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Fuel-quality soft sensor using the dynamic superheater model for control strategy improvement of the BioPower 5 CHP plant

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Abstract

This paper presents an enhanced method for estimating fuel quality in a BioGrate combustion process and the method's use in control strategy improvement. This method is based on a dynamic model that makes use of combustion power estimates – which can be calculated based on the furnaces oxygen consumption – and that makes use of a nonlinear dynamic model of the secondary superheater. The paper focuses to estimate the most essential combustion parameters: fuel moisture and fuel flow. The time delay for detecting a change in fuel moisture and fuel flow is small enough for the method to be used for controlling both air and fuel feed, preventing any steam and pressure oscillations. The proposed control strategy is compared with the method currently used in the BioPower 5 CHP plant. Finally, the results are analyzed and discussed. © 2012 Elsevier Ltd. All rights reserved.

Keywords

BioPower, superheater, fuel quality, moisture, control, power plant

1 Introduction

Increased utilization of renewable energy has created new energy efficiency challenges in industry, where biomass is one of the most important raw materials for renewable energy. Grate firing is one of the main technologies that are currently used in bio-mass combustion for heat and power production [10, 15]. Though the grate firing of biomass has been tried and tested over many years, there are still some problems requiring further study, for instance, the conversion of biomass in the fuel bed, the mixing of burning substances in the fuel bed and on the freeboard, deposit formation and corrosion and their control, and pollutant formation and control [18]. One of the newest successful processes which use wood waste as fuel is a BioGrate boiler technology developed by MW Power. In this process, grate boilers are used as steam generation units to control a steam network. Rapid steam load changes necessitate good stability and load following properties in the system. Therefore, both drying and combustion in the grate must be controlled properly [8]. The main disturbances to the boiler are caused by fuel quality variations. The chemical properties of even the same type of biofuel may differ greatly, for example due to harvesting, storing, and transport conditions [18]. The ability to compensate variations in fuel quality thus plays a key role in controlling the combustion process.

An essential early step in developing a control strategy has been to develop a method for estimating a furnace's fuel flow and combustion power. As shown in the theoretical studies and practical tests by Kortela and Lautala [7] for a coal power plant, a furnace's fuel combustion power can be estimated on the basis of the measured oxygen consumption. On-line measurement of oxygen consumption was used when a new cascade compensation loop was built to optimally control the fuel flow. In that control strategy, the set point of the fuel feed mainly depends on the output of the drum pressure control. The amount of fuel burned is estimated using the flue gas oxygen content, and the fuel feed set point is modified accordingly. The control uses an integrator to remove steady-state offset in the control loop. It was reported that the amplitude

and the settling time of the response of the generator power decreased to about one third of the original when this cascade compensation loop was added to the present system.

Combustion power control (CPC) was implemented also in peat power plants [11], where it was able to stabilize the furnaces. The control actions of the burning air flow decreased when variations in the oxygen consumption were eliminated. The control strategy could thus reduce the standard deviation of the flue gas oxygen content, and the air flow could be lowered close to the optimal flow. As a consequence, the flue gas losses were reduced. Furthermore, the stabilized steam temperatures reduced thermal stress on superheaters and connected pipes. In addition the same approach has been applied to minimize (NO_x), (SO_2) and (CO) emissions in a bubbling fluidized bed boiler in [9, 12], where the use of the combustion power control algorithm made it possible to stabilize the burning conditions in co-combustion. This led to better control of flue gas emissions, and it was reported that the combustion power control reduced steam pressure deviation by 50%.

Model-based predictive control has been used by Havlena and Findejs [3] to enable tight dynamical coordination between air and fuel intake to take into account variations in power levels. The results showed that this approach can be used to increase boiler efficiency while considerably reducing NO_x emissions. Similar results have also been reported for the application of a multi-variable long-range predictive control (LRPC) strategy based on a local model network (LMN) in the simulation of a 200 MW oil-fired drum-boiler thermal plant [16].

However, there are still some challenges and unattained objectives in the development combustion power control. For example, variations in the moisture of fuel should be taken into account in order to correct any estimation errors of combustion power. The varying moisture content of the fuel results in uncertainty in the energy content of the fuel and complicates operation of the combustors. The typical procedure to determine moisture content of the fuel in small or medium-scale grate furnaces is to analyze manually collected samples of each fuel batch delivered to the plant. This method, however, is not accurate enough to predict moisture content of the fuel mix that enters the furnace. A change in moisture content of the fuel has to be detected at a resolution of seconds that the control system is able to make a correct response to the combustion air and the fuel feed system. There is, thus, a special need for a control system or for an operator to have information about moisture content for necessary adjustments of the combustors to be made.

A typical method for determining fuel moisture content is to determine first moisture content of the flue gas from which the moisture content can be then derived by a mass balance calculation [8]. The only delay of the measurement signal in this setup is the transport time of the gas from the furnace to the measurement position. This time delay can be measured in seconds and it opens up thus possibilities of controlling both combustion air and fuel feed.

The Fourier-transform infrared (FT-IR) technology is one method of determining the gas moisture content [1]. However, the accuracy of the FT-IR has been reported to be sensitive to the absolute temperature level, pressure, temperature gradients and particles carried with the gas, complicating measurements directly in the flue gas duct and this weakens its usability.

Another method for measuring the gas moisture content using a relative-humidity (RH) sensor was developed with the aim of improving the accuracy level of indirect determination of the moisture content of the fuel in a biomass furnace in [4]. The accurate implementation was achieved by cooling of an extracted flue gas stream, elevating the RH of the flue gases, before performing the measurement. The results of the tests showed that the method is able to detect variations in moisture content in seconds. However, in order to use this method, new devices, measurements and calibration are needed. There is thus a need for more cost efficient method for measuring fuel moisture, especially for a small scale boiler.

A fuel quality soft-sensor that utilizes combustion power and the energy balance of the boiler has been developed in [6]. However, the current study shows that fuel quality can be estimated both faster and more accurately when the energy balance of the boiler is replaced with the dynamic model of the secondary superheater. Based on the previous work of the authors, this paper sets forth to introduce a method to estimate fuel quality using combustion power and the nonlinear dynamic model of the secondary superheater. In this approach, only one model is needed. Furthermore, the estimation delay is decreased. The paper is organized as follows: Section 2 presents the BioPower 5 CHP plant process. Section 3 presents the enhanced control strategy, combustion power calculation for a BioGrate process and dynamic models of a boiler. The identification of the model of the secondary superheater is presented in Section 4. Section 5 describes the simulation and testing environment of the BioPower 5 CHP plant. The process experiments with varying fuel quality and the diagnosis results are presented in Section 6 and Section 7 in order to demonstrate the applicability of the method, followed by the conclusions in Section 8.

2 Description of the process and its control strategy

In the BioPower 5 CHP plant, the heat used for steam generation is obtained by burning solid biomass fuel: bark, sawdust, and pellets are fed into the steam boiler together with combustion air. As a result, combustion heat and flue gases are generated. The heat is then used in the steam-water circulatory process.

Fig. 1 shows the boiler part of the BioPower 5 CHP plant. The essential components of the water-steam circuit are an economizer, a drum, an evaporator, and superheaters. Feed water is pumped from a feed water tank into the boiler. The water is first led into the economizer (4) that is heated by flue gases. The temperature of the flue gases is decreased by the economizer, which also improves and thus further optimizes the efficiency of the boiler.

The heated feed water is led from the economizer to the drum (5) and through downcomers into the bottom of the evaporator (6) tubes that surround the boiler. From the evaporator tubes, the heated water and steam return back to the steam drum, where steam and water are separated. Steam rises to the top of the steam drum and flows to the superheaters (7). Steam heats up further until it superheats. The superheated high-pressure steam (8) is led into a steam turbine, where electricity is generated.

2.1 Fuel composition and fuel quality

The composition and the quality of fuel have a big effect on its heat value. Thus measurement of fuel quality plays a key role in designing the control strategy of a biopower plant and in guaranteeing its optimal operation. Common elements of all biomass fuels are carbon (C), hydrogen (H), oxygen (O), and nitrogen (N). In addition, biomass fuels contain substances from soil, such as water, minerals, rock materials, and sulphur (S).

The actual combustible components of fuels are carbon, hydrogen, and sulphur. Sulphur is an unwanted component because it forms harmful sulphur dioxide when it is burned. Nitrogen is also harmful; some of the nitrogen in fuel reacts with oxygen to form nitrogen oxides.

The evaporation of water found in fuel requires heat. As a result, moisture decreases the heat value of fuel. Table 1 lists the elemental composition and typical moisture content of wood fuels burned in the BioPower 5 CHP plant. The heat value of a fuel can be determined by means of the equation that has been derived by making use of the heat values of combustible components [2]. The effective heat value of a dry fuel is

$$q_{wf} = 0.348 \cdot w_c + 0.938 \cdot w_H + 0.105 \cdot w_S + 0.063 \cdot w_N - 0.108 \cdot w_O \text{ [MJ/kg]} \quad (1)$$

where w_c is the mass fraction of carbon in the fuel (%), w_H is the mass fraction of hydrogen in the fuel (%), w_S is the mass fraction of sulphur in the fuel (%), w_N is the mass fraction of nitrogen in the fuel (%), and w_O the mass fraction of oxygen in the fuel (%). The effective heat value of a wet fuel is obtained from the equation

$$q_f = q_{wf} \cdot \left(1 - \frac{w}{100}\right) - 0.0244 \cdot w \text{ [MJ/kg]} \quad (2)$$

where w is the moisture content of the wet fuel (%). In order to use Eq. (1), the composition of the fuel has to be known.

Fig. 1. 1. Fuel, 2. Primary air, 3. Secondary air, 4. Economizer, 5. Drum, 6. Evaporator, 7. Superheaters, 8. Superheated steam.

Table 1

The composition of wood fuels burned in the BioPower 5 CHP plant.

Fuel	Dry content (%)					Moisture (%)
	w_c	w_H	w_O	w_N	Ash	
Pine	54.5	5.9	37.6	0.3	1.7	60
Spruce	50.6	5.9	40.2	0.5	2.8	60
Wood mix	50.4	6.2	42.5	0.5	0.4	50

2.2 Control strategy of the biograte process

The main aim of the control strategy of the biograte process is to produce a desired amount of energy by keeping the drum pressure constant. At the highest level of the control strategy, the drum pressure control defines the power of the boiler. At a lower level, the necessary boiler power is produced by controlling the amount of combustion air and fuel as illustrated in Fig. 2.

The primary air flow is controlled by the set point that comes from the pressure control. The fuel feed is controlled to track the primary air flow measurement. The necessary amount of primary air and secondary air for diverse fuel and power levels are specified by air curves that have been calculated in the boiler design phase. The flue gas oxygen controller acts as a master controller while the set point of the secondary air controller is adjusted to provide the desired amount of excess air for the combustion.

3 Enhanced control strategy for the BioPower 5 CHP plant

In the enhanced control strategy (taking into account the oxygen consumption and total air feed), the amount of fuel burned is estimated and the fuel feed set point is modified accordingly. The integrator in Fig. 3 removes steady-state offset in the control loop. In addition, the enhanced control strategy uses the dynamic model of the secondary superheater to take into account variations in the moisture content of the fuel. This makes it possible to control the combustion process dynamically, preventing oscillations in steam temperature and pressure.

The biograte process is characterized by large time constants and long time delays. Therefore the drum pressure control has to be tuned slow to maintain stability. Disturbances in fuel quality and fuel feed have a strong effect on steam pressure and thus exhibit a proportionate direct correlation. These pressure and temperature disturbances settle slowly. To overcome this limitation, a dynamic model has been developed using the combustion power and the model of the secondary superheater. Furthermore, the following section describes heat transfer coefficients for convection, radiation and boiling heat transfer in detail.

3.1 Combustion power estimation

The combustion reaction in fossil fuel power plants occurs mainly between carbon and oxygen. Oxygen consumption is therefore a good measure of heat generation in the furnace [8].

When the fuel composition and combustion reactions are known, the combustion air and the composition of flue gases can be calculated. This information can then be used to conclude the completeness of the combustion and the correctness of the fuel– air ratio. Table 2 gives moles per unit of fuel from mass fractions of the fuel. The amount of oxygen needed for fuel combustion can be determined from the reaction equations. By summing up the oxygen needed for different components and subtracting the amount of oxygen in the fuel, the theoretical amount of oxygen needed to burn one kilogram of fuel completely is

$$N_{O_2}^g = n_C 00.5 \cdot n_{H_2} + n_S - n_{O_2} \text{ [mol/kg]} \quad (3)$$

Air contains mainly oxygen and nitrogen. Argon is often included in the nitrogen portion, so there is 21% oxygen and 79% nitrogen in the air. Theoretically, the corresponding amount of dry air needed is thus

$$N_{Air} = N_{O_2}^g \cdot \frac{1}{0.21} = N_{O_2}^g \cdot 4.76 \text{ [mol/kg]} \quad (4)$$

The flue gases include, in addition to combustion products, the nitrogen N that is contained in the air. Flue gases contain 3.76 times more nitrogen than the oxygen necessary for complete combustion. Incombustible components, for example, water are included in the equation unchanged. The flue gas flow for one kilogram of fuel is

$$N_{fg} = n_C + n_{H_2} + n_S + 3.75 \cdot N_{O_2}^g + n_{N_2} + n_{H_2O} \text{ [mol/kg]} \quad (5)$$

Similarly, the flue gas losses per kilogram of fuel are

$$q_{fg}^g = (n_C C_{CO_2} + n_S C_{SO_2} + (n_{H_2} + n_{H_2O}) C_{H_2O} + (3.75 \cdot N_{O_2}^g + n_{N_2}) C_N + (\frac{F_{Air}}{22.41 \cdot 10^{-3} \cdot m_{gf}}) - 4.76 \cdot N_{O_2}^g) C_{Air} \cdot (T_{fg} - T_0) \text{ [J/kg]} \quad (6)$$

where C_i is the specific heat capacity of the component i (J/molT), F_{Air} is total air flow (m³/s), m_{gf} is fuel flow (kg/s), T_{fg} is the temperature of the flue gas (°C), and T_0 the reference temperature (°C).

The combustion power of the BioGrate boiler is estimated using Eqs. (7)–(11).

The total oxygen consumption is

$$n_{O_2}^{tot} = 0.21 \cdot n_{Air} - \frac{X_{O_2}}{100} \cdot n_{fg} \text{ [mol/s]} \quad (7)$$

Where $n_{O_2}^{tot}$ is total oxygen consumption (mol/s), n_{Air} is total air flow (mol/s), $X_{O_2}(t+\tau)$ is the oxygen content of the flue gas (%), and n_{fg} the flue gas flow (mol/s).

Fig. 2. Current control strategy of the biograte process.

Table 2.

Moles of the components of the fuel per unit mass.

Comp.	Mass fraction (%)	M_i (g/mol)	n_i (mol/kg)
C	$w_C(1 - w/100)$	12.011	$w_C(1 - w/100)10/M_C$
H	$w_H(1 - w/100)$	2.0158	$w_H(1 - w/100)10/M_H$
S	$w_S(1 - w/100)$	32.06	$w_S(1 - w/100)10/M_S$
O	$w_O(1 - w/100)$	31.9988	$w_O(1 - w/100)10/M_O$
N	$w_N(1 - w/100)$	28.01348	$w_N(1 - w/100)10/M_N$
Water	w	18.0152	$10/M_w$

Fig. 3. Enhanced strategy based on oxygen consumption and the dynamic model of the secondary superheater.

The flue gas flow is

$$n_{fg} = m_{gf} \cdot N_{fg} + n_{Air} - 4.76 \cdot m_{gf} \cdot N_{O_2}^g \text{ [mol/s]} \quad (8)$$

On the other hand, the oxygen consumption can be presented in the form:

$$n_{O_2}^{tot} = m_{gf} \cdot N_{O_2}^g \text{ [mol/s]} \quad (9)$$

$$m_{gf} = \frac{(0.21 - \frac{X_0}{100})n_{Air}}{N_{O_2}^g + \frac{X_0}{100}(N_{fg} - 4.76 \cdot N_{O_2}^g)} \text{ [kg/s]} \quad (10)$$

The denominator of Eq. (10) is the amount of oxygen theoretically needed to burn one kilogram of fuel completely, added with the oxygen content in flue gas flow. It would be ideal to minimize the excess oxygen.

Finally, the net combustion power for a given fuel flow is

$$P = (q_f - q_{fg}^g - q_{cr}) \cdot m_{gf} \text{ [MW]} \quad (11)$$

Where q_{cr} is convection and radiation losses (MJ/kg).

3.2 Nonlinear dynamic models of the boiler and moisture estimation

Much of the behaviour of the boiler is captured by global mass and energy balances [19]. The heat released by the combustion of fuel is transferred to the water and steam of the boiler where each section can be considered as a thermal system. In this paper fuel quality is estimated using the dynamic model of the secondary superheater. The models of the economizer, the evaporator and the secondary superheater were developed for a simulation purpose.

The global mass balance is

$$\frac{d}{dt}(Q_s V_s + Q_w V_w) = m_f - m_s \text{ [kg/s]} \quad (12)$$

where Q_s is the specific density of the steam (kg/m³), V_s is the volume of the steam (m³), Q_w is the specific density of the water (kg/m³), V_w is the volume of the water (m³), m_f is the feed water flow (kg/s), and m_s the steam flow rate (kg/s).

The global energy balance is

$$\frac{d}{dt}(Q_s u_s V_s + Q_w u_w V_w + m_t c_p T_m) = Q + m_f h_f - m_s h_s \text{ [MJ/s]} \quad (13)$$

where u_s is the specific internal energy of the steam (MJ/kg), u_w is the specific internal energy of the water (MJ/kg), m_t is the total mass of the metal tubes and the drum (kg), c_p is the specific heat of the metal (MJ/kg K), T_m is the temperature of the metal (K), Q is the heat transfer from the flue gas to the steam/water (MJ/s), h_s is the specific enthalpy of the steam (MJ/kg), and h_f the specific enthalpy of the feed water (MJ/kg).

Since the internal energy is $u = h - p/Q$, the global energy balance is

$$\frac{d}{dt}(Q_s h_s V_s + Q_w h_w V_w + p V_t + m_t c_p T_m) = Q + m_f h_f - m_s h_s \text{ [MJ/s]} \quad (14)$$

where h_w is the specific enthalpy of the water (MJ/kg).

$$V = V_s + V_w \text{ [m}^3\text{]} \quad (15)$$

Multiplying Eq. (12) by h_w and subtracting the result from Eq. (14) gives

$$(h_s - h_w) \frac{d}{dt}(Q_s V_s) + Q_s V_s \frac{dh_s}{dt} + Q_w V_w \frac{dh_w}{dt} - V_t \frac{dp}{dt} + m_t c_p \frac{dT_m}{dt} = Q - m_f (h_w - h_f) - m_s (h_s - h_w) \left[\frac{\text{MJ}}{\text{kg}} \right] \quad (16)$$

Eq. (16) is used for the economizer, evaporator, and superheater subsections of the boiler and it is modified as necessary.

The energy balance of the boiler section and the temperature of the metal walls are considered separately for better model accuracy. If the drum level is kept at a constant set point, the variations in the steam volume are small and the energy balance for a subsection is [13,14]

$$\frac{dh_s}{dt} = \frac{1}{QV} (Q_t + m_f h_f - m_s h_s) \left[\frac{\text{MJ}}{\text{s kg}} \right] \quad (17)$$

where Q_t is the heat transfer from the metal walls to the steam/water (MJ/s). The energy balance for the tube walls is

$$\frac{dT_m}{dt} = \frac{1}{m_t c_p} (Q_m - Q_t) \text{ [K/s]} \quad (18)$$

where Q_m is the heat transfer from the flue gas to the metal walls (MJ/s). The convection heat transfer (superheaters) from the metal walls to the steam/water is

$$Q_t = \alpha_c h_c A_c (T_m - T) \text{ [MJ/s]} \quad (19)$$

where α_c is the correction coefficient, h_c is the convection heat transfer coefficient (W/m² K), and A_c is the heat transfer area (m²). The corresponding equation for boiling heat transfer (water wall) is

$$Q_t = \alpha_b h_b A_b (T_m - T)^3 \text{ [MJ/s]} \quad (20)$$

Where α_b is the correction coefficient, h_b is the boiling heat transfer coefficient (W/m² K), and A_b is the heat transfer area (m²).

$$T = (T_1 + T_2)/2 \text{ [}^\circ\text{C]} \quad (21)$$

where T_1 is the input steam/water temperature (°C), and T_2 the output steam/water temperature (°C).

The heat transfer from the flue gas to the metal walls for mixed convection and radiation heat transfer is

$$Q_m = \alpha_m h_{fg} (T_{fg} - T_m) + k_m (T_{fg}^4 - T_m^4) \quad (22)$$

where α_m is the correction coefficient, h_{fg} is the convection heat transfer coefficient (W/m² K), and k_m is the radiation heat transfer coefficient.

$$T_{fg} = \frac{q_{gf} + c_{p,air} \cdot T_{air} \cdot m_{air} \cdot \lambda}{c_{p,fg} \cdot [m_{fg} + (\lambda - 1)m_{air}]} \text{ [}^\circ\text{C]} \quad (23)$$

where $c_{p,air}$ is the specific heat capacity of air (J/kg K), T_{air} is the air temperature (°C), m_{air} is the theoretically necessary amount of air for combusting one kilogram of fuel (kg/kg), λ is the excess air coefficient, $c_{p,fg}$ is the specific heat capacity of flue gas (J/kg K), and m_{fg} is the flue gas flow for one kilogram of fuel (kg/kg).

The temperature of the secondary superheater is kept constant at a specific temperature. De-superheating spray is used to achieve mixing between the superheated steam at the outlet of the preceding component.

Because the attemperator has a relatively small volume, the mass storage inside it is negligible.

The steady state energy balance yields

$$m_{in} h_{in} + m_{ds} h_{ds} = m_{out} h_{out} \text{ [MJ/s]} \quad (24)$$

In normal operation, the steam flow m_{out} in the secondary superheater is regulated by the load controller, the enthalpy of the primary superheater h_{in} is determined by the upstream superheater and the enthalpy of the de-superheating spray h_{ds} is nearly constant.

3.3 Fuel quality soft sensor

The method proposed in this paper assumes that changes in fuel moisture directly affect the enthalpy of the secondary superheater; the effective heat value q_{gf} of Eq. (2) decreases linearly when fuel moisture increases. As a result, the flue gas temperature T_{fg} in Eq. (23) and thus the enthalpy of the secondary superheater also decrease. Therefore, the value for the moisture parameter w that dynamically affects the fuel quality is obtained by minimizing

$$\min J(w) = \sum_{i=0}^N |h - \hat{h}| \quad (25)$$

where N is the prediction horizon, h is the measured output enthalpy of the secondary superheater (MJ/kg), and $\hat{h}$ is the estimated output enthalpy of the secondary superheater (MJ/kg).

3.4 Dimensionless parameters

Dimensionless parameters are used to model the nonlinear heat transfer coefficients of the economizer, evaporator, and superheaters models.

The boundary layer equations are normalized by first defining the dimensionless independent variables of the forms [5]

$$y^* \equiv \frac{y}{L} \quad (26)$$

and

$$x^* \equiv \frac{x}{L} \quad (27)$$

where L is a characteristic length of the surface of interest. Moreover, dependent dimensionless variables may also be defined as

$$u^* \equiv \frac{u}{V} \quad (28)$$

and

$$v^* \equiv \frac{v}{V} \quad (29)$$

where V is the velocity upstream of the surface, and as

$$T^* \equiv \frac{T - T_s}{T_\infty - T_s} \quad (30)$$

where T_s is the surface temperature (K), and T_∞ the free stream temperature (K). The dimensionless similarity parameters Reynold's number, Re_L , and the Prandtl number, Pr , are used to apply results obtained for a surface experiencing one set of convective conditions to geometrically similar surfaces experiencing entirely different conditions. As long as the similarity parameters and dimensionless boundary conditions are the same for the two sets of conditions, the solutions of the differential equations for the nondimensional velocity and the temperature will also be the same. The relationship between the convection coefficient and dimensionless parameters can be described by the Nusselt number.

$$Nu \equiv \frac{hL}{k_f} = + \frac{\partial T^*}{\partial y^*} \quad (31)$$

where h is the local convection coefficient (W/m² K), and k_f is the thermal conductivity (W/m K). The Nusselt number is equal to the dimensionless temperature gradient at the surface. Furthermore, it provides a measure of the convection heat transfer occurring at the surface. For a prescribed geometry,

$$Nu = f(x^*, Re_L, Pr) \quad (32)$$

The Nusselt number must be some universal function of x^* , Re_L , and Pr . When this function is known, it can be used to compute the value of Nu for different fluids and for different values of V and L . The local convection coefficient h can be calculated based on the value of Nu .

3.4.1 Turbulent flow in circular tubes

The equation for computing the local Nusselt number for (hydrodynamically and thermally) fully developed turbulent flow can be obtained by means of the Dittus-Boelter equation [5,17]

$$Nu_D = 0.023 Re_D^{4/5} Pr^{0.4} \quad (33)$$

This equation describes the convection heat transfer coefficient h_c in Eq. (19).

3.4.2 Flow across tubes banks

Airflow across tube bundles exhibits a correlation of the form [5]

$$Nu_D = 1.13 C_1 Re_{D,max}^m Pr^{1/3} \quad (34)$$

where C_1 and m are constants for a tube bank of 10 or more rows. This equation describes the convection heat transfer coefficient h_{fg} in Eq. (22).

3.4.3 Boiling

The correlation for nucleate boiling is [5]

$$q_s'' = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma} \right]^{1/2} \left(\frac{c_{p,l} \Delta T_e}{C_{s,f} h_{fg} Pr_l^n} \right)^3 \quad (35)$$

where μ_l is viscosity of water (N s/m²), g is gravitational acceleration (m/s²), ρ_l is density of water (kg/m³), ρ_v is density of vapor (kg/m³), σ is surface tension, (N/m), and $c_{p,l}$ is specific heat of water (kJ/kg K). Coefficients $C_{s,f}$ and the exponent n depend on the solid liquid combination.

4 Identification of nonlinear model of the secondary superheater

The identification of the models of the superheaters, the evaporator, and the economizer was performed using the measurement data of the BioPower 5 CHP plant. The model of the BioPower 5 CHP plant's secondary superheater is presented here.

The input variables, steam inlet temperature, steam mass flow, and steam pressure were taken from the process measurement. The flue gas flow and flue gas temperature were calculated according to Eqs. (8) and (23).

The aim of the identification is to determine the model parameters V , m_t , α_c , α_b , α_m , and k_m .

Fig. 4 shows the secondary superheater's estimated and measured outlet steam enthalpy based on the measured data. The inputs, steam inlet temperature, steam mass flow, etc. are illustrated in Fig. 5. The

validation of the identified model was performed on another measurement series. The model performance is presented in Fig. 6. The inputs of the model are given in Fig. 7.

The identified models also work well using the validation data series. The measurement data was compressed. Therefore, the estimated values and the measurements are not exactly the same.

5 Description of the simulation and testing environment

To test the enhanced control strategy, a simulation model of the BioPower 5 CHP plant was built in the MATLAB Simulink environment. The entire model consists of the models of the subprocesses which have been programmed as a function in MATLAB, as well as a function block in Simulink. The blocks in Fig. 8 of the supply processes are used to describe the dynamical characteristics of the actuators and the primary and the secondary air fans and fuel feeder. The input variables for the grate are defined as primary air and secondary air flows and fuel flows. The output variables are flue gas temperature, flue gas flow, and flue gas oxygen content, and they are modelled as described in Chapter 3.1.

Fig. 4. The measured (dashed line) and estimated (solid line) outlet steam enthalpy of the secondary superheater in the identification.

Testing the applicability of the fuel-quality soft sensor in the control strategy is the paramount interest in this paper. Therefore, the detailed nonlinear dynamic models developed in Chapter 3.2 are used for the economizer, evaporator, and superheaters. The controls of the simulation model follow the control structure of the BioPower 5 CHP plant. The drum pressure control gives the set point for the air and fuel allocation block. Then the air and fuel allocation block gives the set point for the primary air and the secondary air actuators and the fuel feeder. Moreover, the flue gas oxygen content controller acts as a master controller for the secondary air control.

Fig. 5. The model input variables, steam inlet temperature, steam mass flow, steam pressure, flue gas temperature, and flue gas flow in the identification.

Fig. 6. The measured (dashed line) and estimated (solid line) outlet steam enthalpy of the secondary superheater in the validation.

Three different air and fuel allocation blocks and fuel quality compensation blocks were implemented to test three different control strategies. Firstly, the current control strategy used in the BioPower 5 CHP plant uses only the air and fuel allocation blocks. Secondly, in the combustion power control strategy, the amount of fuel burned is estimated and then compared with the fuel flow set point given by the fuel allocation block, and then the fuel flow set is modified accordingly. In addition, the integrator is added to remove the steady-state error between the fuel set point and the estimate. Lastly, the enhanced control strategy also takes into account the fuel moisture in both the air and fuel allocation blocks and in an estimation of the amount of fuel burned.

Fig. 7. The model input variables, steam inlet temperature, steam mass flow, steam pressure, flue gas temperature, and flue gas flow in the validation.

Fig. 8. The simulation model of the BioPower 5 CHP plant

6 Performance test of the fuel-quality soft sensor

The performance of the fuel-quality soft sensor was tested with real data obtained from the BioPower 5 CHP plant. Process tests were carried out and ten hours of data were logged with 1 s as the sampling time parameter (benchmark interval). Two different fuels were varied during the time period. The prediction horizon for the estimation of the value of the moisture parameter w was 60 s.

In order to show the performance of the fuel-quality soft sensor, the other variables, drum pressure and furnace temperature, were investigated to study the effect of the fuel quality. The value of the fuel moisture parameter w explains why the power of the boiler drops in Fig. 9 even though the stoker speed and the estimated amount of fuel burned increase in Fig. 10. More heat is needed for the evaporation of water, therefore moisture decreases the heat value of fuel. The parameters of the model were randomly changed by amounts ranging from -20% to +20% but the fuel-quality soft-sensor exhibited robust performance.

Fig. 9. The power of the boiler and the mass fraction of the fuel moisture

The validity of the fuel-quality soft sensor was tested by calculating Pearson's cross-correlation between different values shown in Table 3. The fuel-quality soft sensor uses the temperatures of the secondary superheater to estimate the moisture content of the fuel. It consequently senses the change in fuel quality early, and the delay depends on the prediction horizon used in Eq. (25).

The temperature of the furnace in Fig. 11 drops when the fuel moisture w increases. The change in fuel quality does not however appear until 16 min later than the change in the fuel moisture parameter. The temperature of the furnace is also disturbed by air flows that cool the furnace. A drum pressure controller is used to regulate the power of the boiler. It also compensates for small deviations in fuel quality. Since the controller is not designed to deal with large variations in fuel quality, such large variations result in saturation and in cessation of boiler power regulation.

According to these tests and the previous tests in [6], the fuel-quality soft sensor represents a promising way of measuring changing fuel quality that could be uniquely and successfully used in a control strategy.

7 Testing of the enhanced control strategy

The three different control strategies (the currently used control strategy, the combustion power control strategy, and the enhanced control strategy proposed in this paper) were evaluated using the BioPower 5 CHP plant simulator in a MATLAB simulation environment.

In the first test scenario, the performance of the enhanced control strategy was evaluated by causing disturbances in the fuel quality. The power demand of the boiler was kept at 14 MW while the moisture content of the fuel flow was switched from 55% to 65% during a time period of 1–2 h. The moisture content of the fuel flow used in the simulation of the combustion power control strategy and in the currently used control strategy was 54%. The settling time in the response of the current control strategy was about 2 h, whereas it was about 10 min when using the enhanced control strategy, as shown in Figs. 12–14. With the combustion power control strategy, there were minor oscillations.

In the second scenario, the power demand was changed from 14 MW to 15 MW while the moisture content of the fuel flow is kept at 57%. The moisture content of the fuel flow used in the simulation of the combustion power control strategy and in the currently used control strategy was again 54%. With the control strategy currently used, the change in power demand caused strong oscillations. In addition, oscillations propagated into the flue gas flow and flue gas oxygen content measurements, as shown in Figs. 15–17.

Fig. 10. The estimated fuel flow and the stoker speed.

Fig. 11. The drum pressure and temperature of furnace .

Fig. 12. Boiler power and drum pressure reactions to a change in the moisture content of flue flow.

Fig. 13. Primary air flow and fuel flow reactions to a change in the moisture content of flue flow.

Fig. 14. Reactions of flue gas flow and flue gas oxygen content to a change in the moisture content of flue flow.

Fig. 15. Boiler power and drum pressure reactions to a change in power demand.

Fig. 16. Primary air flow and fuel flow reactions to a change in power demand.

Fig. 17. Reactions of flue gas flow and flue gas oxygen content to a change in power demand.

Table 3

Correlations of different fuel quality indicators.

	Corr	Delay
Furnace temperature, boiler power	0.92	-666
Drum pressure, boiler power	0.81	-424
Temperature over superheater, boiler power	0.64	298
Flue gas temperature, boiler power	-0.67	-1464

For comparison purposes, all the tested control strategies are set to use the same pressure control with the same parameters. With the enhanced control strategy, the better estimation of the net combustion power could be taken into account in the allocation of fuel. As a result, there would be a bigger difference between the enhanced control strategy and the combustion power control strategy. It would also take into account the heat needed for evaporating the moisture in the fuel. This behaviour is shown in Figs. 12-14 as a steady period for the enhanced control strategy after disturbances resulting from the more accurate allocation of fuel.

The net combustion power estimated with 54% moisture content and 2.4 kg/s fuel flow is 18.2 MW. With 57% moisture content, the net combustion power is however 16.6 MW, and with 65% moisture content it is 12.5 MW. There is thus a 31% error in the estimation of net combustion power in the first test and an 8% error in the second test when the combustion power control strategy is used. This could make a significant difference when fuel moisture and combustion power estimations are used in an advanced control such as a model predictive control.

The process simulation tests proved that the enhanced control strategy is able to efficiently stabilize the combustion process. This control strategy managed to keep the pressure level stable by using air flow and

oxygen content measurements and fuel-quality soft sensor to estimate the fuel flow and the allocation of fuel.

8 Conclusions

The paper presents an enhanced method for estimating fuel quality in the BioGrate combustion process and method's use in control strategy improvement at the BioPower 5 CHP plant. This method utilizes the combustion power method and a nonlinear dynamic model of the secondary superheater. In this approach, only one nonlinear model is needed and it also decreases the estimation delay.

The performance of the fuel-quality soft sensor was tested with real data obtained from the BioPower 5 CHP plant. According to the tests, the fuel-quality soft sensor is a promising feasible and effective way to measure changing fuel quality. The enhanced control strategy was tested in a controlled simulation environment. The results of the tests dramatically demonstrate that the enhanced control strategy efficiently stabilizes the combustion process.

The enhanced control strategy has only been implemented in the BioGrate process. It is however usable in general and thus the same advantages can be achieved in other plants regardless of the fuels and burning methods used. The greatest benefits can, however, be attained in plants fueled with inhomogeneous fuels such as peat, coal, bark and waste.

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