COMPARISON OF DIFFERENT APPROACHES FOR THE SIMULATION OF BOILERS USING OIL, GAS, PELLETS OR WOOD CHIPS

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ABSTRACT

A detailed model for the simulation of boilers using oil, gas, pellets or wood chips has been developed and compared with measurements. Approaches of different complexity for the simulation of steady state flue gas losses were tested. The more physical approaches are able to reproduce measured data better than the simpler empirical models, but they also require more model parameters to be determined and a higher simulation effort.

Cycling behaviour of the simple one-node thermal mass approach of the model was compared with measured cycling behaviour of a pellet boiler. With the proper values for the relevant boiler parameters, cycling behaviour is reproduced well.

With the implementation in a FORTRAN-dll that can be called from TRNSYS, a tool is now available that suits the needs of scientists as well as planners and product developers that use energy systems simulation tools.

INTRODUCTION

For energy estimating purposes and emissions simulation, sufficiently detailed boiler models are needed, that reflect the dependency of energetic efficiency and numbers of burner starts on the boundary conditions of the heating load and the boiler control.

The annual fuel consumption of boilers can be estimated by spreadsheet-methods (e.g. Paulsen 1999, CEN 2008). However, for heat load patterns that are more complex or for investigations of the effects of changes in hydraulics and control of the heating system, dynamic simulations are likely to be more convenient. Particularly in combination with solar thermal systems, changes in boiler efficiency induced by changes of the heat load pattern are not easily foreseeable without simulations (Haller et al. 2008, Haller & Konersmann 2008).

From the available literature on boiler simulation and measured field performance of boilers, the criteria for an adequate boiler model were deduced. In particular, the model should be able to:

- simulate boilers for different combustibles (oil, natural gas and biomass),
- distinguish between losses to flue gas and losses to the ambient (room of installation),
- calculate gains of vapour condensation from flue gas,
- reflect the effect of return temperature and power modulation on flue gas losses and condensation gains,
- simulate cooling out of the thermal mass of the boiler and estimate the number of times the burner has to start and stop,
- estimate auxiliary electricity consumption.

For the simulation software TRNSYS (Klein et al. 2004), several boiler models are available. These are e.g. TRNSYS standard Type 6 (Klein et al. 2004), TESS Type 751 (Thornton 2004), HVAC 1 Primary Toolkit (Bourdouxhe 1994), the boiler model from IEA ECBCS Annex 10 (Dachelet 1987), Type 370 (Koschak et al. 1998) or Type 210 (Persson et al. 2009). Most of them do either not include the simulation of a thermal mass, have limited possibilities to reflect the influence of power modulation on flue gas losses, or are restricted to one type of fuel only. Therefore, a new model was developed and compared with results from laboratory measurements on oil, natural gas and biomass boilers.

Table 1 Characteristics of the investigated boilers according to the manufacturers

Short	**P_{max} [kW]**	**V_{wat} [l]**	**m_{empty} [kg]**	**mod. [-]**	**cond. [-]**
Pel1	10	59	312	YES	NO
Pel3	40	158	846	YES	NO
Chp1	150	295	1972	YES	NO
Oil1	15	15	58	YES	YES
Oil2	12	35	107	NO	YES
Gas1	14	n.a.	39	YES	YES
Gas2	14	3.7	45	YES	YES

Pel: pellets; Chp: wood chips; Oil: fuel oil; Gas: natural gas

P_{max}: maximum heating power; V_{wat}: water volume; m_{empty}: empty mass; mod.: power modulation; cond.: condensing;

For the development of this model, as well as for the comparison of modelling results with measurements, several small to medium sized boiler units have been installed in different laboratories. Measurements have been performed in steady state operation in the range of power modulation as well as in cycling operation below the minimum modulation power. The investigated boilers were all constructed later than year 2000, and all oil and gas boilers investigated were condensing boilers. Other features are shown in Table 1.

BOILER MODEL

General simulation concept

The general simulation concept was deduced from ASHRAE (2005), and has been adapted in order to include the thermal mass of the boiler (see Figure 1). The implementation of the simulation model for TRNSYS is referred to as Type 869.

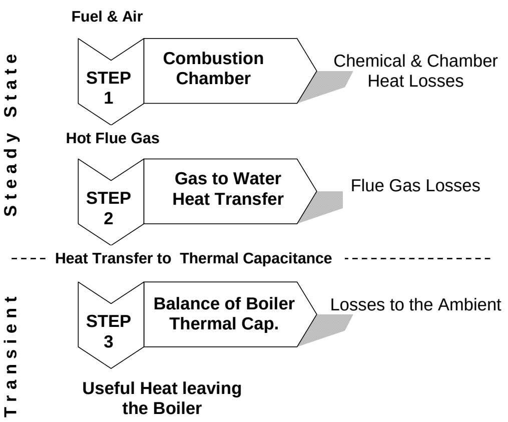

Figure 1: General concept of the boiler model.

In a first step, the combustion chamber is calculated. This includes the amount of air used for combustion and CO emissions in steady state. If necessary, it also includes the hot flue gas temperature after adiabatic reaction of the fuel with the combustion air. In a second step, the flue gas temperature and humidity after the flue gas to water heat exchanger is simulated. The flue gas losses and gains can be calculated based on these values. So far, all calculations are performed assuming steady state conditions. A time dependent, hence transient, calculation is only performed for the heat balance of the boiler's thermal mass (step 3). From this third step, losses from the heat exchanger to the ambient, useful heat leaving the boiler with the water flow, and the change of stored heat in the thermal capacitance of the boiler are obtained.

Excess air and CO emissions in steady state

For modulating boilers, excess-air for combustion and carbon monoxide concentrations in the flue gas are usually dependent on the power modulation. In Type 869, excess-air can be simulated as a quadratic function or a linear function of the burner power. Unless otherwise specified, the linear function has been used for all simulation results shown in this paper. Also carbon monoxide concentrations are simulated as linear functions of the burner power.

Losses from the combustion chamber

Heat losses of boilers have often been attributed to the temperature difference between the water body of the boiler and the ambient only. However, Konersmann et al. (2007) showed that the measured difference between flue gas losses and total losses also correlate with the combustion power for boiler Pel1. Thermographic pictures taken from boiler Pel3 showed that surface temperatures depend on both, the combustion power and the temperature of the boiler water. Therefore, two heat loss terms are taken into account in the model. One is the effective heat transfer area product for the heat exchange between the boiler water and the ambient, the other one is a constant fraction of the fuel energy being attributed to combustion chamber heat losses. A constant fraction is also used for chemical losses from the combustion chamber (unburned residues in ashes) that play a minor role for biomass boilers (0.1% and 0.2% have been calculated based on measurements for Pel3 and Chp1 respectively).

Flue gas losses in steady state

Flue gas losses are usually the predominant losses during steady state operation, especially for modern gas and oil boilers. Therefore, a very good model for the flue gas losses is needed. Three different approaches have been tested for the calculation of the temperature of the leaving flue gas in steady state operation. This temperature is needed for calculating sensible losses as well as latent gains.

Model 1: The empirical delta-T approach

This model is an extension of the model presented by Koschak et al. (1998). A dependency on the power modulation and on the boiler water mass flow rate has been added as shown in equations (1) and (2).

$$t_{fg,out} = t_{wat,in} + dT_{fg,out} \qquad \textbf{(1)}$$

$$dT_{fg,out} = dT_{nom} + ddT_{hxFg} \cdot \left(1 - \frac{\dot{Q}^{GHV}_{fuel}}{\dot{Q}^{GHV}_{fuel,nom}}\right) \cdot 100 \qquad \textbf{(2)}$$

$$+ ddT_{hxW} \cdot \left(1 - \frac{\dot{m}_{Wat}}{\dot{m}_{Wat,nom}}\right) \cdot 100$$

Model 2: The empirical effectiveness approach

In this approach, the flue gas to water heat transfer is modelled with an empirical heat transfer effectiveness. The effectiveness of the heat exchanger (≠ combustion efficiency) is assumed to be composed of a base effectiveness determined at nominal conditions and correction terms that account for changes in the mass flow rates of the flue gas or the boiler water. These corrections, similar to the corrections applied to the delta-T approach shown before, are shown in equations (3) and (4).

$$t_{fg,out} = t_{fg,hot} - \varepsilon_{HX} \cdot \left(t_{fg,hot} - t_{wat,in} \right) \quad \mathbf{(3)}$$

$$\varepsilon_{HX} = \varepsilon_{nom} + d\varepsilon_{hxFg} \cdot \left(1 - \frac{\dot{Q}^{GHV}_{fuel}}{\dot{Q}^{GHV}_{fuel,nom}} \right) + d\varepsilon_{hxW} \cdot \left(1 - \frac{\dot{m}_{Wat}}{\dot{m}_{Wat,nom}} \right) \quad \mathbf{(4)}$$

In this case, also the temperature of the hot flue gas before the heat exchanger $t_{fg,hot}$ has to be calculated. Assuming an adiabatic reaction, this temperature can be calculated based on mass and energy balances of the reaction.

Model 3: Detailed effectiveness-NTU approach

The detailed effectiveness-NTU approach is based on methods proposed by Lebrun et al. (1993) for large oil boilers. The method has been further developed based on suggestions by Lebrun (2007) and tests on the condensing gas boiler Gas1 (Dröscher 2008). The flue gas to water heat exchange is calculated with the effectiveness-NTU method. The overall heat transfer rate area product UA_{hx} of the heat exchanger is the inverse of an overall resistance R^*_{tot}, as shown in equations (5) and (6).

$$UA_{hx} = \frac{1}{R^*_{tot}} \quad \mathbf{(5)}$$

$$R^*_{tot} = \frac{R^*_{hxFg1}}{\left(\dot{m}_{fg} / [kg/h] \right)^{m_{hxFg}}} + \frac{R^*_{hxW1}}{\left(\dot{m}_{Wat} / [kg/h] \right)^{m_{hxW}}} \quad \mathbf{(6)}$$

For the effectiveness-NTU calculations, counterflow, parallel flow, and crossflow can be chosen. Best results have been achieved with counterflow for condensing boilers and cross flow for biomass boilers that have to avoid condensation and usually have a larger, to a large degree mixed, water body.

The effectiveness ε from the effectiveness-NTU relationship (Kays & London 1984) is applied to enthalpy differences instead of temperature differences for a dry heat exchanger. If condensation occurs, wet bulb temperature differences are taken as the driving force for the wet part of the heat exchanger. This approach is derived from the Merkel theory of evaporative cooling (Merkel 1925), and is also used for the calculation of dehumidifying coils (VDI 2006, ASHRAE 2000). Its application for the calculation of heat exchangers of condensing boilers has been proposed by Lebrun (2007).

If necessary, the heat transfer resistances for the wet part can be increased with the parameter fac_{wet}. The overall heat transfer resistance term $R^*_{tot,wet}$ of a fully wet heat exchanger is calculated as shown in equation (7).

$$R^*_{tot,wet} = fac_{wet} \cdot \frac{R^*_{hxFg1}}{\left(\dot{m}_{fg} / [kg/h] \right)^{m_{hxFg}}} + \frac{R^*_{hxW1}}{\left(\dot{m}_{Wat} / [kg/h] \right)^{m_{hxW}}} \quad \mathbf{(7)}$$

The overall heat transfer coefficient area product of a wet section and a dry section, respectively, are calculated according to equations (8) and (9), where $fr_{hx,wet}$ is the fraction of the effective heat exchanger surface that is considered to be wet.

$$UA_{hx,wet} = \frac{fr_{hx,wet}}{R^*_{tot,wet}} \quad \mathbf{(8)}$$

$$UA_{hx,dry} = \frac{1 - fr_{hx,wet}}{R^*_{tot}} \quad \mathbf{(9)}$$

Programming of an additional combustion air preheater has been necessary because measured flue gas temperatures of the investigated boiler Oil1 were lower than the return temperatures of the heated fluid. This additional counterflow flue gas to combustion air heat exchanger is also calculated with the effectiveness-NTU method. As the mass flow rates on both sides of this heat exchanger are almost equal, the simplified dependency of the UA value is based on the mass flow rate of the flue gas only:

$$UA_{gg} = UA_{gg1} \cdot \left(\dot{m}_{fg,wet} / [kg/h] \right)^{0.8} \quad \mathbf{(10)}$$

No splitting into dry and wet section is calculated for this heat exchanger.

Sensible losses, chemical losses, and latent gains

Excess air for combustion, temperature of the leaving flue gas and carbon monoxide concentration in the flue gas are known from the calculations presented above. Thus, sensible losses and chemical losses can be calculated based on these values and basic equations of mass conservation and thermodynamics (Baehr 2005).

Latent gains are sometimes calculated assuming saturation of the flue gas with water vapour whenever its temperature is below the dew point (Koschank et al. 1998; CEN 2008). However, literature of thermodynamics (Baehr 2005, p. 475) as well as own measurements indicate that undersaturation may well occur. Therefore, Type 869 calculates latent gains based on the known water load of the flue gas, the temperature at which the flue gas leaves the boiler and the assumption of a maximum relative humidity that may not be surpassed. Thus, the maximum water load of the flue gas $w_{fg,vap,\max}$ and the condensate mass flow $\dot{m}_{cond}$ are:

$$w_{fg,vap,\max} = R_{ratio} \cdot \frac{RH_{fg,\max} \cdot p_{sat,fg}}{p_{air} - RH_{fg,\max} \cdot p_{sat,fg}} \quad \mathbf{(11)}$$

$$\dot{m}_{cond} = MAX\left(w_{fg,hot} - w_{fg,vap,\max}; 0 \right) \cdot \dot{m}_{fg,dry} \quad \mathbf{(12)}$$

Transient behaviour (step 3)

Transient behaviour is assumed to be relevant only for the boiler's thermal capacitance (Figure 1). This is a simplification, since the transients from one steady burner operation to the next steady burner operation will also lead to transients in flue gas temperatures and condensation gains. However, assuming the new steady state efficiency too early will be compensated to some extent by assuming the next steady state efficiency too early as well. In the long run, these effects are expected to compensate each other.

To calculate the energy balance of the thermal capacitance of the boiler, the following simplifications are made:

- Thermal capacitance of the empty boiler and thermal capacitance of the water in the boiler are added and treated as one thermal node.
- The boiler's active thermal capacitance is simulated like a fully mixed water body. Thus, the mean temperature of the boiler's thermal capacitance equals the outlet temperature of the boiler.

To calculate $t_{wat,out}(\tau)$ after a certain time τ and $t_{wat,out,avg}$ of a time step with constant inlet temperature and mass flow, the following exponential equations are derived from the energy balance of the active thermal capacitance:

$$t_{wat,out}(\tau) = t_{\inf} - \left(t_{\inf} - t_{wat,out,A}\right) \cdot EXP\left(-G_1 \cdot \tau\right) \quad \textbf{(13)}$$

$$t_{wat,out,avg} = t_{\inf} + \frac{\left(t_{\inf} - t_{wat,out,A}\right) \cdot \left[EXP\left(-G_1 \cdot \tau\right) - 1\right]}{G_1 \cdot \tau} \quad \textbf{(14)}$$

Where $t_{\inf}$ is the temperature of the water outlet of the boiler after an infinite time (in steady state):

$$t_{\inf} = \frac{G_2}{G_1} \quad \textbf{(15)}$$

$$G_1 = \frac{UA_{hx-amb} + \dot{C}_{wat}}{C_{B,therm}} \quad \textbf{(16)}$$

$$G_2 = \frac{\left(\dot{Q}_{HXgw} + UA_{hx-amb} \cdot t_{amb} + t_{wat,in} \cdot \dot{C}_{wat}\right)}{C_{B,therm}} \quad \textbf{(17)}$$

$t_{wat,out,A}$ is the water outlet temperature at the beginning of the time step:

$$t_{wat,out,A} = t_{wat,out}\left(\tau = 0\right) \quad \textbf{(18)}$$

Start and stop procedures

Especially for biomass boilers, start and stop procedures may need special assumptions for burner power, auxiliary electricity demand, and carbon monoxide emissions. This is included in the simulation model by setting a fixed amount of time needed whenever the burner has to start, and setting different values for fuel consumption power, electricity demand and CO emissions during this time period.

COMPARISON WITH MEASUREMENTS

Air used for combustion and CO emissions in steady state

Examples of measurement results and fit functions for the amount of air used for combustion – represented by the λ-value – and carbon monoxide emissions in steady state are shown in Figure 2 and Figure 3. These linear fit functions are applied in all three models used to determine the flue gas losses that are discussed in the following section.

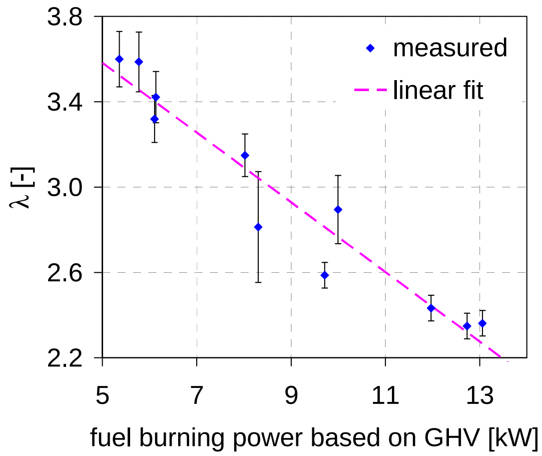

Figure 2: Dependency of the amount of air used for combustion – represented by λ - on burner modulation for the boiler Pel1

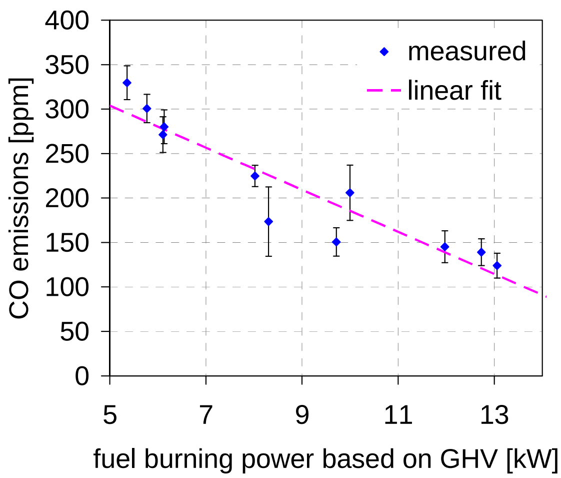

Figure 3: Dependency of carbon monoxide emissions on burner modulation for the boiler Pel1

Comparative steady state efficiency results

Figure 4 shows the comparison of the performance of five different models for the prediction of the combustion efficiency. Combustion efficiency has been defined as 100% minus sensible flue gas losses plus latent flue gas gains. All models used a linear fit for the dependency of lambda on the burner power. Models 1 to 3 are based on the empirical delta-T, the empirical effectiveness, and the detailed effectiveness-NTU approaches presented in section BOILER MODEL, respectively. Model 1a and 1b are based on the empirical delta-T approach without the correction for water mass flow (1a) and without the correction for the water mass flow and the modulation (1b), respectively. Apart from 1b, all approaches were able to reproduce the measured data with a RMSE in the range of 1% or lower, which is in the range of – or only slightly above – the estimated uncertainty of the measurements.

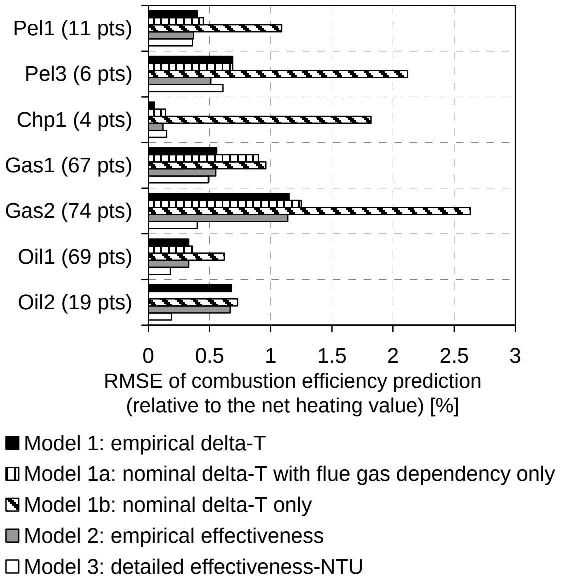

Figure 4: Comparison of the performance of five different models for the prediction of the combustion efficiency. RMSE = root mean square error, pts: number of measured efficiency points

Comparative cycling results

A test run with an average heating load of 0.87 kW has been performed for pellet boiler Pel1. Figure 5 compares measured inlet and outlet temperatures and the heating power with outlet temperature and heating power obtained with a simulation.

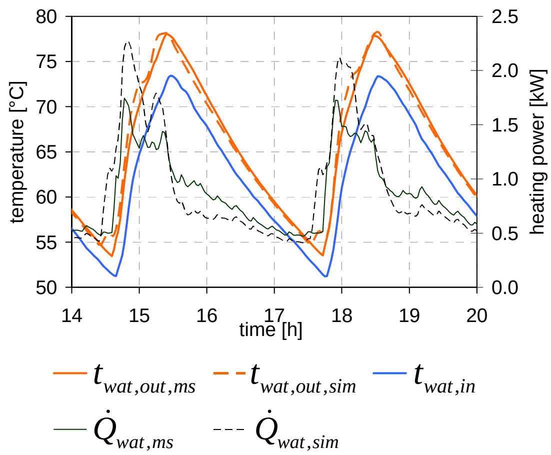

Figure 5: Simulated and measured outlet temperature and heating power of pellet boiler Pel1.

The simulation used the empirical delta-T approach for flue gas loss calculation. Measured values of the inlet temperature and the fuel consumption rate have been used as inputs for the simulation model.

Fuel consumption has been measured by weight loss of the pellet storage device. It is clearly visible that this weight loss leads to an almost instant increase in flow temperature and heating power for the simulation at time 14.5 h and 17.6 h. For the measurements of the real boiler, however, these increases are time-delayed as the burning process has to start before heat can be released. It can also be seen that the simulated thermal mass of the boiler reacts faster than the real thermal mass, since power increases faster and higher at startup and decreases faster at shutdown.

Figure 6 shows the cycling behaviour of a simulation where the same boiler parameters have been used as for the simulation shown in Figure 5, but fuel consumption and return temperature have now been simulated.

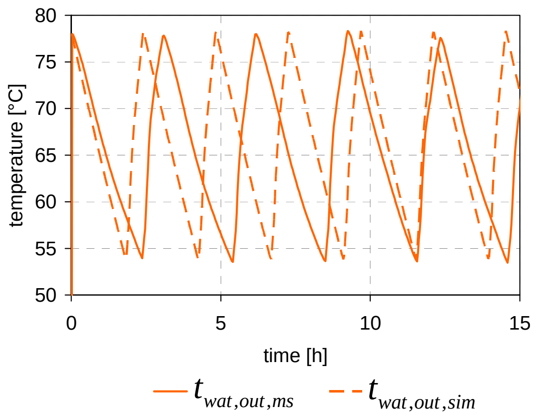

Figure 6: Simulated and measured cycling of pellet boiler Pel1 with boiler's thermal mass determined beforehand.

Fuel consumption has been simulated with the following assumptions based on the observed boiler characteristics:

- burner start when flow temperature decreases below 53.6 °C.
- 0.1 hours of startup time with an average fuel consumption power of 50% of the maximum power.
- After startup time, the boiler modulates between 53% and 100% to reach an outlet temperature of 75 °C.
- If 75 °C is exceeded at lowest possible power modulation, the burner turns off at 78.5 °C.
- the boiler's active thermal capacitance has been determined by measuring the heat on the water side while cooling the boiler from a uniform temperature of about 60 °C to ambient. The

burner was not running for a long time before this measurement.

Figure 6 shows that the simulated boiler cycles 25% more frequently than the real boiler. Figure 7 shows that the deficiency of too frequent cycling can be corrected by increasing the active thermal capacitance of the boiler from the value measured with the procedure described above to a value that is 28% higher. The average boiler efficiency η_B^{Hu} determined by this simulation is very close to the measured average boiler efficiency. Temporarily, the difference $\eta_{B,sim}^{Hu} - \eta_{B,ms}^{Hu}$ becomes large, mainly due to the time-delay between fuel consumption and heat release at startup for the measured boiler. With an increased number of cycles, these deviations even out and the difference between simulated and measured average boiler efficiency after 20 hours of cycling operation is in the range of +/-1%, not including electricity demand.

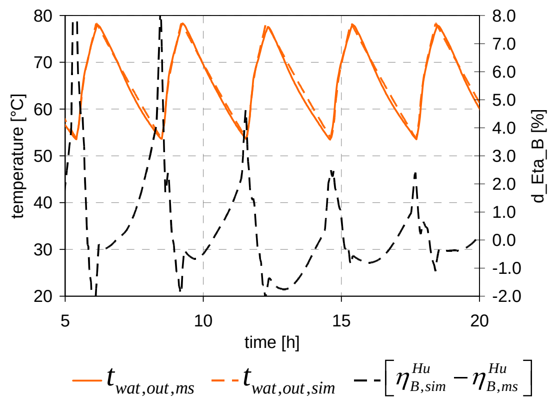

Figure 7: Measured and simulated cycling of pellet boiler Pel1 with boiler's thermal mass adapted to observed cycling behaviour: Comparison of outlet temperatures, power and average boiler efficiency.

DISCUSSION

The accuracy achieved with a simple delta-T approach for the flue gas temperature is in the range of measurement uncertainties, if the effect of power modulation on delta-T is included in the model. For simulations whose objective is an overall system approach, this model is convenient since only few parameters are needed, and these can be obtained from standard test results. However, this simple approach can not foresee e.g. the effect of lowering excess air for combustion on the leaving flue gas temperature. For this type of studies, the effectiveness-NTU approach might be the better choice.

The assumption that the flue gas is saturated with water vapor if it has been cooled below its dew point may be correct for some boilers, but not for others. Undersaturation has been observed for boilers Oil2 and Gas1 and is therefore also considered in the new boiler model.

For biomass boilers, heat losses to the ambient may not only be driven by the temperature of the boiler water, but also by the combustion power. A higher combustion power may lead to higher temperatures of the outer boiler surfaces, most likely due to heat losses of the combustion chamber. This effect is now implemented in the newly developed boiler model.

Based on current boiler testing standards, The distinction between the two different heat loss terms described above is difficult and sometimes even impossible to make. The effect of this distinction on the results of annual simulations is still to be investigated.

The boiler model presented here was able to reproduce boiler cycling in the case of a pellets boiler quite accurately if the appropriate thermal capacitance of the boiler was chosen. Also the boiler's thermal efficiency in cycling operation was reproduced within +/-1%. In addition to the steady state efficiency values, the described factors for heat losses to the ambient and thermal capacitance of the boiler were needed for this purpose.

For the simulation of boiler cycling, the effective thermal capacitance of the boiler plays a crucial role. For pellets and wood-chips boilers, this value may be significantly larger than the value obtained from the water content of the boiler alone. Therefore, well defined methods for the determination of the effective thermal capacitance should be developed and included into standard testing practices.

CONCLUSION

An extended and physically detailed boiler model has been developed for the simulation of oil, gas and biomass boilers for domestic applications. Several approaches of different complexity have been tested to simulate the steady state efficiency. The effectiveness-NTU approach with splitting of the heat exchanger into a dry part and a wet part was able to reproduce the measured values best, when the missing boiler parameters were fitted to the measurements. However, also simpler models were able to reproduce the measured values if the influence of modulation on the leaving flue gas temperature was taken into account.

It can be concluded that the quality and the quantity of available boiler performance data as well as the aim of the simulation task influence the choice of the boiler simulation approach decisively.

For accurate simulation of boiler cycling, data is needed that may currently not be derived from standard test procedures and have to be estimated based on manufacturer's data. Better simulation results could be achieved if standard tests would also include cycling on/off operation of the boiler's burner.

In summary, the newly developed model has been able to reproduce detailed measurements for oil, gas and biomass boilers in stationary and transient running modes. With the implementation in a TRNSYS-Type, a tool is now available that suits the needs of scientists as well as planners that use energy systems simulation tools.

ACKNOWLEDGEMENT

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NOMENCLATURE

Symbols

$C_{B,therm}$	active thermal capacitance, J/K
$\dot{C}$	capacity flow rate, W/K
ε	heat exchanger effectiveness, -
η	efficiency, -
fac_{wet}	factor for heat resistance of wet heat exchanger surfaces, -
$fr_{hx,wet}$	fraction of heat exchanger with wet surfaces, -
GHV	gross (or upper) heating value of the fuel, J/kg
λ	lambda value of combustion, i.e. the ratio of air used for combustion to the air that would be needed for a stoichiometric reaction of the fuel, -
m	exponent for the dependency of the heat transfer resistance on the mass flow, -
$\dot{m}$	mass flow, kg/s
p	pressure, Pa
R^*	overall resistance term of the gas water heat exchanger that includes heat exchanger area, K/W
R_{ratio}	ratio between molar mass of water and molar mass of dry gas, -
RMSE	Root Mean Square Error
RH	relative humidity, -
t	temperature, °C
τ	time, s
UA	overall heat transfer coefficient area product, W/K
w	water load of air or flue gas per kg dry gas, kg/kg

Subscripts

amb	ambient
B	boiler
avg	average
$cond$	condensate
dry	dry heat exchanger surface, dry flue gas
fg	flue gas
$fuel$	fuel
gg	(flue) gas to (combustion air) gas heat exchanger
hot	hot flue gas before heat exchanger (flue gas temperature after adiabatic combustion)
hx	flue gas to water heat exchanger
in	inlet (boiler return line)
inf	after an infinite time
ms	measured
nom	under nominal conditions (of water mass flow and burner power)
out	outlet (boiler flow line or outlet of flue gas)
sat	saturation (with water vapor)
sim	simulated
tot	total
vap	vapor
wat	water
wet	wet heat exchanger surface, wet flue gas
1	at (hypthetical) 1 kg/h mass flow rate

Superscripts

GHV	(based on) the gross (or upper) heating value
Hu	(based on) the net (or lower) heating value

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