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Model based control of a biomass grate furnace

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Abstract

Currently the full potential for low-emission operation of a biomass furnace at high efficiencies, which is in principle possible due to optimised furnace geometries as well as combustion air staging strategies, cannot be exploited since the plant control systems used just partly or even not at all consider the couplings and nonlinearities of the plants. For this reason a model based control strategy for biomass grate furnaces which allows an explicit consideration of the couplings and nonlinearities of the system was developed and experimentally verified at a pilot-scale furnace (moving grate furnace with a nominal boiler capacity of 180 kW). The paper will present the new process control as well as the results achieved by its application in comparison to conventional systems.

Keywords: biomass, grate combustion, modelling, model based control

1 Introduction

Grate combustion is a common and well developed technology for burning biomass fuels. But currently the full potential for low-emission operation of a biomass grate furnace at high efficiencies, which is in principle possible due to optimised furnace geometries as well as combustion air staging strategies, cannot be exploited since the plant control systems used just partly or even not at all consider the couplings and nonlinearities of the plants. State-of-the-art control methods for nonlinear multivariable systems (e.g. a biomass grate furnace), which consist almost exclusively of model based control strategies [9], would allow an explicit consideration of the couplings and nonlinearities of the system to be controlled. For this reason the objective of the work described in this paper was to develop a model based control strategy for biomass grate furnaces.

2 Methodology

A necessary prerequisite for all model based control strategies is a simple mathematical model of the system consisting of as few as possible low-order ordinary differential equations. This implies that the simple model describes reality more approximately than other models might do, but this does not matter since inaccuracies of the model as well as disturbances are compensated by the controller. Thus, in the first step appropriate models for all relevant parts of the furnace were developed and experimentally verified (section 4.1) on the basis of a pilot-scale furnace of the research centre BIOENERGY 2020+ (depicted in section 3). Subsequently the resulting sub-models were connected properly to an overall model for a biomass grate furnace which is suitable as a basis for model based control strategies (section 4.2). In the next step the resulting model was used to design an appropriate controller (section 4.3). Since all state variables are required by the designed controller but not all of them can be measured, an extended Kalman filter for their estimation had to be designed afterwards (section 4.4). Furthermore, the operating range of all actuators (e.g. fans, valves) is limited and therefore not all desired operating points are actually achievable. The limitation of the amplitude of an input variable can result in a very unpleasant operating behaviour and therefore a so called reference shaping filter has been designed (section 4.5) which avoids this problem a priori by modifying the set points to feasible values. Finally, the developed model based control was implemented and experimentally verified at the pilot-scale furnace (section 5).

3 Experimental setup

All experiments were conducted at a pilot-scale biomass grate furnace (Figure 1) of the research centre BIOENERGY 2020+. The pilot plant is a downscaled version (180 kW_{th}) of a typical medium-scale furnace in terms of geometry and instrumentation with a 2 duct fire tube hot water boiler (typical application for district heating). It is equipped with a screw conveyor, a horizontally moving grate (1 m long and 0.45 m wide), two primary air zones, a flue gas recirculation system above the grate and furthermore no furnace cooling takes place (refractory lining). The biofuel used is wood chips (mixture of oak, chestnut and pine) with an average moisture content of approximately 40 % (w.b.) and a typical composition for wood (carbon 49 % (d.b.), hydrogen 6 % (d.b.), oxygen 44 % (d.b.) and ash 1 % (d.b.)).

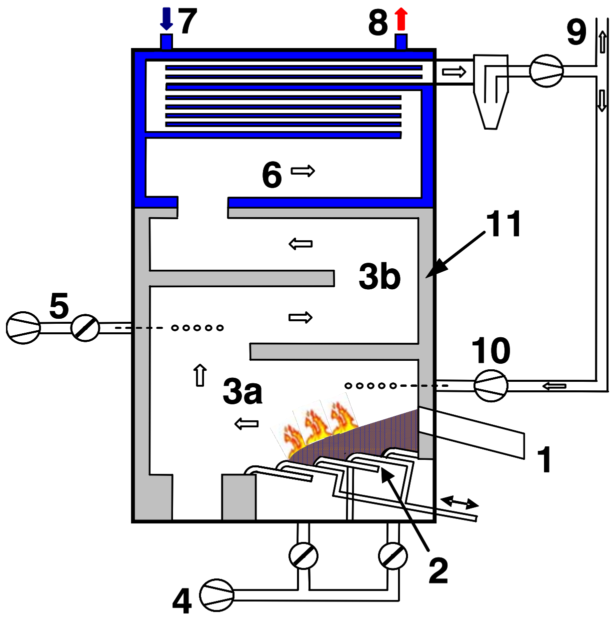

Figure 1: Pilot-scale furnace of the BIOENERGY 2020+: 1 ... fuel feed, 2 ... grate, 3a ... primary combustion chamber, 3b ... secondary combustion chamber, 4 ... primary air, 5 ... secondary air, 6 ... heat exchanger (boiler), 7 ... return, 8 ... feed 9 ... flue gas, 10 ... flue gas recirculation, 11 ... refractory lining

4 Model based control

4.1 Modelling

The modelling was done separately for the relevant parts of the biomass furnace and the boiler since the less complex models are better manageable. Thus, appropriate models for the fuel bed, the combustion of the dry fuel, the storage effect of the refractory lining, the heat exchanger (boiler) and the temperature sensor used in the secondary zone were developed and experimentally verified on the basis of the pilot-scale furnace depicted in section 3.
Additionally the interrelationship between the manipulable valve positions, fan and screw frequencies, the pressure conditions and the resulting volume and mass flows of fuel, primary air, secondary air and recirculated flue gas were modelled. Thereby, the fuel flow at the furnace inlet was assumed to be proportional

to the rotational speed of the screw conveyor [1, 5]. For the primary and secondary air supply as well as for the flue gas recirculation appropriate static models for the pressures and volume flows were developed respectively [2]. Thus, the valve positions as well as the fan and screw frequencies needed to supply the mass flows required by the actual controller can be determined by the particular models. Since these models are static and not used for the actual control unit design they are not discussed any further within this article but they are elaborately illustrated in [2].

4.1.1 Fuel bed

An extended literature review pointed out that among experts two completely different burning behaviours for grate combustion counter-current with ignition at the top of the bed and co-current with ignition at the bottom of the bed are assumed. Therefore, prior to modelling, the present combustion situation was determined. A detailed investigation of the fuel bed proved that the plant studied definetly shows co-current behaviour (ignition at the bottom of the bed) with biofuel and operation conditions as specified in section 3. Based on the insight gained into the basic combustion situation, a simple model for co-current combustion (ignition at the bottom of the bed) was derived [5].
Basically the model developed is based on the division of the fuel bed into a *dead zone* in which the fuel is heated up due to heat conduction along the grate bars and radiation from the flames and furnace walls above the bed but apart from that remains unchanged, a *water evaporation zone* in which most of the water evaporates and a *thermal decomposition zone* in which the devolatilisation and char burnout of the dry biomass takes place. Since the application of different air distributions between the two primary air zones had no effect due to insufficient sealing between the zones [2, 5], the two primary air zones were considered as one. Finally, the resulting mathematical model consists of only two ordinary differential equations, which represent the mass balances for the water in the water evaporation zone and the dry biomass in the thermal decomposition zone

$$\frac{dm_{\mathrm{w}}(t)}{dt} = -c_{\mathrm{wev}} m_{\mathrm{w}}(t)\,\alpha_{\mathrm{wev}}(t) + \frac{dm_{\mathrm{w,inlet}}(t - T_{\mathrm{d,fb}}(t))}{dt} \tag{1}$$

$$\frac{dm_{\mathrm{ds}}(t)}{dt} = -c_{\mathrm{thd}} m_{\mathrm{ds}}(t)\,\alpha_{\mathrm{thd}}(t)\left[\dot{m}_{\mathrm{pa}}(t) + \dot{m}_{\mathrm{pa,0}}\right] + \frac{dm_{\mathrm{ds,inlet}}(t - T_{\mathrm{d,fb}}(t))}{dt} \tag{2}$$

at time t, with the mass of water in the water evaporation zone m_{w}, the mass of dry fuel in the thermal decomposition zone m_{ds}, the experimentally determined, constant model parameters (c_{wev}, c_{thd}, $\dot{m}_{\mathrm{pa,0}}$), the experimentally determined, grate position depending factors (α_{wev}, α_{thd}), the primary air mass flow $\dot{m}_{\mathrm{pa}}$, the cumulating water mass $m_{\mathrm{w,inlet}}$ as well as the cumulating dry biomass $m_{\mathrm{ds,inlet}}$, which are introduced just for mathematical convenience and calculated from the water of the moist fuel flow $\dot{m}_{\mathrm{w,inlet}}$ and the dry biomass flow rate $\dot{m}_{\mathrm{ds,inlet}}$ at the fuel inlet respectively

$$m_{\mathrm{w,inlet}}(t) = \int_0^t \dot{m}_{\mathrm{w,inlet}}(\tau)\,d\tau \tag{3}$$

$$m_{\mathrm{ds,inlet}}(t) = \int_0^t \dot{m}_{\mathrm{ds,inlet}}(\tau)\,d\tau \tag{4}$$

and the dead time $T_{\mathrm{d,fb}}$

$$T_{\mathrm{d,fb}}(t) = c_{T_{\mathrm{d,fb}}} \frac{m_{\mathrm{w}}(t)}{\dot{m}_{\mathrm{ds,inlet}}(t)} \tag{5}$$

with the experimentally determined constant $c_{T_{\mathrm{d,fb}}}$.

4.1.2 Combustion

The model for the combustion is based on a common mass and energy balance for the evaporated water, the decomposed dry biomass, the supplied air and the recirculated flue gas what finally leads to the mass flow $\dot{m}_{\mathrm{fg}}$ as well as the chemical composition $\mathbf{x}_{\mathrm{fg}}$ of the flue gas and to the adiabatic combustion temperature T_{ad} [1, 6].

4.1.3 Storage effect of the refractory lining

To model the storage effect of the refractory lining a theoretical, averaged refractory lining temperature $T_{\rm rl}$ has been defined. In the end, the resulting model consists of only one ordinary differential equation and one algebraic equation representing the energy balance for the refractory lining

$$\begin{aligned} \frac{dT_{\rm rl}}{dt} &= a_{\rm rl}\left[T_{\rm ad} - T_{\rm rl}\right] \qquad (6) \\ T_{\rm fg,in} &= c_{\rm rl} T_{\rm rl} + \left[1 - c_{\rm rl}\right] T_{\rm ad} \qquad (7) \end{aligned}$$

with the time t, the adiabatic combustion temperature $T_{\rm ad}$, the flue gas temperature at the heat exchanger inlet $T_{\rm fg,in}$ and the experimentally determined model parameters ($a_{\rm rl}$, $c_{\rm rl}$) for whose general dependency on the flue gas mass flow different approaches were suggested [1, 6]. Since constant parameters were used for the actual control unit design this is not discussed further within this article.

4.1.4 Heat exchanger

The difference between the return temperature $T_{\rm ret}$ and the feed temperature $T_{\rm feed}$ of typically 20 °C is very small compared to the temperature spread of the flue gas of approximately 800 °C. Hence the water temperature can be assumed constant for the flue gas part which is why the modelling of the heat exchanger was done separately for the flue gas and the water part. Finally, the resulting model consists of one algebraic equation for the flue gas part and a first order ordinary differential equation for the water part [4]

$$\begin{aligned} \text{flue gas part:} \quad \dot{Q} &= c_{\rm he}\left[T_{\rm fg,in} - T_{\rm w}\right]^{q_{\rm he1}} \dot{m}_{\rm fg}^{q_{\rm he2}} \qquad (8) \\ \text{water part:} \quad \frac{dT_{\rm feed}}{dt} &= -\frac{\dot{m}_{\rm w}}{c_{\tau,\rm he}} T_{\rm feed} + \frac{\dot{m}_{\rm w}}{c_{\tau,\rm he}}\left[\frac{\dot{Q}}{\dot{m}_{\rm w} c_{\rm w}} + T_{\rm ret}\left(t - \frac{c_{\rm T_{d,he}}}{\dot{m}_{\rm w}}\right)\right] \qquad (9) \end{aligned}$$

with the transferred heat flow $\dot{Q}$, the experimentally determined, constant model parameters ($c_{\rm he}$, $q_{\rm he1}$, $q_{\rm he2}$, $c_{\tau,\rm he}$, $c_{\rm T_{d,he}}$), the flue gas temperature at the heat exchanger inlet $T_{\rm fg,in}$, the constant, averaged water temperature $T_{\rm w}$ for the flue gas part, the flue gas mass flow $\dot{m}_{\rm fg}$, the time t, the water mass flow $\dot{m}_{\rm w}$ and the specific heat capacity of water $c_{\rm w}$.

4.1.5 Temperature sensor

The temperature sensors used in biomass furnaces for continuous flue gas temperature measurement are in principle not able to measure the actual flue gas temperatures. For this reason mathematical models describing the relation between process values and measured temperatures were developed for the sensors used in the investigated plant [3]. Thereby, an appropriate model for the temperature sensor used at the end of the secondary zone corresponding to the flue gas temperature at the heat exchanger inlet $T_{\rm fg,in}$ was developed.

Since the temperature sensor is no direct part of the plant it was not used for the control unit design and the corresponding simulations in the first step [1]. But the time delay of the temperature sensor corresponding to its robust design would lead to an exceeding integration of the control deviation by the integrating controller and possibly even to input saturations (in particular of the flue gas recirculation) and the corresponding difficulties for the control [8]. However, the model developed for the temperature sensor [3] is adequate for the estimation of the real temperature but too complex to be used directly for the control unit design since it consists of a second order model for the sensor itself and a second order model for the surrounding refractory lining. But for the control unit design it is sufficient to model the time delay of the temperature sensor by a simple, first order differential equation

$$\frac{dT_{\rm se}}{dt} = \frac{1}{\tau_{\rm se}}\left[-T_{\rm se} + T_{\rm fg,in}\right] \qquad (10)$$

with the sensor temperature T_{se}, the time t, the experimentally determined, constant model parameter τ_{se} and the flue gas temperature at the heat exchanger inlet $T_{\mathrm{fg,in}}$.

4.2 Overall model

The connection of the sub-models developed (sections 4.1.1 up to 4.1.4) in their full complexity leads to an overall model which can be used to simulate the behaviour of the biomass furnace. For a model based control unit design the overall system should be preferably simple and depicted as a system of ordinary first order differential equations of the form

$$\frac{d\mathbf{x}}{dt} = \mathbf{f}\left(\mathbf{x}, \mathbf{u}\right) \tag{11}$$

with the state vector $\mathbf{x}$ and the input vector $\mathbf{u}$. For this reason partial further simplifications of the sub-models have been carried out and the state variables as well as the input variables have been chosen properly in particular to achieve a preferable simple overall model.

Thereby the dry biomass flow rate $\dot{m}_{\mathrm{ds,inlet}}$ at the fuel inlet was replaced by a slightly averaged dry biomass flow rate $\dot{\bar{m}}_{\mathrm{ds,inlet}}$ in the formula for the dead time in the fuel bed (5) to ensure that stepwise changes of the biomass flow rate do not affect the dead time immediately. In a further simplification the slightly averaged dry biomass flow rate $\dot{\bar{m}}_{\mathrm{ds,inlet}}\left(t\right)$ at time t was assumed to be equal to the actual dry biomass flow rate $\dot{m}_{\mathrm{ds,inlet}}\left(t - T_{\mathrm{d,fb}}\right)$ before the dead time. And additionally the factor α_{wev} describing the grate position dependency of the water evaporation as well as the the parameters (q_{he1}, q_{he2}) in the model for the heat exchanger have been equalised to one [1].

As state variables the mass of water in the water evaporation zone $x_1 = m_{\mathrm{w}}$, the mass of dry fuel in the thermal decomposition zone $x_2 = m_{\mathrm{ds}}$, the averaged refractory lining temperature $x_3 = T_{\mathrm{rl}}$, the feed temperature $x_4 = T_{\mathrm{feed}}$ and the temperature of the temperature sensor used at the end of the secondary zone $x_5 = T_{\mathrm{se}}$ were selected. The selected input variables are the dry biomass flow rate $u_1\left(t\right) = \dot{m}_{\mathrm{ds,inlet}}\left(t - T_{\mathrm{d,fb}}\left(t\right)\right)$ delayed by the dead time $T_{\mathrm{d,fb}}$, the primary air mass flow $u_2\left(t\right) = \alpha_{\mathrm{thd}}\left(t\right)\left[\dot{m}_{\mathrm{pa}}\left(t\right) + \dot{m}_{\mathrm{pa,0}}\right]$ with the model parameters ($\alpha_{\mathrm{thd}}\left(t\right)$, $\dot{m}_{\mathrm{pa,0}}$), the mass flow of the recirculated flue gas $u_3 = \dot{m}_{\mathrm{fgrec}}$ and the sum of primary and secondary air mass flow $u_4 = \dot{m}_{\mathrm{pa}} + \dot{m}_{\mathrm{sa}}$. The disturbance variables were defined as the specific enthalpy of the supplied air $d_2 = h_{\mathrm{air}}$, the specific enthalpy of the recirculated flue gas $d_3 = h_{\mathrm{fgrec}}$, the water mass flow $d_4 = \dot{m}_{\mathrm{w}}$, a delayed return temperature $d_5 = T_{\mathrm{ret}}\left(t - \frac{c_{T_{\mathrm{d,he}}}}{\dot{m}_{\mathrm{w}}}\right)$. Disturbance variable d_1 corresponds to the water content of the fuel $w_{\mathrm{H_2O,fuel}}$ delayed by the dead time $T_{\mathrm{d,fb}}$ and is calculated by $d_1\left(t\right) = \frac{w_{\mathrm{H_2O,fuel}}\left(t - T_{\mathrm{d,fb}}\left(t\right)\right)}{1 - w_{\mathrm{H_2O,fuel}}\left(t - T_{\mathrm{d,fb}}\left(t\right)\right)}$.

The acausal correlation in the definitions of the input variable u_1 and the disturbance variable d_1 at first seem to be problematic. However, the water content of the fuel $w_{\mathrm{H_2O,fuel}}$ cannot be influenced anyway, it would just be possible to calculate the disturbance variable d_1 in case the water content is known. On the other hand the dry biomass flow rate $\dot{m}_{\mathrm{ds,inlet}}\left(t - T_{\mathrm{d,fb}}\left(t\right)\right)$ has already been fed to the furnace at time $t - T_{\mathrm{d,fb}}$ when the corresponding input variable $u_1\left(t\right)$ is requested by the controller at time t. But short term load changes principally cannot be achieved by the fuel feed. Hence the fuel feed is just used to assure the long term load requests and for short term load changes the primary air is used.

Finally, the resulting overall model consists of five first order ordinary differential equations

$$\frac{dx_1}{dt} = -\frac{c_{11}}{1 + c_{12}d_1}x_1 + \frac{d_1}{1 + c_{12}d_1}u_1 \tag{12}$$

$$\frac{dx_2}{dt} = \frac{c_{21}}{1 + c_{12}d_1}x_1 - c_{22}x_2u_2 + \frac{1}{1 + c_{12}d_1}u_1 \tag{13}$$

$$\frac{dx_3}{dt} = \frac{-c_{31}x_1 + c_{32}x_2u_2 + c_{33}u_4 + c_{34}u_3}{c_{11}x_1 + c_{22}x_2u_2 + u_3 + u_4} - c_{35} \tag{14}$$

$$\frac{dx_4}{dt} = c_{41}x_1 + c_{42}x_2u_2 + c_{43}u_3 + c_{44}u_4 + c_{45}d_4d_5 - c_{45}d_4x_4 \tag{15}$$

$$\frac{dx_5}{dt} = -c_{91}x_5 + c_{92}\left[\frac{-c_{71}x_1 + c_{72}x_2u_2 + c_{73}u_3 + c_{74}u_4}{c_{11}x_1 + c_{22}x_2u_2 + u_3 + u_4} - c_{75}\right] \tag{16}$$

with the state variables (x_1 ... x_5), the input variables (u_1 ... u_4), the disturbance variables (d_1 ... d_5) and the parameters (c_{11} ... c_{92}) which are constant or just slowly varying in time and therefore assumed to be constant for the control unit design. Since the averaged temperature of the refractory lining $x_3 = T_{\text{rl}}$ changes very slowly compared to the remaining state variables (x_1, x_2, x_4, x_5) it was moved to the model parameters (c_{35}, c_{41}, c_{42}, c_{43}, c_{44}, c_{75}) for the control unit design.

4.3 Control unit design

First of all it is not reasonable to influence the temperature of the refractory lining directly by the control. Thus the corresponding state equation (14) can be neglected and the control process is described by the four differential equations (12), (13), (15) and (16) sufficiently. The oxygen content of the flue gas $x_{\text{O}_2,\text{fg}}$, the measured temperature in the secondary zone T_{se} and the feed temperature T_{feed} are essential operating parameters of a biomass furnace. Hence it is obvious to chose them as the first three (controlled) output variables

$$y_1 = x_{\text{O}_2,\text{fg}} = \frac{c_{51}x_2u_2+c_{52}u_4}{c_{53}x_1+c_{54}x_2u_2+c_{55}u_4} \qquad y_2 = T_{\text{se}} = x_5 \qquad y_3 = T_{\text{feed}} = x_4 \tag{17}$$

with the constant parameters (c_{51} ... c_{55}).
A frequently applied design method for coupled, nonlinear multivariable systems is the input-output linearisation [1, 9]. In doing so, first of all, the output variables need to be differentiated until every output variable is depending on at least one input variable. In this case only y_2 and y_3 need to be differentiated once

$$y_1 = g_1\left(x_1, x_2, u_2, u_4\right) \qquad \frac{dy_2}{dt} = g_{21}\left(x_1, x_2, x_5, u_2, u_3, u_4\right) \qquad \frac{dy_3}{dt} = g_{31}\left(x_1, x_2, x_4, u_2, u_3, u_4\right) \tag{18}$$

until the dependency on an input variable appears for every output variable. Thus the system has the relative degree $\rho = 2$ what would lead to a nonobservable second order sub-system since the overall system is of order $n = 4$ and additionally the resulting control law would not contain the input variable u_1. To overcome this problems the mass of dry fuel in the thermal decomposition zone has been chosen as a fourth (controlled) output variable $y_4 = x_2 = m_{\text{ds}}$. In doing so the relative degree is increased to $\rho = 3$, the resulting nonobservable first order system is stable and all four input variables can be set explicitly.
A reasonable set point for the fourth output variable r_4 results from a closer consideration of the systems equilibrium point and its requested averaged air ratio in the fuel bed for steady state operation $r_{\lambda_{\text{fb}}}$. Equalising the time derivative of the state variables in the state equations used for the control unit design, (12), (13), (15) and (16), to zero leads together with the output equations (17) to the input variables ($u_{1,\text{eq}}$, $u_{3,\text{eq}}$, $u_{4,\text{eq}}$) needed for the constant set points ($r_{1,\text{eq}}$, $r_{2,\text{eq}}$, $r_{3,\text{eq}}$) in the equilibrium state. The input variable corresponding to the primary air $u_{2,\text{eq}}$ needed in the equilibrium state to achieve the requested air ratio in the fuel bed $r_{\lambda_{\text{fb}}}$ is

$$u_{2,\text{eq}} = -\frac{c_{51}}{c_{22}c_{52}} r_{\lambda_{\text{fb}}} u_{1,\text{eq}} + \dot{m}_{\text{pa},0} \tag{19}$$

with the constants (c_{51}, c_{22}, c_{52}, $\dot{m}_{\text{pa},0}$). This leads to a reasonable set point for the fourth output variable r_4

$$r_4 = \frac{c_{52}\left[u_{2,\text{eq}} - \dot{m}_{\text{pa},0}\right]}{-c_{51} r_{\lambda_{\text{fb}}} u_{2,\text{eq}}} \tag{20}$$

which ensures the desired averaged air ratio in the fuel bed for the steady state.
In the end the concept of input-output linearisation led to an appropriate control law for the first three output variables (17) in which an additional integrator was used to improve the stationary accuracy [1]. The resulting correlation between the set points (r_1, r_2, r_3) and the corresponding output variables (y_1, y_2, y_3) is linear and decoupled according to the desired transfer functions

$$T_1\left(s\right) = \frac{\omega_{1,0}}{s+\omega_{1,0}} \qquad T_2\left(s\right) = \frac{\omega_{2,0}}{s^2+\omega_{2,1}s+\omega_{2,0}} \qquad T_3\left(s\right) = \frac{\omega_{3,0}}{s^2+\omega_{3,1}s+\omega_{3,0}} \tag{21}$$

with the coefficients ($\omega_{1,0}$ ... $\omega_{3,1}$).

However, this method is not applicable to the fourth output variable $y_4 = x_2 = m_{\mathrm{ds}}$ because of the acausal relation between the dry biomass flow rate and the corresponding input variable $u_1(t) = \dot{m}_{\mathrm{ds,inlet}}(t - T_{\mathrm{d,fb}}(t))$. But, as already mentioned before, the fuel feed is just used to assure the long term load requests and therefore set constant to the value needed for the steady state $u_{1,\mathrm{eq}}$ and additionally a proportional controller just interferes in case of a control deviation of the available dry fuel in the thermal decomposition zone $y_4 = x_2 = m_{\mathrm{ds}}$ [1].

4.4 Extended Kalman filter

The controller designed requires all state variables but not all of them can be measured. For this reason an extended Kalman filter [7] was developed to estimate the state variables statistically optimally with respect to the least mean squares of the estimation error. The Kalman filter is generally based on the mathematical model of the system and additionally corrects the state variables predicted with measurements.
In principle any measurement linearly related to the state variables can be used. However, as already mentioned the measurement of the flue gas temperature in the secondary zone is very problematic and therefore not suitable for the Kalman filter. Finally, only the oxygen content of the flue gas and the feed temperature have been used. Since the fifth state equation of the overall model (16) solely represents the storage effect of the temperature sensor used in the secondary zone the model for the Kalman filter reduces to the first four state equations, (12) ... (15), in which the averaged temperature of the refractory lining $x_3 = T_{\mathrm{rl}}$ must not be moved to the constant model parameters any more.
Furthermore, the (unknown) amount of leak air as well as the (unknown) fuel properties, water content and energy density, usually vary during operation time what needs to be considered explicitly in the Kalman filter. Because of the comparatively big time constants of the system the direct estimation of the water content does not work properly. However, an increase of the water content finally has the same impact on the operating behaviour of the plant as a simultaneous decrease of the leak air and the fuel energy density. Therefore it is sufficient to consider only these two variables, leak air and fuel energy density, in the Kalman filter. In doing so, not the leak air and the fuel energy density itself but their time derivatives are assumed to be random variables to avoid stepwise changes which would be unrealistic especially for the fuel energy density. This yields to two additional state variables, the leak air mass flow $x_6 = \dot{m}_{\mathrm{la}}$ and a fuel feed factor $x_7 = k_{\mathrm{ff}}$ with which the input variable corresponding to the fuel feed is multiplied $u_1 \to k_{\mathrm{ff}} u_1$.
The resulting nonlinear plant and measurement model

$$
\begin{aligned}
\frac{dx_1}{dt} &= -\frac{c_{11}}{1+c_{12}d_1}x_1 + \frac{d_1}{1+c_{12}d_1}x_7u_1 + w_1 \\
\frac{dx_2}{dt} &= \frac{c_{21}}{1+c_{12}d_1}x_1 - c_{22}x_2u_2 + \frac{1}{1+c_{12}d_1}x_7u_1 + w_2 \\
\frac{dx_3}{dt} &= \frac{-c_{31}x_1 + c_{32}x_2u_2 + c_{33}\left[u_4+x_6\right] + c_{34}u_3}{c_{11}x_1 + c_{22}x_2u_2 + u_3 + u_4 + x_6} - \left[c_{36}x_3 + c_{37}\right] + w_3 \\
\frac{dx_4}{dt} &= \left[c_{81}x_3 + c_{82}\right]x_1 + \left[c_{83}x_3 + c_{84}\right]x_2u_2 + \left[c_{85}x_3 + c_{86}\right]u_3 + \\
&\quad + \left[c_{85}x_3 + c_{87}\right]\left[u_4 + x_6\right] + c_{45}d_4d_5 - c_{45}d_4x_4 + w_4 \\
\frac{dx_6}{dt} &= w_5 \\
\frac{dx_7}{dt} &= w_6 \\
y_1 &= \frac{c_{51}x_2u_2 + c_{52}\left[u_4+x_6\right]}{c_{53}x_1 + c_{54}x_2u_2 + c_{55}\left[u_4+x_6\right]} + v_1 \\
y_3 &= x_4 + v_3
\end{aligned}
\tag{22}
$$

with the state variables (x_1 ... x_7), the input variables (u_1 ... u_4), the disturbance variables (d_1 ... d_5), the measured output variables (y_1, y_3), the constants (c_{11} ... c_{87}) as well as the zero mean, white plant (w_1 ... w_7) and measurement noise (v_1, v_3) respectively is linearised and discretised in every control step and subsequently the Kalman filter equations [7] are applied to the resulting discrete, linear model. This finally leads to the estimated state variables ($\hat{x}_1$ ... $\hat{x}_7$) which are used in the actual controller designed in section 4.3.

4.5 Reference shaping filter

Input saturations will occur in every typical biomass furnace during operation. This input saturations are not a problem for the fuel feed as long it is generally able to cover all fuel mass flows necessary for the desired loads since no integrating controller is applied (compare section 4.3). But a saturation of one of the other input variables (u_2 ... u_4) corresponding to the output variables (y_1 ... y_3) could result in a very inappropriate operating behaviour. An input saturation firstly would lead to a wind up of the integrator used to improve the stationary accuracy (compare section 4.3) and secondly the input variables applied would not decouple the three control variables any more. This could result in severe cross-couplings and is called the problem of directionality in MIMO systems.

In [8] a reference shaping filter that prevents the problem of directionality systematically is introduced. However, this approach is rather complex and limited to operating ranges where all desired set points are generally feasible for steady state operation. But in typical biomass furnaces it is necessary to run the plant in operating ranges where some set points are not achievable even for the steady state since mostly actuators with a poor controlling range are used. At partial load for instance, the desired oxygen content of the flue gas very often principally cannot be achieved because of input saturations. In this cases it is important to adjust the corresponding set points properly to avoid the phenomena previously described.

However, what action is reasonable in case of an input saturation strongly depends on the saturation effective. For example is it meaningful to increase the set point of the oxygen content if it is not possible to decrease the amount of air supplied. But we must not ever decrease the set point of the oxygen content if it is not possible to supply enough air since this would lead to exorbitant carbon monoxide emissions.

For this reason a static reference shaping filter was developed which is generally based on the calculation of the input variables ($u_{1,\mathrm{eq}}$, $u_{2,\mathrm{eq}}$, $u_{3,\mathrm{eq}}$, $u_{4,\mathrm{eq}}$) needed for the constant set points ($r_{1,\mathrm{eq}}$, $r_{2,\mathrm{eq}}$, $r_{3,\mathrm{eq}}$, $r_{\lambda fb,\mathrm{eq}}$) in the equilibrium state (compare section 4.3). Depending on the input limit which would be exceeded an appropriate modification of the set points is performed to avoid the foreseen input saturation a priori. In case of dynamic operating point changes and because of model inaccuracies input saturations still could happen sometimes. However, for this cases it is sufficient to avoid the wind up of the integrators. Therefore, the input and output variables are related in meaningful pairs (secondary air with oxygen content of the flue gas y_1, recirculated flue gas with measured temperature in the secondary zone y_2, primary air with feed temperature y_3) and in case of an input saturation the wind up of the corresponding integrator is avoided by keeping its output constant.

5 Experimental verification

Finally the developed model based control was implemented and experimentally verified at a pilot-scale furnace of the research centre BIOENERGY 2020+ (depicted in section 3). Different stepwise load changes as well as stepwise changes of the set point for the residual oxygen content of the flue gas and the temperature in the secondary zone were performed with the old and new control system.

As an example for all tests performed, the progress of the feed temperature after a stepwise increase of its set point from $r_3 = 85$ °C to $r_3 = 89$ °C at time $t = 0$, corresponding to a load change from 120 kW to 180 kW, is displayed in Figure 2 for the old and new control respectively. In doing so, the set points for the oxygen content of the flue gas $r_1 = 6$ vol.%w.b. (corresponding to a total air ratio of $\lambda_{\mathrm{tot}} = 1.5$), the temperature in the secondary zone $r_2 = 900$ °C and the averaged air ratio in the fuel bed $r_{\lambda_{\mathrm{fb}}} = 0.7$ remained constant and the fuel used was wood chips as specified in section 3.

It is obvious that the new control was able to perform the load change much quicker compared to the old control. In strict sense the new control did perform the load change within 4 minutes, which is approximately ten times faster than with the old control. Additionally to the much faster response, the new control is able to keep the steady control deviation in a much smaller range.

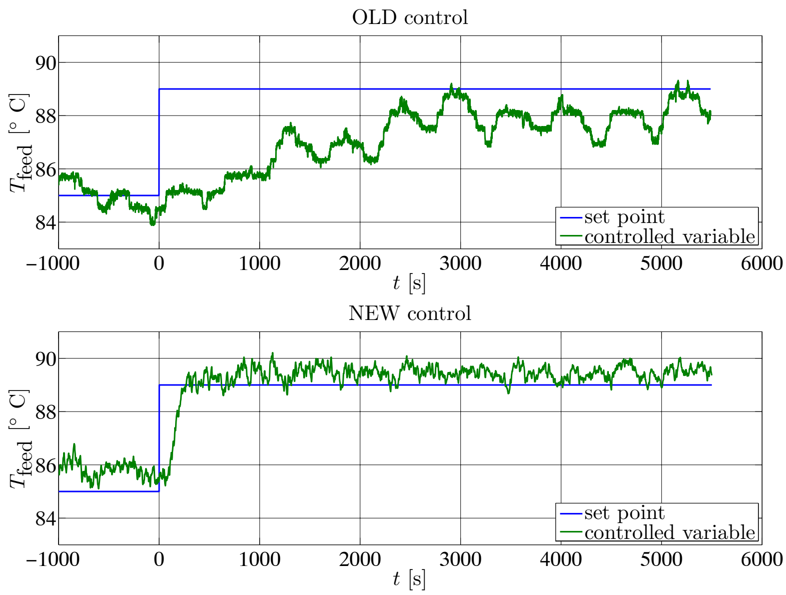

Figure 2: Feed temperature at a stepwise increase of the set point for the feed temperature

Simultaneously the new control was also able to hold the other control variables closer to their set point, what can be seen from the progress of the oxygen content of the flue gas during this experiment displayed in Figure 3. In particular the periodically appearing low oxygen contents during the operation with the old control were prevented by the new control satisfactorily. This leads to lower carbon monoxide emissions as well as the possibility to reduce the set point of the oxygen content and consequently to an increase of the boiler efficiency.

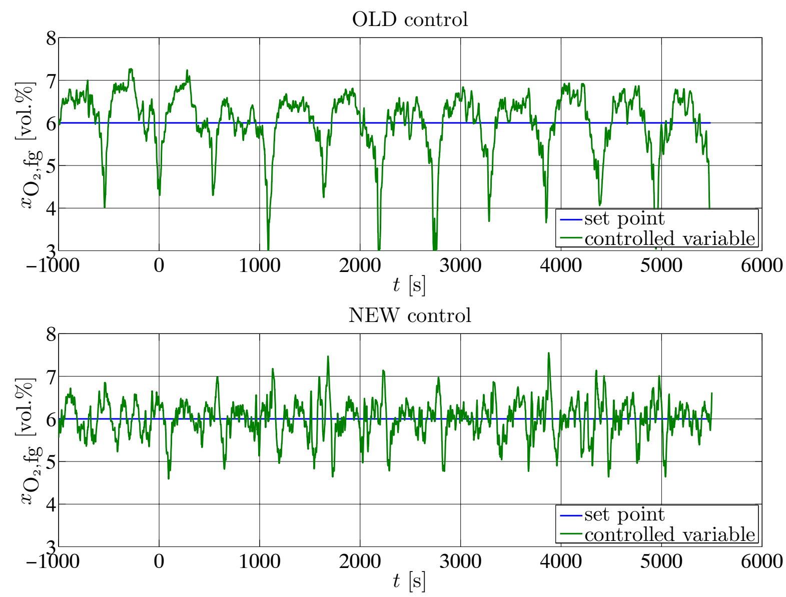

Figure 3: Oxygen content at a stepwise increase of the set point for the feed temperature

6 Summary

Within the work described in this paper for the first time ever a model based control strategy was developed and successfully implemented in a biomass grate furnace. In doing so, appropriate models for all relevant

parts of the furnace were developed and experimentally verified on the basis of a pilot-scale furnace of the research centre BIOENERGY 2020+ (moving grate furnace with a nominal boiler capacity of 180 kW). These models were connected to an overall model for a biomass grate furnace suitable as a basis for model based control strategies subsequently. The resulting overall model was used to design a model based control strategy including an extended Kalman filter to estimate the plant status and a reference shaping filter to avoid difficulties resulting from input saturations which is able to compensate the couplings and nonlinearities of the furnace explicitly.
Test runs performed with the new model based control showed a significant improvement of the plant operating behaviour. For instance, a load change from 120 kW to 180 kW was performed within 4 minutes, which is approximately ten times faster than with the old control. In addition, the control deviation of the other control variables was kept in a much smaller range. In particular the better control of the residual oxygen content of the flue gas leads to a significant reduction of the carbon monoxide emissions and additionally to the possibility to decrease the oxygen content set point and therefore to an increase of the boiler efficiency.

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