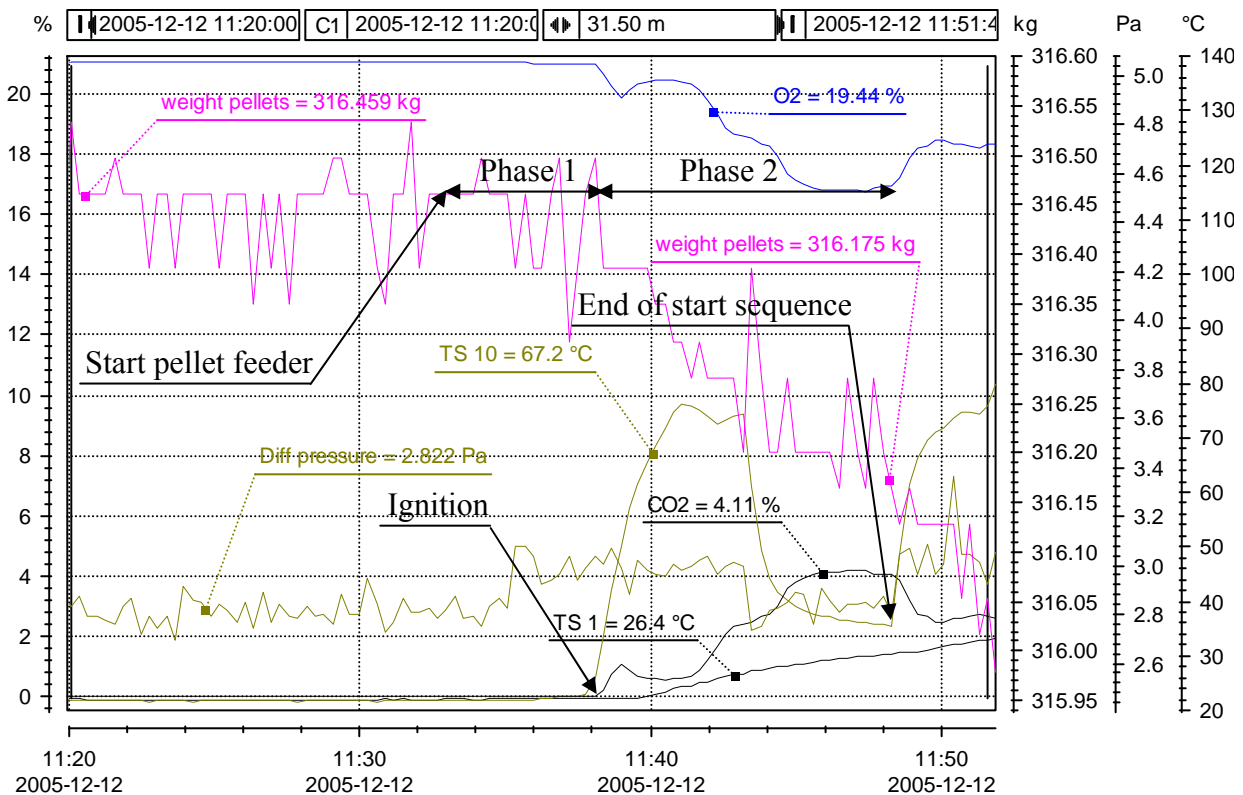


Fig. 6.6. Measurements of a start sequence, starting by making a new ignition. The start time of the pellet feeder is found from the voltage measurement of the screw motor. Phase one is assumed to start when the pellet feeder starts and phase 2 when the ignition occurs. There is a period with low fan operation and no pellet feeding during phase 2 before the thermostat is allowed to control the burner. TS 10 is the flue gas temperature and TS 1 is the boiler water temperature.

6.2.8 Fan operation and mass flow after stop

Parameter (15 and 16) have been identified from measurements of a stop sequence (Fig. 6.7). The stop time of the pellet feeder is found from the voltage measurement of the screw motor and the stop of the fan is found from the voltage measurement of the fan. The mass flow during the fan operation is the calculated average from the Prandtl measurements during the period between the screw motor stops and the fan stops.

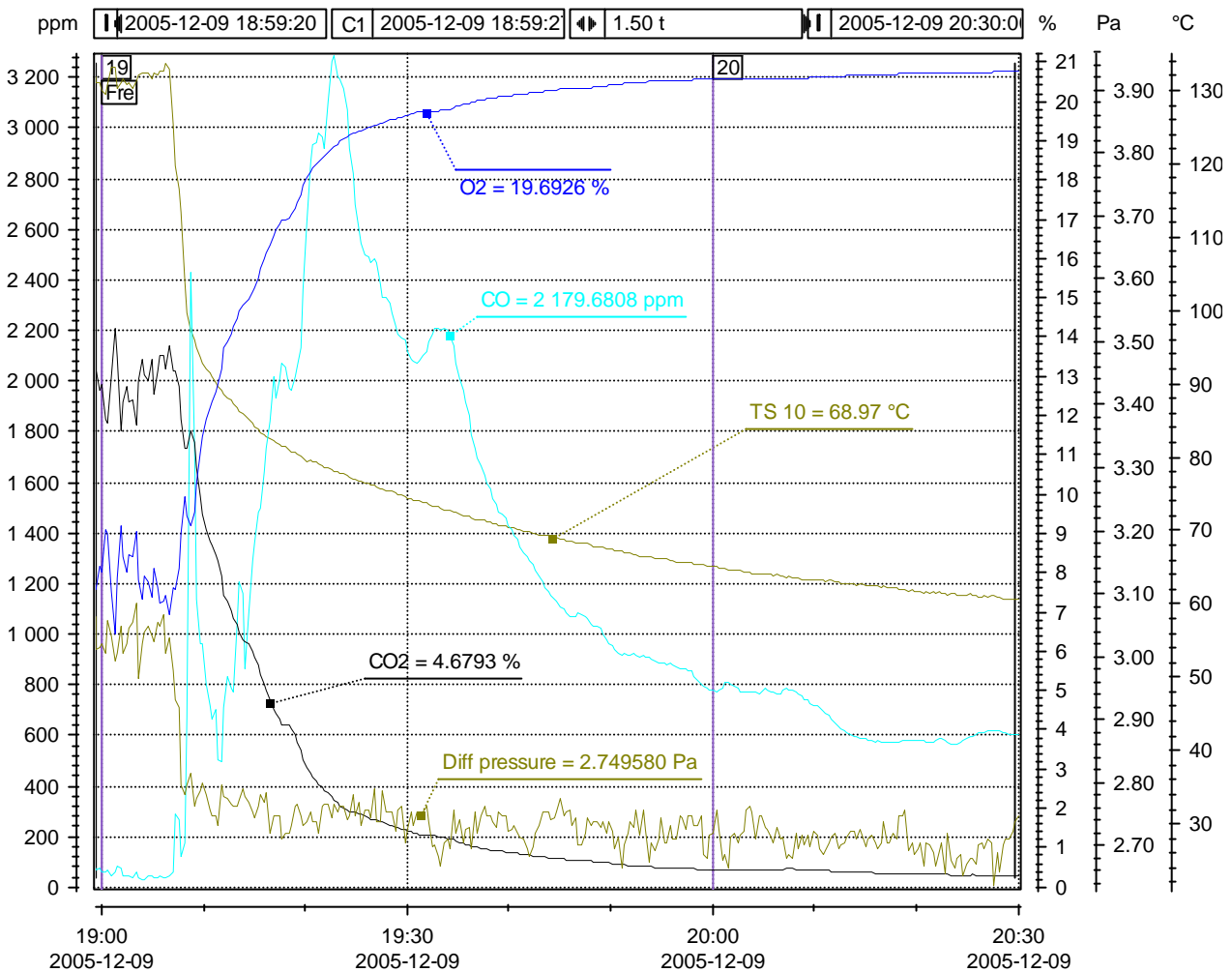


Fig. 6.7. Measurements of a stop sequence. The stop time of the pellet feeder is found from the voltage measurement of the screw motor and the stop of the fan is found from the voltage measurement of the fan. TS 10 is the flue gas temperature.

6.2.9 Leakage flow rate.

When identifying the leakage losses through the chimney during standby (parameter 18) the chimney length must be taken into account. If measurements like in Fig. 6.1 are available, the data must be recalculated for a certain chimney height. The draught in a particular chimney can be calculated from the air density using Equ. 5.64 and assuming a chimney height of 5 m give an under pressure of 8.57 Pa using Equ. 6.5. Using Fig. 6.1 this pressure causing a heat loss coefficient for the self draught $UA_{los,flu}$ of 0.98 W/K. The mass flow rate at these particular conditions can then be calcu-

lated as using Equ 6.6:

$$\dot{m}_{air} = \frac{UA_{los,flu}}{c_{p,air,25^{\circ}C}} = \frac{0.98}{1007.2} = 0.000973 \text{ kg / s} = 3.50 \text{ kg / hr}$$

6.2.10 Electricity consumption

The electrical consumption is modelled as a linear function depending on the combustion power using parameter 11 and 12. Measured electricity consumption at stand by and at maximum combustion power (Fig. 6.8) is used to identify the parameters. The electricity power consumption during the start sequence is calculated from a measured average and was found to be 1094 kJ/hr.

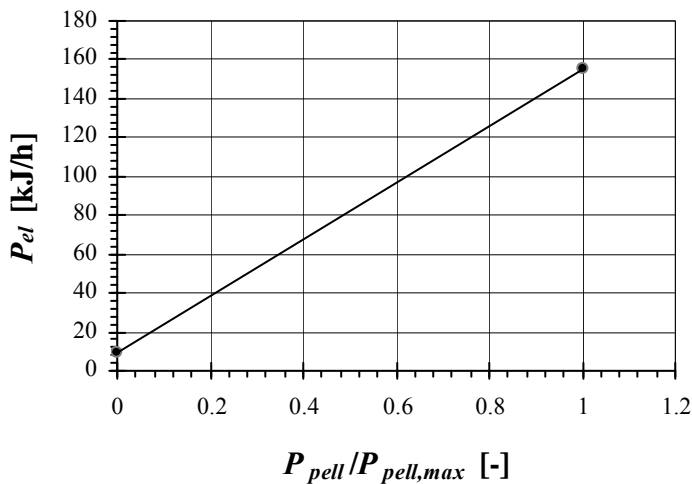


Fig. 6.8. Measured (as dots) and modelled (as a line) electrical consumption of the Thermia Bionet boiler.

6.3 Parameters for the Rebus boiler

The so called REBUS boiler is a prototype of a very compact modulating pellet boiler with a nominal combustion power range between 3.4 and 12 kW. The original pellet heater is manufactured by the Austrian company RIKA and marketed under the name EVO AQUA. This is in fact a water mantled pellet stove that is usually installed in the living area of a house. The reason to use this stove for the REBUS project was that the core part of the stove, the water mantled combustion unit, is very compact and fits into a 60x60cm cabinet. The front window was replaced by a door and insulation was placed around the boiler, reducing the heat transfer to the ambient to a minimum. Furthermore, modifications of the pellet transport system were accomplished. However, modifications of the heat flows have influence on the combustion so that the original settings for the combustion had to be adapted to the changes.

6.3.1 Minimal and maximal combustion power

The original water mantles stove has a total combustion power range from 3.6 to 12.7 kW (BLT, 2003). The pellet supply and by that the combustion power is controlled by the frequency of the feeding screw. This means the rotation speed is constant but the motor is clocked. The boiler has 10 combustion stages with default clocking times. For each combustion stage the boiler is doing a fine

regulation based on the flame temperature measurement in the combustion chamber. After the feeding screw was replaced by a longer flexible feeding screw an adjustment of the motor speed was necessary in order to maintain the same power range. Later, steady state tests with 3 different combustion power stages were performed. From the pellet weight measurement the three combustion powers have been calculated. In figure 6.3.1 these powers are presented in comparison with the original stove for the 10 power stages. It can be seen that the new feeding screw has a non linear characteristic, which has made it necessary to modify also the combustion settings. Using a polynomial fit, the combustion power for the lowest heating stage is 3.4 kW. The maximal combustion power for H10 is 12.04 kW.

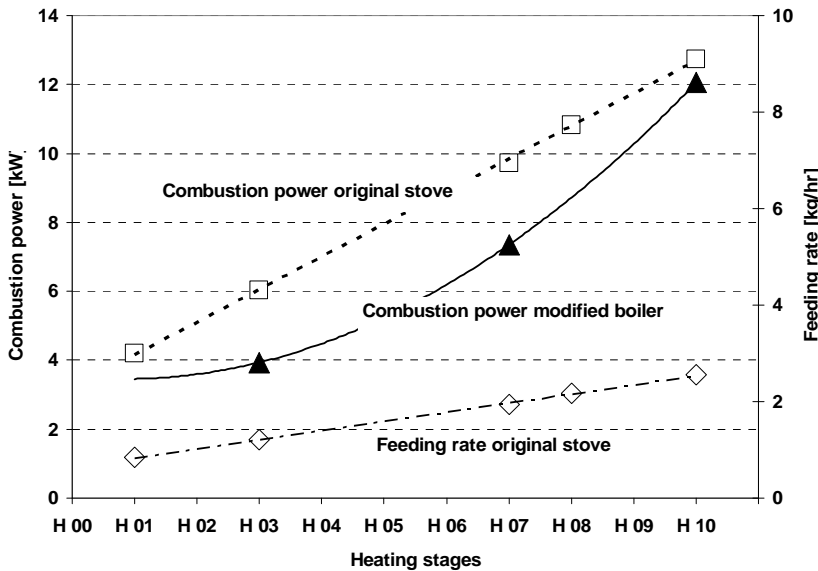


Fig. 6.9. Combustion power of the heating stages for the original and modified pellet heater.

6.3.2 UA value identification

Type 210 models the heat transfer of the boiler by two thermal masses and five heat transfer coefficients (see figure 3.1). These UA-values are not constant rather depending on power and/or temperature. Thus, Type 210 models the UA-values with a constant and a variable coefficient which is dependent on the combustion power. Consequently, 10 values need to be identified with the solver in Excel parameter identification sheet. As input values data from several sequences with constant combustion power are necessary. The sequences should cover the whole combustion power range of the boiler and as more sequences as more accurate results can be expected.

The great number of parameter makes it necessary to reduce the range of parameter so that the solver is not looking for solutions far from realistic values. Here start parameter can be defined which are known from other similar boiler and the first runs can be limited to only a few parameter. It is also possible to let the solver only search for solutions for the power independent parameters. Furthermore restrictions can be added, ranges were the parameter are expected to be found. Moreover the minimal heat loss coefficient can be identified by measurements and can be given as a boundary condition for the parameter identification.

For the Rebus boiler the minimal heat loss coefficient has been determined by several tests where the boiler was heated by pumping water, which is heated by an electrical heater, through the water mantle of the boiler. The obtained heat loss coefficient depends on the water temperature and can be seen in figure 6.3.2. For normal operation conditions that gives heat losses of 150 W to 250 W depending on the boiler temperature.

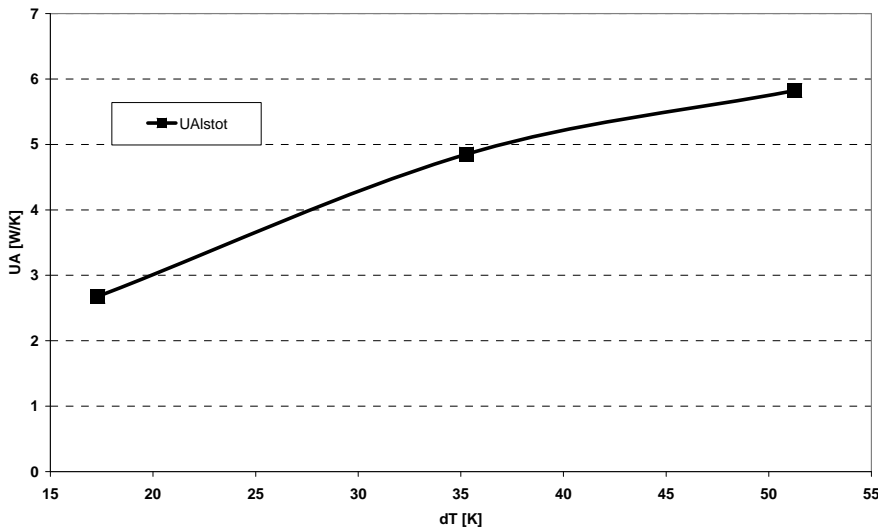


Fig. 6.10. Heat loss coefficient to the ambient depending on the temperature difference between boiler and ambient without combustion

In reality the heat losses during operation of the boiler are much higher (500W – 1000W) since this test does not simulate the heat losses from the front door of the boiler. The largest part of the losses occur from the front door since the door is not water mantled and reaches temperatures above 200°C even if insulated from inside. However, the identified value can be used as a lower limit for the solver. The identified heat loss coefficient can not directly be assigned to the boiler model. The heat losses of the boiler are governed by the heat loss factor UA_{m1-Amb} and UA_{m1-m2} . Both represent the total heat losses from the boiler to the ambient and can be expressed as (see also chapter 6.1):

$$\frac{1}{UA_{loss}} = \frac{1}{UA_{m1-Amb}} + \frac{1}{UA_{m1-m2}} \quad \text{Equ. 6.10}$$

For UA_{lost} the value from the heat loss test is applied and added as a condition for the solver. That allows the solver only to find solutions were UA_{loss} is greater than 5.5 W/K (see also Equ.6.1). The parameter finally identified for all UA-values can be found in Appendix 1.1.

6.3.3 Thermal masses

Before m_2 is going to be determined the after glowing time should be identified (parameter 16). In the case of the REBUS boiler this was done for by looking at the flue gas temperature during the stop phase of the boiler. The time between the boiler was stopped and the flue gas temperature starts to decrease drastically was measured for several sequences. A value of about 2.25 min was identified.

Subsequently, parameter 6 for the thermal mass m_2 was identified by simulating a measured cooling down phase of the boiler and varying the value for m_2 . The gradient of the measured and simulated boiler temperature curves should be as similar as possible (figure 6.3.3a). The thermal mass m_1 was identified by simulating a measured start phase of the boiler. Here m_1 is varied until the temperature of m_1 (this is an output of Type 210) is between the boiler and flue gas temperature with a similar slope (figure 6.3.3b). The two simulations need to be done in turn since the m_1 and m_2 influence both start and stop phase.

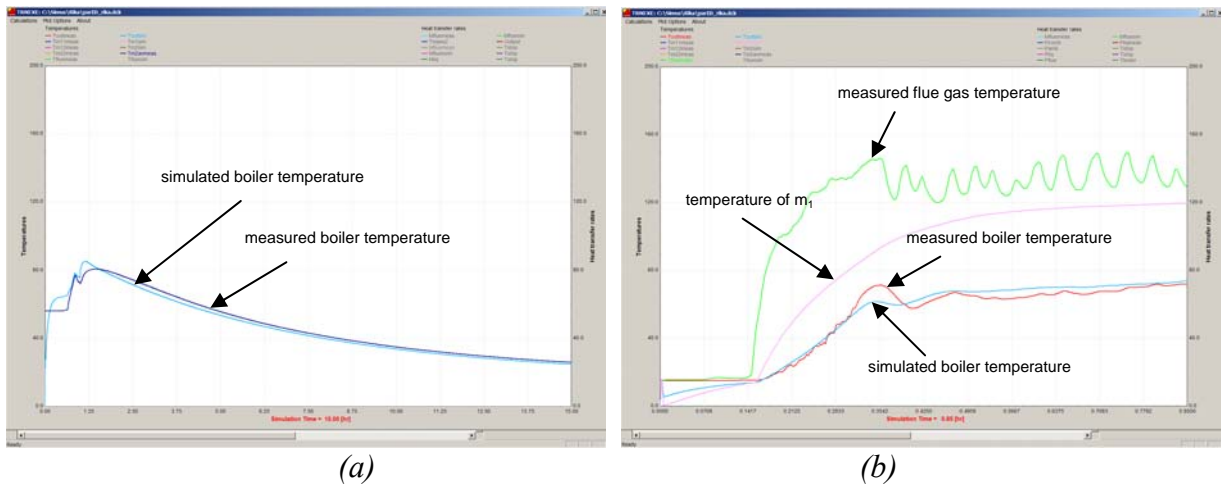


Fig. 6.11. Measured and simulated boiler temperature during cooling down phase of the boiler (a). Measured and simulated boiler temperature, simulated temperature of m_1 and simulated flue gas temperature (b).

6.3.4 Air factor

An equation for the air factors has been determined for three measurements with steady state conditions for different combustion powers of the boiler. The air factors have been calculated based on O_2 values in the flue gas. For each of the three sequences an average O_2 value has been calculated applying equation 5.38. The O_2 value was used due to better calibration of the flue gas analyzer for O_2 compared to CO_2 . The correlation between combustion power and air factor are shown in figure 6.3.5.

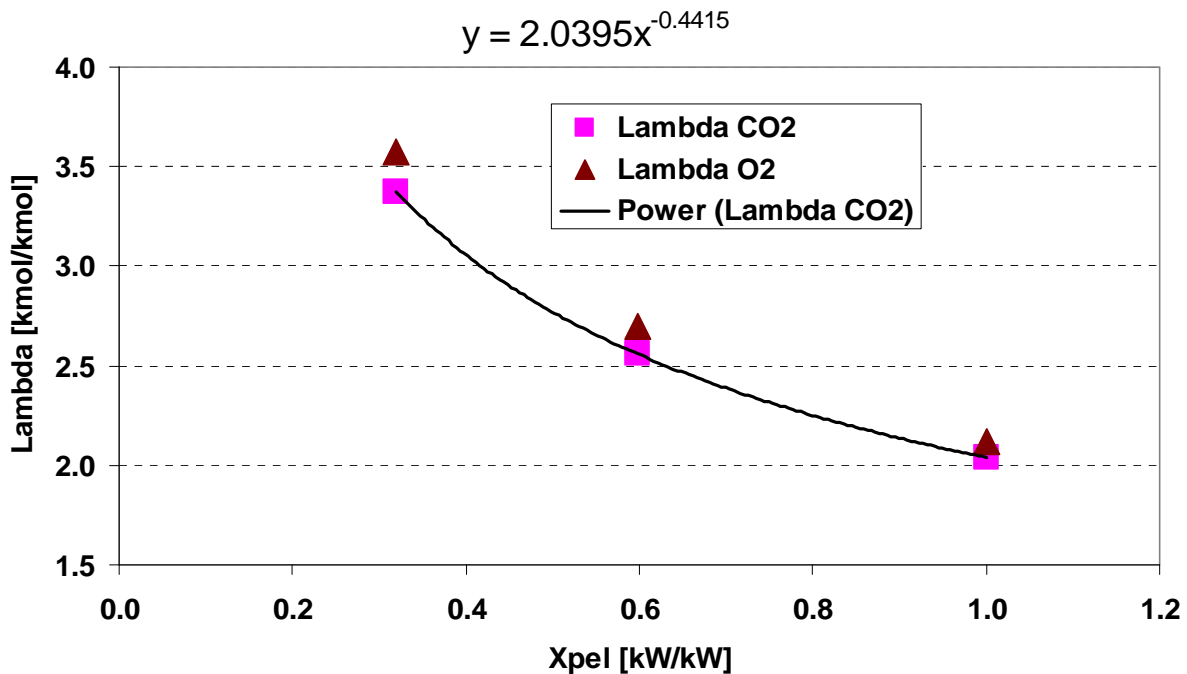


Fig. 6.12. Air factor Λ calculated with equation 5.35 and equation 5.38

6.3.5 CO factor

The values for the CO emissions have been determined after the boiler design modified. Due to these modifications it was also necessary to adjust the combustion settings. The flame set temperature was increased by 3%. Furthermore the combustion air flow was adjusted by decreasing the air flow by 7% for 4 kW, by 5% for 7 kW and 2% for 12 kW. The adjustment criteria were the CO emission. The values are presented in figure 6.3.6. The CO factors are modelled as a fixed and power dependent parameter for Type 210 which makes it difficult used the power fit of the CO emissions. Consequently, the simplified less accurate linear fit was used for the parameter. If the boiler is operated on/off or with a fixed combustion power the correct value should be chosen from the power fit, otherwise the error can be significantly. A similar fit was found for the Bionet boiler so that it is suggested to change the CO emissions from a parameter to a input so that the CO emission can be modelled with an external equation.

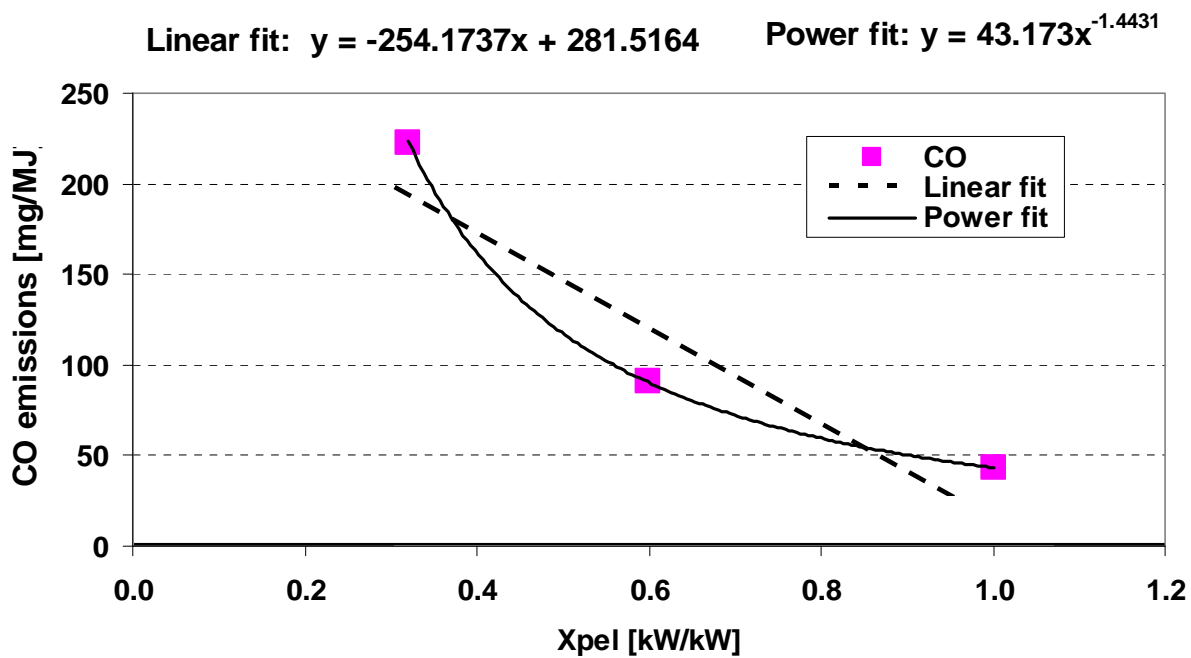


Fig. 6.13. CO emissions depending on the fraction of the maximal combustion power

6.3.6 CO emission for start/stop

The parameters for the CO emissions during start and stop/start have been determined from the test sequences in figure 6.3.7. In the first sequence the boiler starts with a cold water volume and combustion chamber whereas a “warm start” was performed for the second and third sequence. For the start emissions and average value of 4.4 g CO has been determined and for the stop emissions an average value of 1.4 g CO has been determined.

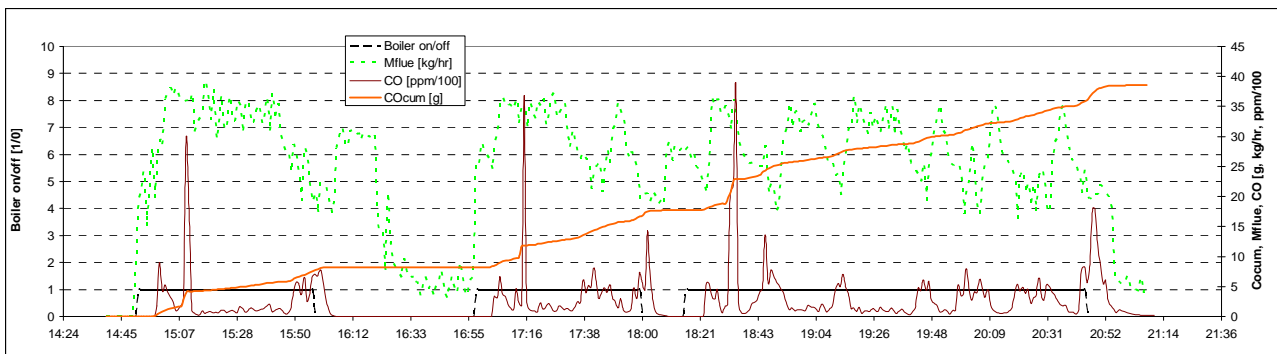


Fig. 6.14. Continuous and cumulated CO emissions for three combustion sequences

6.3.7 Duration of the two start phases

The first start phase without combustion (parameter 13) was measured to be approximately 9 minutes. This is the time the boiler is performing the cleaning of the grate and filling the combustion chamber with the start amount of pellets.

The second start phase has been set to zero and the corresponding combustion power for this phase to 12.04 kW (parameter 9). It has been shown that it is not necessary to model a second start phase since the identified thermal masses create the same effect as a lower combustion power during start – a slow heating up of the boiler.

6.3.8 Fan operation and mass flow after stop

Both parameter (15 and 16) have been identified from lab measurements of the boiler. The values given in Appendix 1.1 are determined for stops when the boiler was before the stop operating with the lowest combustion power. This is assumed to be the normal case when the boiler is operating with modulating power. If the boiler was operating with higher combustion power (e.g. when the boiler is on/off controlled) these values are usually different since the fan is speed controlled depending on the temperature in the combustion chamber.

6.3.9 Leakage flow rate.

The leakage flow rate has been determined from the measurements of the flue gas mass flow measurements. The accuracy of the flue gas mass flow measurement based on the dynamic pressure measurement is rather low. By contrast, for steady conditions the flue gas mass flow rate can be rather exactly determined based on the combustion calculations described in chapter 5. From the three stationary measurements a correction factor for the flue gas mass flow rate was determined and applied. With this calibration the leakage mass flow rate was measured after the boiler was stopped for a temperature difference of 50 K between boiler temperature and ambient (outside) temperature. From two sequences an average value of 1.8 kg/hr was calculated. The obtained value should be validated by other measurements as performed by Nordlander (2004a). Furthermore the effect of under pressure in the combustion chamber should be studied as for the Bionet boiler (chapter 6.2).

6.3.10 Electricity consumption

The parameters for the electricity consumption (par. 10-12) have not been measured for the actual boiler. The values presented in the parameter list in Appendix 1.1 have been taken from the test report for the Evo Aqua stove (BLT, 2003). The values for the boiler ignition power (parameter 10) and the standby operation should be equal since the construction is identical. The power dependent value might deviate as the motor for the actual feeding screw has a higher electrical power than the original one. However, no final decision was made what motor will be used in future.

6.4 Characteristics of the models

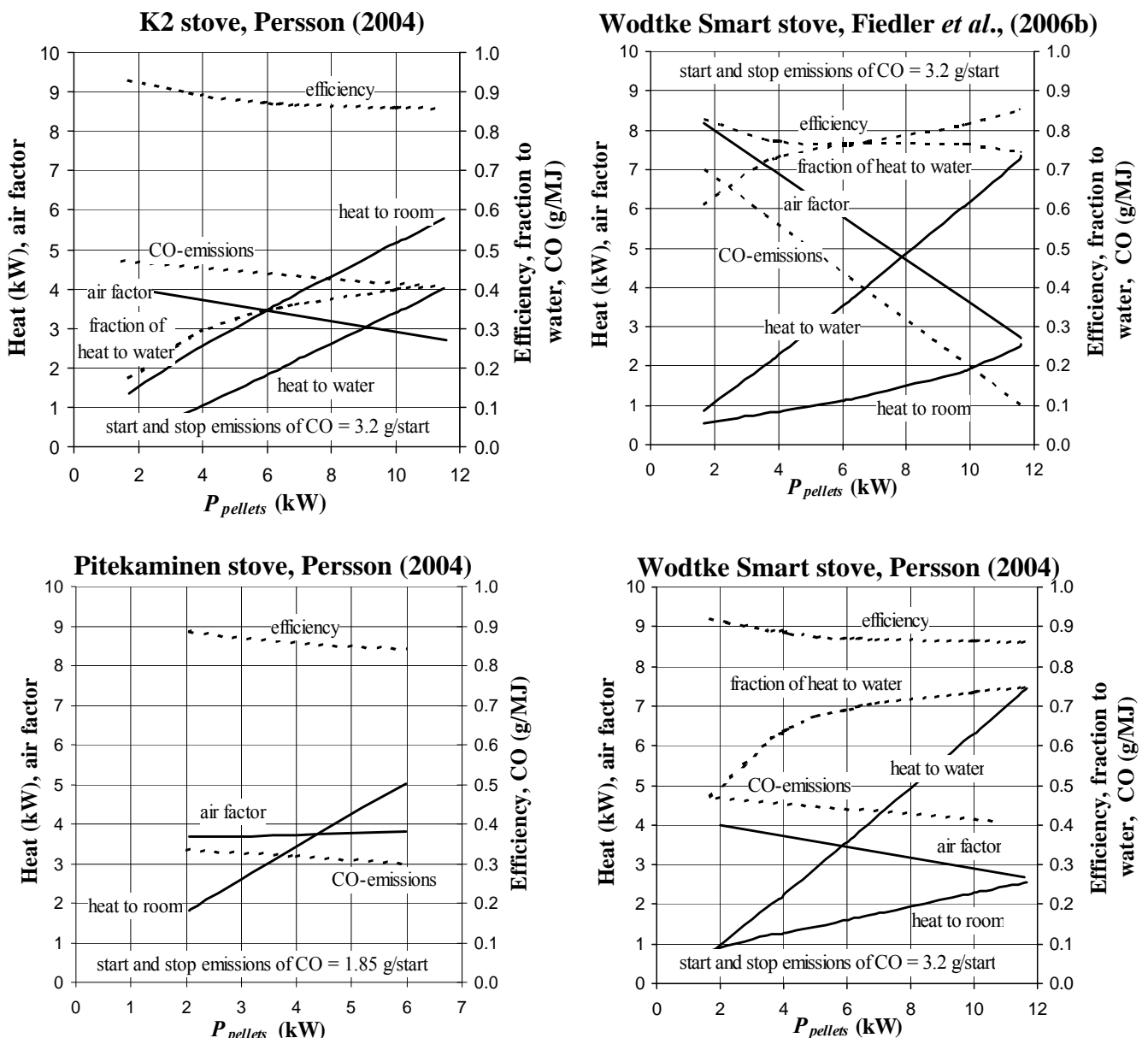
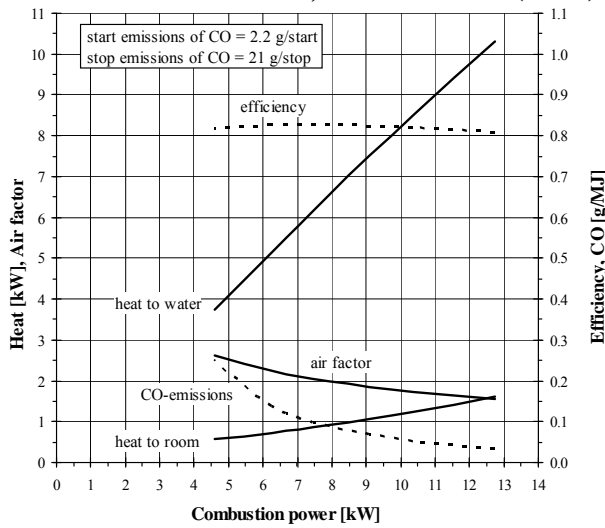


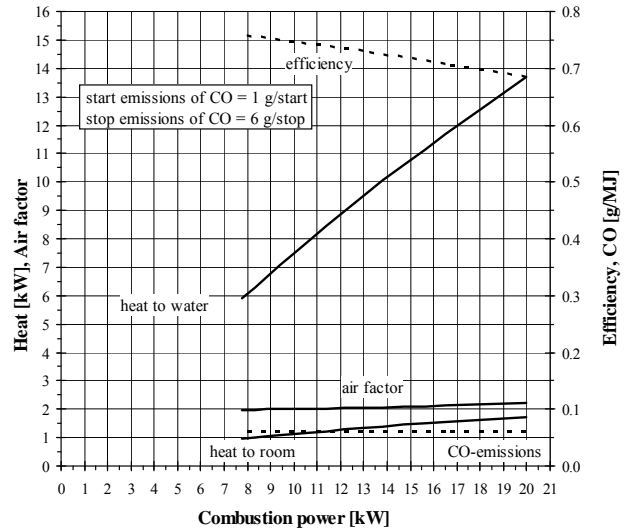
Fig. 6.15. Characteristics of four stove models (Pitekaminen, two versions of Wodtke Smart and K2) at stationary conditions. The inlet water temperature is 55°C for the water jacketed stoves. The water flow rate is 0.111 kg/s for the water jacketed stove and ambient air temperature is 22°C. The stove K2 is a generic stove (a modified version of Wodtke smart used by Persson (2004)). The parameters for the models are collected in Appendix 1:1.

Performance charts of two boiler and two stove models during stationary operation are presented in Fig. 6.15 and Fig. 6.16. Characteristic for all boilers and stoves are decreased air factor and CO-emissions with increased combustion power, except for the Pitekaminen, that has a small increasing air factor for increasing combustion power. The efficiency is higher for the stoves than for the boilers as the heat rate to the room is considered as losses in the efficiency calculation for the boilers, but not for the stoves. The efficiency decreases with increased combustion power for the stoves indicating that the flue gas temperature increase with the combustion power. For the boilers the efficiency is quite constant over the entire operation range with the maximum efficiency at the average combustion power. The major parameters influencing the efficiency is the air factor and the flue gas temperature, and for the boilers of course the heat losses to the room.

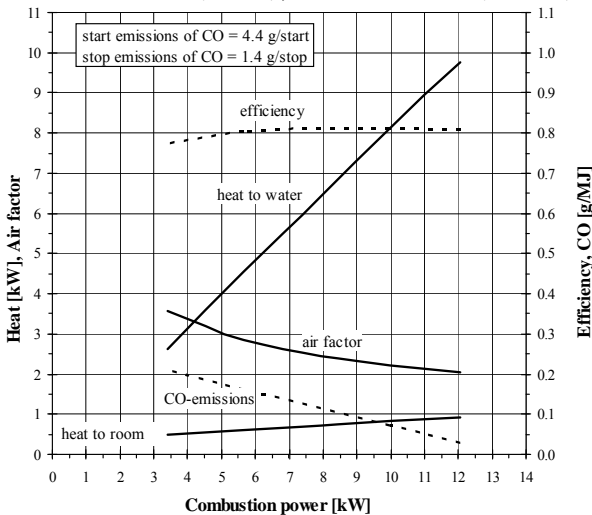
Thermia Bionet boiler, Persson *et al.* (2006)



Thermia Biomatic boiler, Fiedler *et al.* (2006b)



Rebus boiler (Rika), Fiedler *et al.* (2006a)



Pell-X burner + Biosol boiler, Fiedler *et al.* (2006b)

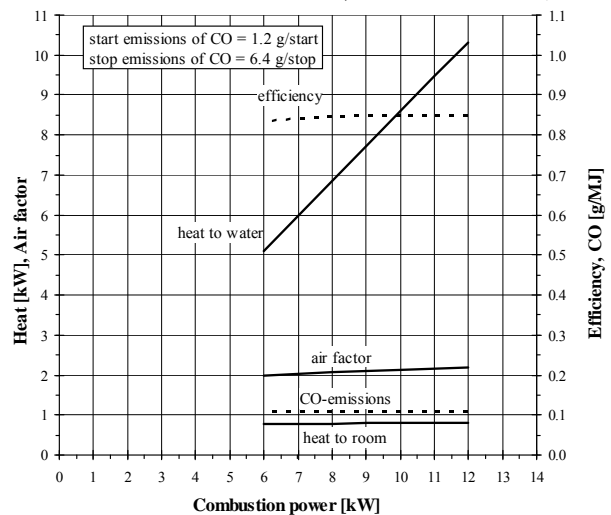


Fig. 6.16. Characteristics of four different boiler models (Thermia Bionet, Thermia Biomatic, Rebus and Pell-X burner + Biosol boiler) at stationary conditions. The inlet water temperature is 65°C for the boilers, the water flow rate is 0.222 kg/s for the boilers and ambient air temperature is 22°C. The Thermia Biomatic boiler is considered to be a generic boiler as the parameter identification process was very simplified. The parameters for the models are collected in Appendix 1:1 and 1:2.

7 Verification and validation

Comparisons between measurements and simulations during stationary operation and dynamic conditions showing high agreement between the measurements and the simulations. The results are shown in figures 7.1 to 7.5 below. Validation or verification of the models comparing with a sequence with realistic operation conditions that not have been used during the identification process has not yet been performed.

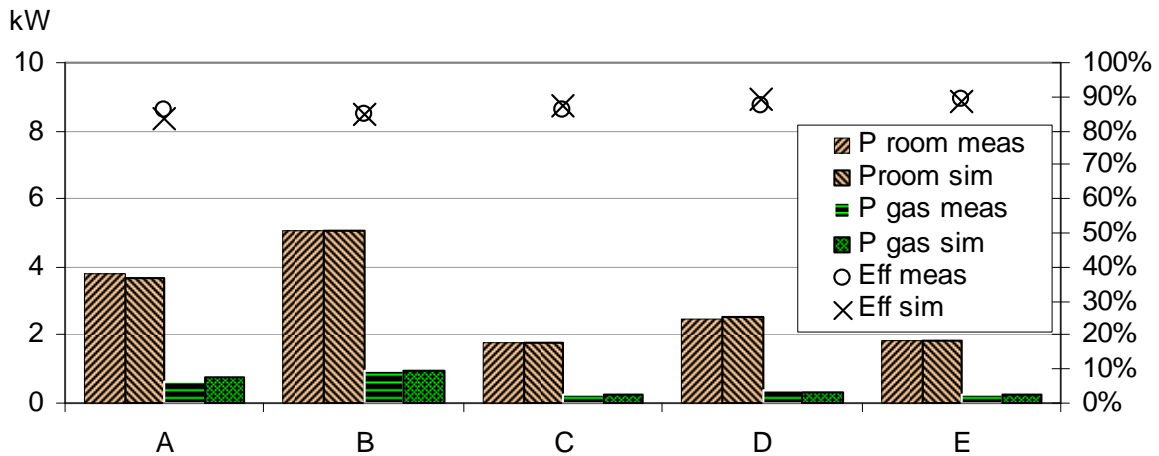


Fig. 7.1. Comparison between measured and simulated values for the stove "Pitekaminen" during stationary operation conditions.

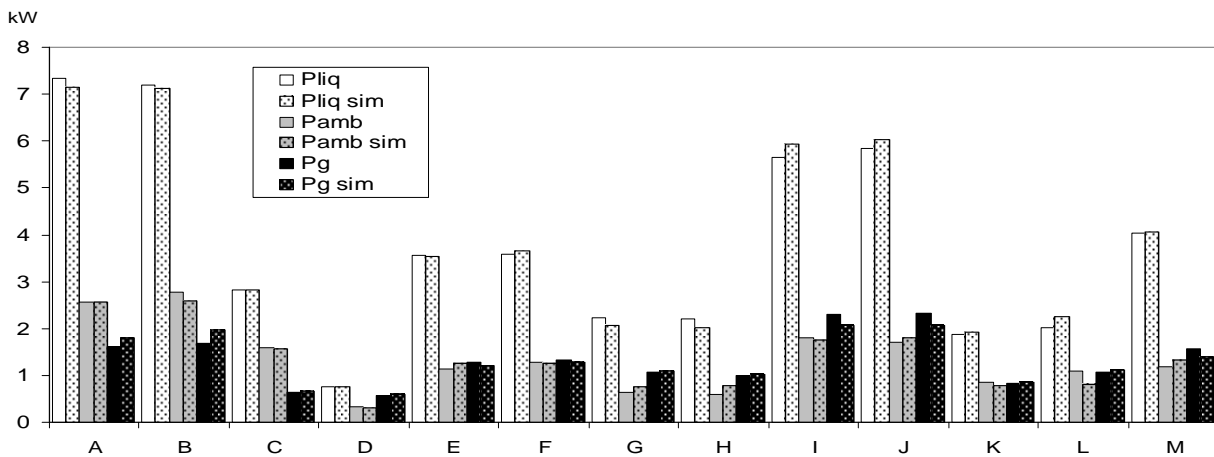


Fig. 7.2. Comparison between measured and simulated values for the stove "Wodtke Smart" during stationary operation conditions.

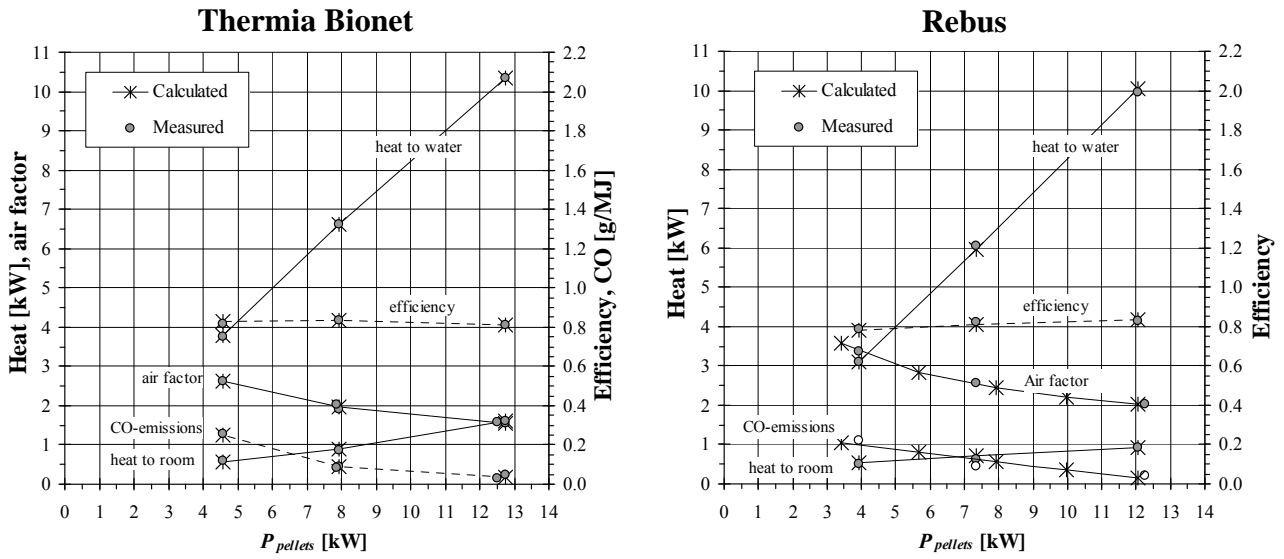


Fig. 7.3. Comparison between measured and simulated values during stationary conditions for the Thermia Bionet boiler and the Rebus boiler.

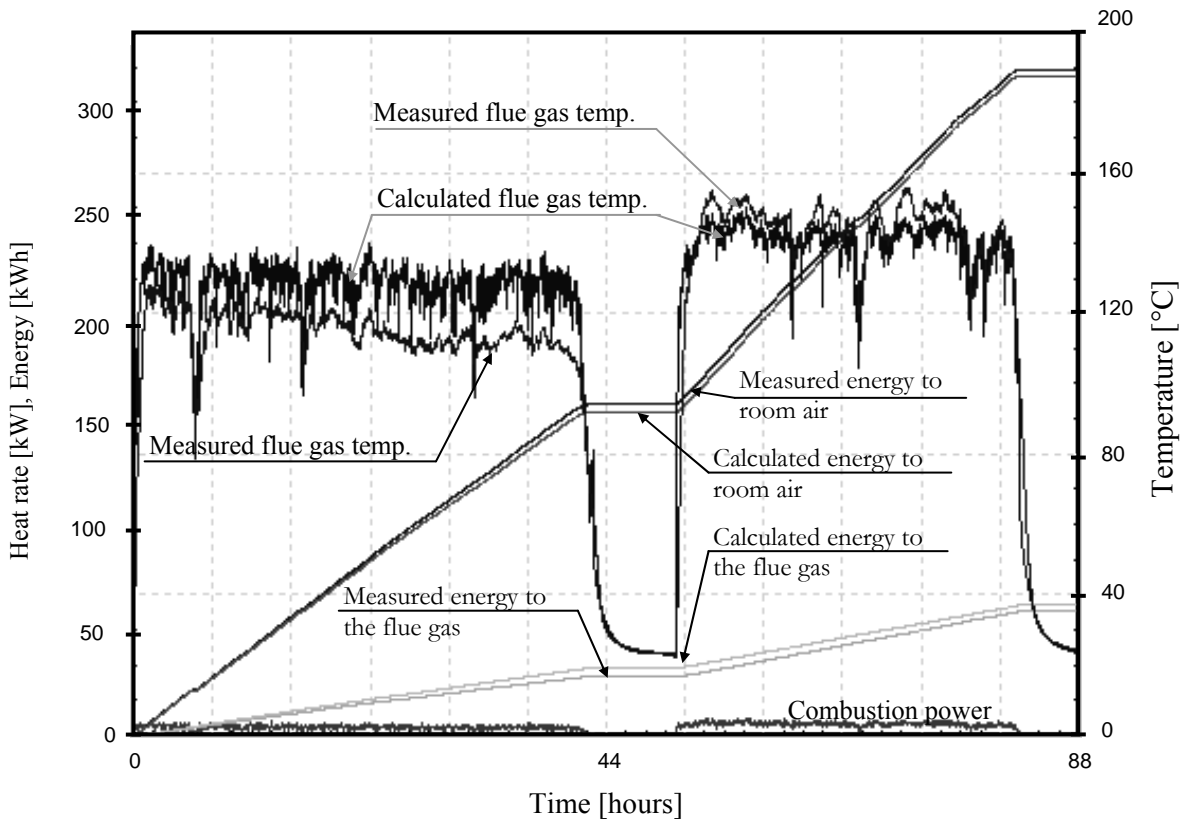


Fig. 7.4. Comparison between measured and simulated values for the stove "Pitekaminen" during dynamic operation conditions.

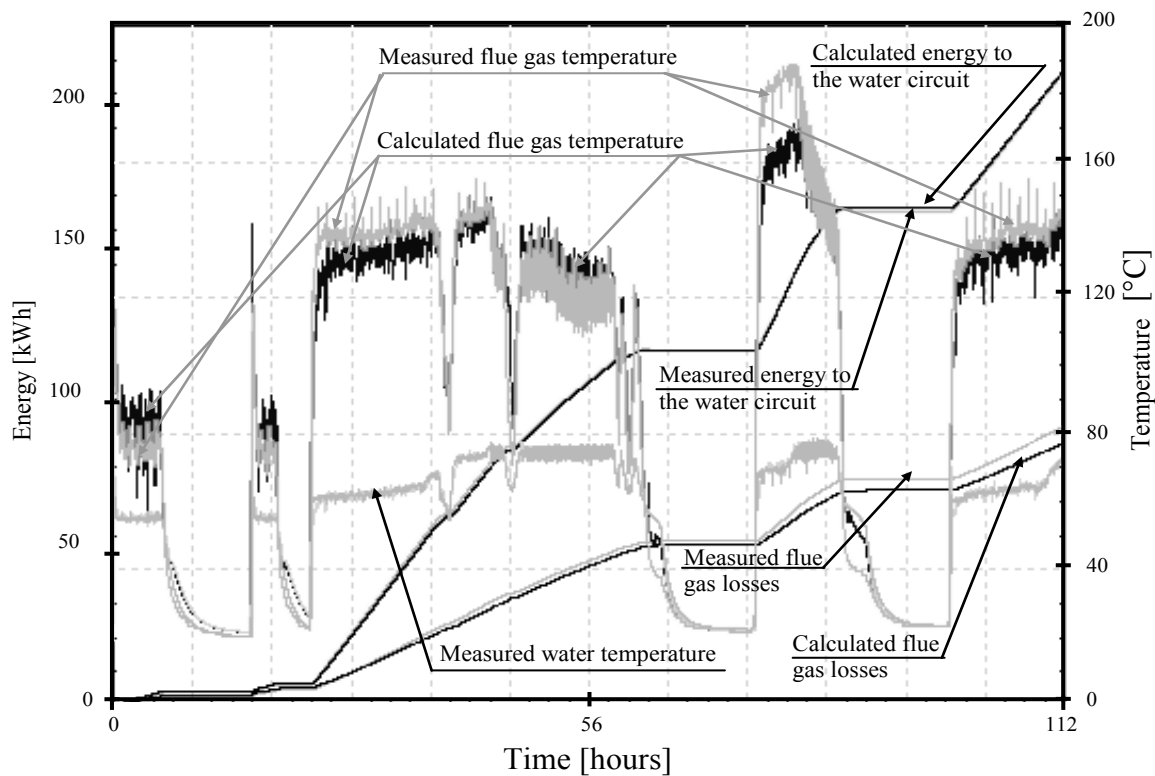


Fig. 7.5. Comparison between measured and simulated values for the stove "Wodtke Smart" during dynamic operation conditions.

From the figures 7.1 to 7.5. it can be observed that the energy balance of the simulations at both stationary and dynamic conditions has proved to have high agreement with measured data. However in section 4, the accuracy of the measurements is discussed showing that there can be quite large uncertainties in the measured heat rate to the ambient and the flue gas flow rate. The uncertainties in the flue gas flow rate cause uncertainties in leakage losses, flue gas losses and CO-emissions.

The major simplification in the model is that the water circuit behaves as a fully mixed water volume, though a real boiler would act more like a plug flow model. However by combining the model with a buffer store model Type 140 as described by Persson (2003) this eventual problem can be solved. The measurements also indicate that an exponential function describing the CO-emissions and the air factor for some cases would be a better option, thus the model should allow higher flexibility for these calculations than just a linear approach.

Nomenclature

A	Chimney area [m ²]
A_f	Air/fuel ratio at stoichiometric combustion [kg humid air/kg humid fuel]
$c_{p,air}$	Heat capacity of air [J/(kg·K)]
$c_{p,Ar}$	Heat capacity of argon [J/(kg·K)]
c_{p,CO_2}	Heat capacity of carbon dioxide [J/(kg·K)]
$c_{p,g}$	Heat capacity of the flue gas [J/(kg·K)]
c_{p,H_2O}	Heat capacity of water vapour [J/(kg·K)]
c_{p,N_2}	Heat capacity of nitrogen [J/(kg·K)]
c_{p,O_2}	Heat capacity of oxygen [J/(kg·K)]
$c_{p,steel}$	Heat capacity of steel [J/(kg·K)]
$c_{p,wat}$	Heat capacity of water [J/(kg·K)]
g	Ground acceleration = 9.82 [m/s ²]
H	Chimney height [m]
K	Koefficient for calculation of density for humid flue gas [kg·K/m ³]
K_{dry}	Koefficient for calculation of density for dry flue gas [kg·K/m ³]
LHW	Lower heating value of the pellet [J/kg]
m_{ash}	Mass of ashes in the fuel [kg]
$m_{air,dry}$	Mass of dry air [kg]
$m_{air,exc}$	Mass of dry excess air [kg]
$m_{air,stoic}$	Mass of dry stoichiometric air [kg]
$m_{C,fuel}$	Mass of carbon in the fuel [kg]
m_{CO}	Mass of emitted CO [kg], [g/MJ fuel]
\dot{m}_{CO}	Mass flow of emitted CO [kg/s], [g/hr]
m_{fuel}	Mass of dry fuel [kg]
$m_{fuel,h}$	Mass of humid fuel [kg]
$\dot{m}_{fuel,h}$	Mass flow rate of combusted humid fuel [kg/s]
m_g	Mass of flue gas [kg]
\dot{m}_g	Mass flow of flue gas [kg/s]
$\dot{m}_{H_2O,g}$	Mass flow of water in the flue gas [kg/s]
$m_{H_2,fuel}$	Mass of hydrogen in the fuel [kg]
$m_{H_2O,air,exc}$	Mass of water in the excess air [kg]
$m_{H_2O,air,stoic}$	Mass of water in the stoichiometric air [kg]
m_{H_2O}	Mass of water [kg]

$m_{H_2O, fuel}$	Mass of water in the fuel [kg]
$m_{O_2, fuel}$	Mass of oxygen in the fuel [kg]
m_i	Mass of substance i [kg]
m_{pell}	Mass flow rate of burned pellet [kg/s]
m_{steel}	Mass of steal in the boiler that is hot, when the water is hot [kg]
MW_{Ar}	Molecule weight of argon = [kg/kmol]
m_{wat}	Mass of water in the boiler [kg]
MW_C	Molecule weight of carbon [kg/kmol]
MW_{CO}	Molecule weight of carbon monoxide [kg/kmol]
MW_{CO_2}	Molecule weight of carbon dioxide [kg/kmol]
MW_{H_2}	Molecule weight of hydrogen [kg/kmol]
MW_{H_2O}	Molecule weight of water [kg/kmol]
MW_{N_2}	Molecule weight of nitrogen [kg/kmol]
MW_{O_2}	Molecule weight of oxygen [kg/kmol]
MW_i	Molecule weight of substance i [kg/kmol]
MW_X	Molecule weight of substance X [kg/kmol]
MW_{air}	Molecule weight of dry air [kg/kmol]
MW_{mix}	Molecule weight of the mixture [kg/kmol]
N_{air}	The number of moles of dry air at real combustion [kmol]
$N_{air, stoic}$	Number of moles of dry air that is required at stoichiometric combustion [kmol]
$N_{Ar, stoic, air}$	Number of moles of Ar in the air at stoichiometric combustion [kmol]
$N_{air, exc}$	Number of moles of dry excess air [kmol]
N_{Ar}	Number of moles of argon in the flue gas at real combustion conditions [kmol]
$N_{Ar, stoic}$	Number of moles of argon in the flue gas at stoichiometric combustion [kmol]
N_{CO_2}	Number of moles of CO_2 in the dry flue gas at real combustion conditions [kmol]
$N_{CO_2, stoic}$	Number of moles of CO_2 in the dry flue gas at stoichiometric combustion [kmol]
$N_{CO_2, air}$	Number of moles of CO_2 in the air [kmol]
N_{flue}	Number of moles of dry flue gas at real combustion [kmol]
$N_{flue, stoic}$	The number of moles of dry flue gas per kg of fuel at stoichiometric combustion [kmol]
N_{H_2O}	Number of moles of H_2O in the flue gas at real combustion [kmol]
N_{N_2}	Number of moles of N_2 in the flue gas at real combustion [kmol]
$N_{N_2, stoic}$	Number of moles of N_2 in the flue gas at stoichiometric combustion [kmol]
$N_{N_2, stoic, air}$	Number of moles of N_2 in the air at stoichiometric combustion [kmol]
N_i	Number of moles of substance i [kmol]

N_{O_2}	Amount of oxygen in the dry flue gas at real combustion [kmol]
$N_{O_2,stoic}$	Required amount of oxygen for stoichiometric combustion [kmol]
N_{tot}	Total number of moles in the mixture [kmol]
P	Pressure [Pa]
P_d	Dynamic pressure [Pa]
$P_{pellets}$	Combustion power [W]
Q	Energy [J]
Q_c	Calculated transferred energy through the heat exchanger [J]
$Q_{el,h}$	Heat losses from the electric heater circuit [J]
$Q_{el,h+b}$	Heat losses from the boiler and the electric heating circuit [J]
Q_m	Measured transferred energy through the heat exchanger [J]
Q_{int}	Energy to internal heat capacity [J]
Q_{wat}	Energy to the water circuit [J]
Q_{pell}	Energy from the combusted pellet [J]
\dot{Q}_{flu}	Heat rate to the exhaust gases [W]
\dot{Q}_{int}	heat rate to internal heat capacity [W]
\dot{Q}_{wat}	heat rate to the water circuit [W]
\dot{Q}_{pell}	Heat rate from the combusted pellet [W], [kJ/K]
Q_{wat}	Energy to the water [J]
Q_{pell}	Energy content of the combusted pellet [J]
Q_{comb}	Combustion power [W]
Re	Reynolds number [-]
R_u	The universal gas constant = 8314.51 [J/(kmol·K)]
t	Temperature [°C]
t_s	length of the time step [°C]
t_l	Time at the start of the sequence [s]
t_f	Time at the end of the sequence [s]
$\overline{T}_{amb,h}$	Average ambient temperature for the test with only the electric heater [K]
$\overline{T}_{amb,h+b}$	Average ambient temperature for the test with both the electric heater and boiler [K]
$\overline{T}_{boil,h}$	Average boiler water temperature for the test with only the electric heater [K]
$\overline{T}_{boil,h+b}$	Average boiler water temperature for test with both the electric heater and boiler [K]
T	Temperature, Average water temperature, Average gas temperature [K]
T_{bl}	Temperature of the water in the boiler at start of the test sequence [K]

T_{b2}	Temperature of the water in the boiler at the end of the test sequence [K]
T_g	outgoing flue gas temperature from the boiler [K]
$\overline{T}_{boil,h}$	Average water temperature of the boiler for the test with only the heater [K]
$\overline{T}_{boil,h+b}$	Average water temperature of the boiler with both the heater and the boiler [K]
T_{bout}	Outlet water temperature of the boiler [K]
t_{bout}	Outlet water temperature of the boiler [°C]
t_{bin}	Inlet water temperature of the boiler [°C]
T_{bin}	Inlet water temperature of the boiler [K]
T_{DHW}	Domestic hot water temperature [K]
T_{flow}	Flow temperature [K]
T_{gout}	Flue gas temperature out of the boiler [K], [°C]
T_{gint}	Inlet air temperature of the boiler [K], [°C]
T_{outd}	Outdoor air temperature [K], [°C]
T_{∞}	Ambient air temperature [K], [°C]
UA_{loss}	heat loss coefficient for the boiler [W/K], [kJ/(hr·K)]
UA_{m1-Amb}	UA-value between thermal mass 1 and ambient [W/K], [kJ/(hr·K)]
UA_{m1-m2}	UA-value between thermal mass 1 and mass 2 [W/K], [kJ/(hr·K)]
\dot{V}_{CO}	Volume flow rate of CO [m ³ /s]
$\dot{V}_{g,dry,Tref}$	Volume flow rate of dry flue gas at reference temperature [m ³ /s]
\dot{V}_{wat}	Water volume flow rate [m ³ /s]
\bar{v}	Average gas velocity [m/s]
$x_{H_2O,air}$	Water content in air per kg of dry air [kg/kg]
Y_i	Mass fraction of component i [kg/kg]
Y_{Ar}	Mass fraction of argon [kg/kg]
Y_{CO_2}	Mass fraction of carbon dioxide [kg/kg]
$Y_{H_2O,g}$	Mass fraction of water in the flue gas [kg/kg]
Y_{N_2}	Mass fraction of nitrogen [kg/kg]
Y_{O_2}	Mass fraction of oxygen [kg/kg]
$Y_{H_2O,fuel}$	Water content in the fuel per kg of humid fuel [kg/kg]
z_{Ar}	Compressibility factor of argon = 1.000 [-] (Vester, 2000)
z_{CO_2}	Compressibility factor of carbon dioxide = 0.993 [-] (Vester, 2000)
z_{H_2O}	Compressibility factor of water vapour = 0.945 [-] (Vester, 2000)
z_i	Compressibility factor of component i [-]

z_{N_2} Compressibility factor of nitrogen = 1.000 [-] (Vester, 2000)

z_{O_2} Compressibility factor of oxygen = 1.000 [-] (Vester, 2000)

Greek

χ Mole fraction [kmol/kmol]

χ_i Mole fraction of spice i [kmol/kmol]

χ_X Mole fraction of spice X [kmol/kmol]

$\chi_{Ar,hf}$ Mole fraction of argon in humid flue gas [kmol/kmol]

χ_{CO} Mole fraction of CO on dry flue gasses [kmol/kmol]

χ_{CO_2} Mole fraction of CO₂ on dry flue gasses [kmol/kmol]

$\chi_{CO_2,air}$ Mole fraction of CO₂ in the dry air [kmol/kmol]

$\chi_{CO_2,hf}$ Mole fraction of carbon dioxide in humid flue gas [kmol/kmol]

$\chi_{H_2O,hf}$ Mole fraction of water in humid flue gas [kmol/kmol]

$\chi_{N_2,hf}$ Mole fraction of nitrogen in humid flue gas [kmol/kmol]

$\chi_{O_2,hf}$ Mole fraction of oxygen in humid flue gas [kmol/kmol]

χ_{O_2} Mole fraction of O₂ in dry exhaust gasses [kmol/kmol]

$\chi_{O_2,air}$ Mole fraction of O₂ in the dry air [kmol/kmol]

$\chi_{CO_2,stoic}$ Mole fraction of CO₂ on dry exhaust gasses at stoichiometric combustion conditions [kmol/kmol]

ρ_{air} Density of air [kg/m³]

$\rho_{air,0^\circ C}$ Density of air at 0°C and atmospheric pressure [kg/m³]

$\rho_{air,50^\circ C}$ Density of air at 50°C and atmospheric pressure [kg/m³]

ρ_{CO} Density of carbon monoxide [kg/m³]

$\rho_{CO,20^\circ C}$ Density of carbon monoxide at 20°C [kg/m³]

ρ_g Density of the flue gas [kg/m³]

$\rho_{g,50^\circ C}$ Density of the flue gas at 50°C and atmospheric pressure [kg/m³]

$\rho_{g,dry}$ Density of the dry flue gas [kg/m³]

$\rho_{g,dry,Tref}$ Density of the dry flue gas at reference temperature [kg/m³]

ρ_{wat} Density of water [kg/m³]

ρ_X Density of gas X [kg/m³]

λ Air factor [-]

ω_{CO} volume concentration of CO on dry gas [m³/m³]

ω_i volume concentration [m³/m³]

References

- Äfab/Konsumentverket, 2005. Test av moderna och automatiskt eldade pellets pannor, Äfab, Lidköping, Sverige, www.afabinfo.com. (Available from: http://www.afabinfo.com/pdf_doc/rapporter/moderna_pellets_pannor_2005.pdf)
- Alvares, H., 1990. Energiteknik, Del 1. Studentlitteratur, Lund, Sweden. ISBN 91-44-31471-X.
- BLT 2003. Prüfbericht Pelletskaminofen EVO AQUA. Wieselburg, Austria, Bundesanstalt für Landtechnik.
- Drück, H., 2000. *Multiport Store - Model for TRNSYS, Type 140, Version 1.99B*. Institut für Thermodynamik und Wärmetechnik, Universität Stuttgart, Stuttgart, Germany. (Available from: http://80.66.96.158/ts/download/doc/typ_140.pdf).
- Fiedler, F., Nordlander, S., Persson, T., Bales, C. 2004. Heat Losses and Thermal Performance of Commercial Combined Solar and Pellet Heating Systems. Intern. Sonnenforum, Proceedings 3, *EuroSun 2004*, Freiburg, Germany, 2004, pp. 437-446.
- Fiedler, F., Bales, C., Persson, T., Thür, A., 2006a. Design method for solar heating systems in combination with pellet boilers/stoves. *Eurosun 2006*, 27-30 June, Glasgow, UK. (www.eurosun2006.org)
- Fiedler, F., Nordlander, S., Persson, T., Bales, C., 2006b. Thermal performance of combined solar and pellet heating systems. *Renewable Energy* **31** (2006), pp. 73-78.
- Granryd, E., 1995. Heat Transfer, Collection of formulas and Tables of Thermal Properties. Department of Applied Thermodynamics and Refrigeration, The Royal institute of Technology KTH, Stockholm, Sweden. (www.kth.se)
- Incropera, F.P., DeWitt, D.P., 1996. *Fundamentals of Heat and Mass Transfer. Fourth Edition*. John Wiley & Sons, Inc. USA. ISBN 0-471-30460-3.
- Klein *et al.*, 2000. *TRNSYS 15, a transient system simulation program –reference manual*. Solar Energy Laboratory, University of Wisconsin-Madison, Madison, USA. (<http://sel.me.wisc.edu/trnsys/>)
- Nordlander, S., 2003, *TRNSYS model for Type 210 Pellet stove with liquid heat exchanger. Documentation of model and parameter identification. Preliminary version, May 2003*. Report ISRN DU-SERC—79—SE, Solar Energy Research Center, Högskolan Dalarna, Borlänge, Sweden. (Available from: <http://dalea.du.se/research/>)
- Nordlander, S., 2004a, Parameteridentifiering för Pitekaminen samt resultat för Wodtke samt ytterst preliminära parametrar för Thermia. Unpublished documentation, Solar Energy Research Center SERC, Högskolan Dalarna, Borlänge, Sweden. (www.serc.se).
- Nordlander, S., 2004b, Parametrar för type 210, Parametrar för Wodtke och Pite, Parameterlista för type 210, Fortran-kod för type 210. Unpublished documentation, Solar Energy Research Center SERC, Högskolan Dalarna, Borlänge, Sweden. (www.serc.se).
- Persson, T., 2003. *Konvertering av elvärmda hus till pellet- och solvärme, -Beskrivning av datormodell för byggnader och system*. Report ISRN DU- SERC—79—SE, Solar Energy Research Cen-

ter, SERC, Högskolan Dalarna, Borlänge, Sweden. (Available from: <http://dalea.du.se/research/default.aspx?authorId=367>)

Persson, T., 2004. *Elbesparing med pelletkaminer och solvärme i direktvärmdda småhus* (Electrical savings by using wood pellet stoves and solar heating in single-family houses heated by electric resistance heating). Licentiate thesis. Trita REFR Report No 04/43, KTH Energiteknik, Royal Institute of Technology, Stockholm, Sweden. (Available from: <http://media.lib.kth.se/licengrefhit.asp?licnr=1766>)

Persson, T., Nordlander, S., Rönnelid, M., 2005. Electrical Savings by Use of Wood Pellet Stoves and Solar Heating Systems in Electrically Heated Single-family Houses. *Energy and Buildings*, Vol. 37, Nr. 9, 2005, pp. 920-929. (Available from: www.sciencedirect.com)

Persson, T., Fiedler, F., Rönnelid, M. Bales, C., 2006. Increasing efficiency and decreasing CO-emissions for a combined solar and wood pellet heating system for single-family houses. Proc on USB of *Pellets 2006*, 30 May - 1 June 2006, Jönköping, Sweden, pp 207-211. (www.pellets2006.com)

Wester, L., 2002. *Förbrännings- och rökgasreningsteknik*. Compendium to the course: "Förbrännings- och rökgasreningsteknik" at Mälardalens Högskola, Västerås, Sweden.

Appendix 1:1. Identified parameters for the different boilers and Stoves

Collection of identified parameters for TRNSYS non-standard Type 210 for the boilers and stoves (Persson, 2003; Persson, 2004; Nordlander, 2004a; Nordlander, 2004b). The parameters in brackets are older parameters used by Persson (2004) and Persson (2005).

Parameter	Stove K1 Pite- kaminen	Stove K3 Wodtke smart	Thermia Biomatic (Prel.)	Thermia Bionet	Rebus Boiler	Biosol and PellX	Unit
1 Operation mode	1	1	1	1	1	1	-
2 Stoichiometric air/fuel ratio	5.57 (5.864)	5.57 (5.864)	5.57	5.8646	5.8646	5.57	kg/kg
3 LHW of fuel	17460(17 480)	17460 (17 480)	17 460	17 964	17 964	17 460	kJ/kg
4 Specific heat capacity of water	4.186	4.186	4.186	4.186	4.186	4.186	kJ/(kg·K)
5 Thermal mass of m ₁	32	13	400	1	45	1	kJ/K
6 Thermal mass of m ₂	1 (0.001)	9 (12)	400	1200	500	1	kJ/K
7 Maximum combustion power	21 528	41 724	72 000	45 076	43380	43 196	kJ/hr
8 Minimum combustion power	7 344	7 200	28 000	16 206	12240	11 000	kJ/hr
9 Combustion power at start phase part 2	16 056	12 276	45 000	21 000	43380	25 000	kJ/hr
10 Electric power at start phase part 1	3168 (0) ⁶⁾	936 (0) ⁶⁾	1 465	1094	1202	2354.5	kJ/hr
11 Electric power consumption at standby	7.2 (0) ⁶⁾	7.2 (0) ⁶⁾	18	9	10.8	32.4	kJ/hr
12 Electric power consumption slope	82.44 (0) ⁶⁾	82.8 (0) ⁶⁾	417.6	146	151.2	417.6	kJ/hr
13 Time for start phase part 1 (no flame)	0.067 (0.075)	0.2 (0.075)	0.125	0.100	0.153	0.1	hr
14 Time for start phase part 2	0.283 (0.275)	0.287 (0.275)	0.2	0.190	0.001	0.2	hr
15 Time of fan operation during stop phase	0 (0.1167)	0.75	0.03	0.025	0.42 ⁵⁾	0	hr
16 Time constant for after glow phase	0.1 (0.01)	0.05 (0.01)	0.03	0.06	0.04	0.1	hr
17 Flue gas mass flow rate during stop phase 1	0 (5)	36	36	18	24.8 ⁵⁾	0	kg/hr
18 Flue gas mass flow rate during standby at (Texh-Tout)=50°C	3	5	6	3.5	1.8	4	kg/hr
19 not used	-	-	-	-	-	-	-
20 not used	-	-	-	-	-	-	-
21 UA gas-m1	71.6	33.08 (32.7)	6.12	104.062	18.092	30	kJ/hr-K
22 UA gas-m1 slope	241.9	66.13 (81.3)	24.5	121.967	3.636	150	kJ/hr-K
23 UA gas-m ₂	0.01 (0)	101.1 (101.9)	244.8	181.500	367.428	0.01	kJ/hr-K
24 UA gas-m ₂ slope	0.01 (0)	202.2 (253.6)	550.8	96.773	26.118	0.02	kJ/hr-K
25 UA m ₁ -Ambient	69.8	13.26 (10.52)	24.5	16.773	23.755	0.01	kJ/hr-K
26 UA m ₁ -Ambient slope	281.9	3.456 (4.07)	0.01	-2.120	7.563	0.7	kJ/hr-K
27 UA m ₂ -liquid	0.01 (0)	720 (1 644.8)	1 346.4	72000.00	5760.000	1	kJ/hr-K
28 UA m ₂ -liquid slope	0.01 (0)	648 (0)	0.01	0.000	-11.693	0.01	kJ/hr-K
29 UA m ₁ -m ₂	0.01 (0)	3.96 (0.36)	306.0	171.500	182.939	0.01	kJ/hr-K
30 UA m ₁ -m ₂ -slope	0.01 (0)	0.36 (0)	0.01	-100.051	-79.219	0.01	kJ/hr-K
31 CO emission factor at P/P _{max} = 0	0.3518	0.8 (0.48)	0.061	- ²⁾	0.282	0.11	g/MJ fuel
32 CO emission factor slope	-0.313	-0.7 (-0.078)	0	- ²⁾	-0.254	0	g/MJ fuel
33 Lumped CO emission during start phase	0.50	2.0	1	2.2	4.4	1.2	g/start
34 Lumped CO emission during stop phase	1.35	1.2	6	21±3	1.4	6.4	g/stop

Input

1 Fraction of maximum combustion	0.341 - 1.0	0.1726-1.0	0.389-1.0	0.359-1.0	0.282-1.0	0.255-1.0	-
2 Air factor at P/P _{max} = 0	3.59	9.1 (4.27)	1.8	- ³⁾	- ⁴⁾	1.8	-
3 Air factor, slope	0.036 (0.2145)	-6.37 (-1.57)	0.4	0	0	0.4	-
4 Set temperature for combustion start	-	-	-	-	-	-	°C
5 Set temperature for combustion stop	-	-	-	-	-	-	°C
6 Temperature for boiler control	-	-	-	-	-	-	°C
7 Ambient room temperature	-	-	-	-	-	-	°C
8 Outdoor air temperature	-	-	-	-	-	-	°C
9 Temperature of combustion air	= input 7	= input 7	= input 7	= input 7	= input 7	= input 7	°C
10 Mass flow rate of incoming water	-	-	-	-	-	-	kg/hr
11 Temperature of incoming water	-	-	-	-	-	-	°C

¹⁾ At P_{pell}=12.165 kW

²⁾ CO emissions are modelled externally (se Fig. 6.4)

³⁾ Air factor is modelled externally (se Fig. 6.3)

⁴⁾ Air factor is modelled as follows: $\lambda=2.0395 \cdot x^{-0.4415}$, where x is the fraction of the maximum combustion power

⁵⁾ only valid if the boiler stops after operating with the smallest combustion power (relatively low flame temperature), for higher combustion power the values become higher since they depend on the temperature in the combustion chamber

⁶⁾ modelled externally in another way using Type 209, se Persson (2004).

External modelling

Fraction of heat to ambient that is convective	70						%
Fraction of heat to ambient that is radiative		30					
CO-emissions				31056043· P _{pell} ^{-1.917884}			g/MJ

Appendix 1:2. Identified parameters for the different boilers and Stoves

Identified parameters by Persson (2003), for the Biosolpannan using type 140 (Drück, 2000).

Parameters		Unit	
1	Height	1,428	m
2	Volume	0,631	m ³
3	Heat capacity of water	4,186	kJ/(kg·K)
4	Density of water	992	kg/m ³
5	Effective vertical heat conduction	1,4	W/(m·K)
8	Heat loss coefficient, bottom	1	W/K
9	Heat loss coefficient, top	0,68	W/K
10	Relative length zone 1	0,54	-
11	Heat loss coefficient from zone 1	3,05	W/K
12	Relative length, zone 2	0,46	-
13	Heat loss coefficient from zone 2	4,29	W/K
17	Relative height inlet of radiator return	0,501	-
18	Relative height outlet radiator supply upper	1	-
21	Relative height inlet radiator return	0,501	-
22	Relative height outlet radiator supply lower	0,614	-
58	Relative height sensor solar collector	0,266	-
61	Relative height sensor pellet burner	0,866	-
65	Relative height electric auxiliary heater	0,708	-
66	Relative height sensor auxiliary heater	0,866	-
67	Stop temperature auxiliary heater	69,0	°C
68	Hysteres electric auxiliary heater	5,0	°C
69	Relative height inlet solar heat exchanger	0,406	-
70	Relative height outlet solar heat exchanger	0,028	-
71	Volume solar heat exchanger	-0,0028	m ³
72	Fluid specific heat in solar heat exchanger	3,78	kJ/(kg·K)
73	Fluid density in solar heat exchanger	1026	kg/m ³
74	UA-value in solar heat exchanger	18,04	kJ/(h·K)
75	Exponent depending on mass flow rate	0	-
76	Exponent depending on temperature difference	0	-
77	Exponent depending on average temperature level	1,163	-
82	Relative height inlet lower DHW heat exchanger	0,186	-
83	Relative height outlet lower DHW heat exchanger	0,512	-
84	Volume lower DHW heat exchanger	-0,0028	m ³
85	Specific heat capacity lower DHW heat exchanger	4,186	kJ/(kg·K)
86	Density fluid in lower DHW heat exchanger	992	kg/m ³
87	UA-value lower DHW heat exchanger	6037	kJ/(h·K)
88	Exponent depending on mass flow rate	0,3086	-
89	Exponent depending on temperature difference	0	-
90	Exponent depending on average temperature level	5,64E-05	-
95	Relative height inlet upper DHW heat exchanger	0,73	-
96	Relative height outlet upper DHW heat exchanger	0,972	-
97	Volume upper DHW heat exchanger	-0,0028	m ³
98	Specific heat capacity upper DHW heat exchanger	4,186	kJ/(kg·K)
99	Density fluid in upper DHW heat exchanger	992	kg/m ³
100	UA-value upper DHW heat exchanger	129,8	kJ/(h·K)
101	Exponent depending on mass flow rate	0,0878	-
102	Exponent depending on temperature difference	0	-
103	Exponent depending on average temperature level	0,9965	-
108	Relative height inlet flue gas from burner Type 210	0,58	-
109	Relative height outlet flue gas from burner Type 210	0,923	-
110	Volume of the flue gas heat exchanger	0,05	m ³
111	Specific heat capacity of the flue gas	1,2	kJ/(kg·K)
112	Density of the flue gas	1,2	kg/m ³
113	UA-value of the flue gas	2350	kJ/(h·K)
114	Exponent depending on mass flow rate	0,6	-
115	Exponent depending on temperature difference	0	-
116	Exponent depending on average temperature level	0	-
121	Accuracy for calculations of temperatures	0,001	-
122	Accuracy for calculations of UA-values	10	-
123	Accuracy for the mixing process	1·10 ⁷	-
125	Number of nodes	100	-