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Control strategy for the combustion optimization for waste-to-energy incineration plant ⋆

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Abstract: Waste disposal is becoming more and more challenging. Indeed, global population is still increasing and countries that do not have enough space to create big landfills need to find other solutions to deal with this problem. The incineration of municipal solid waste (MSW), if well controlled, is a possible solution. According to Cheng and Hu (2010) incineration can reduce the volume occupied by MSW down to 90% while producing thermal and/or electrical energy. Also the clinker of incineration can be used in road building and the construction industry. But air pollution control remains a major problem in the implementation of incineration for solid waste disposal. Despite the long history of work in this area, the proposed control schemes of these waste-to-energy plants are quite basic. This paper presents a way to optimize such a plant by using Advanced Control techniques. The aim of this operation is to control the steam flow rate, and, therefore the energy production, while ensuring a complete combustion, which is synonym of minimal pollution emission.

Keywords: Power and Energy Systems, Automation, Modeling, Linear Control, Robust Control, Optimal Control.

1. INTRODUCTION

Waste incineration is a very complex process, mainly because of the high variability of the fuel composition and the interconnection between the combustion parameters. The industry still mostly relies on the use of PID controllers that tend to greatly simplify the physical model of the plant (monovariable approach). This inaccurate approach was observed during the visit of an incineration plant in Paris. The operators need to change often (several times by shift) some set points when the control is not good enough. So basically the control loop is not any more a closed loop but an open loop where the controller is the operator changing the set-point when needed. The classical PID control is not very effective because of the multivariable nature of the combustion process. In order to take into account this interconnection between the variables a more accurate model has to be made. There have been a lot of papers that use adaptative fuzzy logic such as Shen et al. (2005), and Krause et al. (1994) in order to control the process. The problem of fuzzy logic is that it relies a lot on the experience of the operator for the fuzzy rule base implementation and very little on the physical model of the plant. So some cases can be interpreted falsely by the operator and lead to an error. Also some papers have modeled the physical system and proposed other advanced control techniques. For instance Leskens et al. (2005) proposed a model predictive control strategy for the automation of the incineration plant. Bardi and Astolfi (2010) made a nonlinear control strategy for the combustion optimisation. These two strategies are based on physical and empirical models (mass, energy, etc). This paper uses a different control technique called Linear Quadratic Regulator (LQR), which is also based on the plant model. This advanced control strategy will be used with a different modeling than the previous ones in order to perform the combustion control of the incineration plant. The advantage of this control strategy is that the tuning of the controller is easier than in the previous techniques.Therefore this paper will firstly present the modeling of the combustion process then the control strategy for the optimization and finally some results and comparisons with the real process value.

2. PLANT MODELING

2.1 Presentation

The MSW incinerator that we will model has the configuration showed in Fig. 1. The plant has slight differences compared to standard waste-to-energy (WTE) plants. We can see in Fig. 1 the classical layout of a combustion chamber with the entrance of the waste, the air necessary to the combustion (Primary + Secondary airs) and, the heat source which is the already existing flame.

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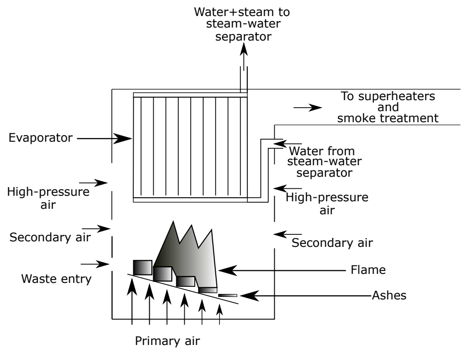

Fig. 1. Configuration of the incinerator

There are 4 transformations that the waste will undergo :

- Drying at the entrance of the combustion chamber which liberates some water vapour contained in the fuel;
- Pyrolysis which will liberate gases (mainly hydrocarbons: C_xH_y, carbon monoxide: CO, carbon dioxide: CO_2, water vapour: $H_2O_{(g)}$ and hydrogen: H_2) and transforms the remaining solid fuel into charcoal;
- Gasification which transforms some part of the charcoal that is not volatile into gases (H_2 and CO);
- Combustion which transforms the charcoal and the gaseous products of the two previous steps into carbon dioxide and water vapour.

Unlike most incineration plants, this one does not have an auxiliary burner. We can observe that there is also a high-pressure air injector above the secondary air, which purpose is to lower the temperature of the gas and to protect the furnace's walls against flames. The hot flue of gas which is mainly composed of water vapour ($H_2O_{(g)}$), carbon dioxide ($CO_{2(g)}$), nitrogen ($N_{2(g)}$), and some oxygen ($O_{2(g)}$) exchanges heat with the water contained in the evaporator as shown in Fig. 2.

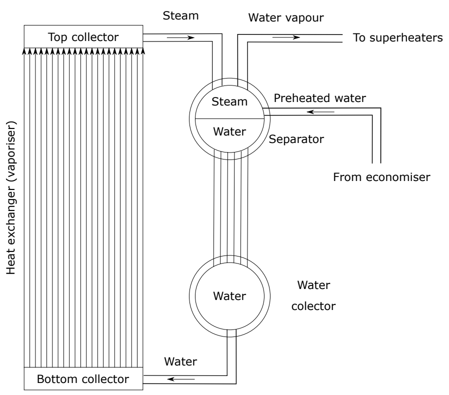

Fig. 2. Boiler: water vapour separator

This water boils and a buoyancy force sets in motion the mixture water+steam creating a flow rate. Then the remaining water is separated from the steam in a separator. The separator has 2 inputs (water+steam coming from the evaporator and, the preheated water coming from the economiser) and 2 outputs (pure steam going to the turbines/heat-exchangers and, preheated water going to the evaporator). Finally the steam is superheated by 3 heat-exchangers in order to obtain the nominal steam temperature for the turbines. Our modeling will be focused on the combustion chamber, before the separator.

2.2 Characteristics of the fuel and the combustion

On of the main challenges for the combustion modeling is the evaluation of the calorific value of the MSW. There are two types of approach for the analysis of a fuel. The first one is called ultimate analysis (chemical composition of the fuel), the second one is named proximate analysis (characterization of the fuel by some parameters such as moisture, volatile matter, ash and fixed carbon). Once the composition of the fuel is well known we can deduce the fuel main properties :

- High heating value (HHV) : The amount of energy released by the combustion of the fuel when the products of the combustion are taken at $0°C$, with water being entirely in the liquid state (vaporisation enthalpy is recovered). This value is used for instance in condensing boilers where water latent heat of vaporization is recovered.
- Low heating value (LHV) : The amount of energy released by the combustion of the fuel when the products of the combustion are taken at $0°C$, with water being entirely in the vapour state (vaporization enthalpy is not recovered). This value is used for classical boilers where the smokes at the exit of the process contain water vapour of the combustion. Which means that water latent heat of vaporization has not been recovered.
- Stoichiometric air-fuel ratio (V_a) : The volume of air needed for the theoretical total combustion of one unit of fuel mass (for solid fuels) or volume (for liquid and gaseous fuels). It has therefore the following units Nm^3_{air}/kg_{fuel} or Nm^3_{air}/Nm^3_{fuel}.
- Stoichiometric dry smoke-fuel ratio (V_{ds}) : The volume of dry flue gas (without water vapour) generated by the theoretical total combustion of one unit of fuel mass (for solid fuels) or volume (for liquid and gaseous fuels). It has therefore the following units Nm^3_{air}/kg_{fuel} or Nm^3_{air}/Nm^3_{fuel}.
- Stoichiometric wet smoke-fuel ratio (V_{ws}) : The volume of wet flue gas (with water vapour) generated by the theoretical total combustion of one unit of fuel mass (for solid fuels) or volume (for liquid and gaseous fuels). It has therefore the following units Nm^3_{air}/kg_{fuel} or Nm^3_{air}/Nm^3_{fuel}.

In real-life, combustion is not stoichiometric. The reaction is done in excess of air (the volume of air is bigger than the stoichiometric air-fuel ratio) or absence of air (the volume of air is smaller than the stoichiometric air-fuel ratio). Let V'_a be the real amount of air used for the combustion. We can define the *air excess* e (in %) by :

$e = 100 \times \left(\frac{V'_a}{V_a} - 1\right)$. This parameter is very important because it is linked to the quality of the combustion. Table 1 shows the different types of combustion according to the value of this coefficient.

This definition of the air-fuel ratio and the excess of air does not take into account the moisture content of the air used. In most industrial process atmospheric air is used,

Table 1. Different types of combustion

Coefficient	Oxidative	Stoichiometric	Reductive
e	> 0	$= 0$	< 0

which contains a certain amount of water vapour. The amount of moisture of air is characterised by the relative humidity given by: $\varphi = \frac{p_w}{p_{sat}(T_{air})}$ where p_w is the partial pressure of water vapour in the air in Pa, $p_{sat}(T_{air})$ is the saturation pressure of water vapour at the temperature of the air in Pa. If we consider that all the gases in the air are perfect we can use Dalton's law which states that the partial pressure of water vapour is equal to the total pressure times the molar fraction of water vapour. For perfect gases it is completely equivalent to calculate the molar fraction or the volume fraction because they are both related by the molar volume of perfect gases which is constant for a given temperature and pressure. So we can define the quality of the combustion air by considering the volume (or molar) ratio ϕ_{wd} of water vapour to dry air : $\phi_{wd} = x_{wd} = \frac{p_w}{p_{air}-p_w} = \frac{\varphi \cdot p_{sat}(T_{air})}{p_{air}-\varphi \cdot p_{sat}(T_{air})}$ where ϕ_{wd} is the volume ratio of water vapour to dry air, x_{wd} is the molar ratio of water vapour to dry air.

All incineration plants have O_2 sensors that measures the ratio of O_2 in the flue gas. This ratio enable us to calculate the excess of air of the combustion in real time as : $e = 100 \times \frac{V_{ds}}{V_a} \cdot \frac{\gamma_{d,O_2}}{0.21-\gamma_{d,O_2}}$

In France a comprehensive study of the MSW composition was conducted by Lopez et al. (2013). The ultimate and proximate analysis of the MSW led to table 2 which summarises the bulk composition and characteristics of waste.

Table 2. Proximate and ultimate analysis

Element	Mean composition (% dry weight)
C	39
H	5.73
O	33.00
S	0.16
N	0.75
Total	78.64
Characteristics	Mean value (% total weight)
Moisture	36.7
Energetic characteristics	Mean value (MJ/kg_{waste})
LHV (dry basis)	16.12
LHV (wet basis)	9.28
stoichiometric characteristics	Mean value (Nm^3/kg_{waste})
V_a	3.9
V_{ds}	3.8
V_{ws}	4.4

2.3 Primary, secondary and high-pressure air

The incineration plant under study has three different air inputs as shown in Fig. 1. The primary and secondary air are both preheated to maximize the combustion efficiency, whereas the high pressure air is at the atmospheric temperature in order to regulate the temperature within the combustion chamber. We can therefore assume that high-pressure air has little effect on the combustion process. So the combustion of MSW in the grate will be mainly controlled by the primary and secondary air, and more precisely by the first one. The flow rate of burnt fuel is given by : $q_{comb} = \frac{(1-\beta)\cdot Q_p+Q_s}{\left(1+\frac{e}{100}\right)\cdot V_a}$ where β is the percentage of primary air used for the drying and the cooling of the MSW, Q_p is the primary air volumetric flow rate in Nm^3/s, Q_s is the secondary air volumetric flow rate in Nm^3/s, q_{comb} is the combustion rate of MSW in the grate in kg/s. It is important to keep in mind that V_a does not take into account the composition in inerts and moisture of the fuel. So the real mass that has been incinerated is : $q'_{comb} = \frac{q_{comb}}{1-w_{moisture}-w_{inerts}}$ where q'_{comb} is the combustion rate of MSW in the grate in kg/s, $w_{moisture}$ is the total moisture content of the MSW, w_{inerts} is the total inert content of the MSW.

2.4 Combustion on the grate

In order to model the combustion of MSW on the grate we must make some assumptions which are :

Hypothesis 1. We will consider that the waste bed speed is a constant V_w and that primary and secondary air flows are evenly distributed on the grate.

Hypothesis 2. We will consider that for a given position on the grate, which is taken horizontally, the density is homogeneous in the waste bed along y and z axis.

Hypothesis 3. We will consider that the waste bed is distributed in 3 zones which are a homogeneous waste bed at the beginning, a combustion zone and a homogeneous ash (clinker) zone at the end. Only inert and combustible matter contribute to the height of the waste bed

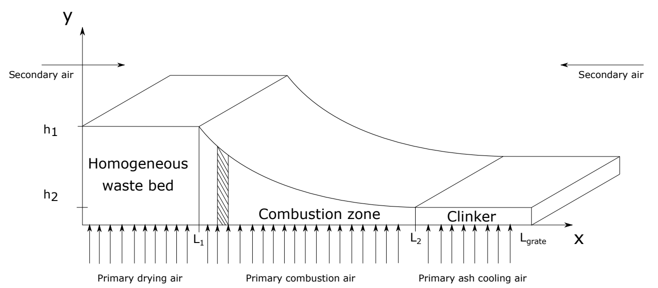

Fig. 3. Waste bed model

When modeling the combustion of the waste bed, it is common to make an energetic balance that will lead to a differential equation of the bed surface temperature as it is done by Bardi and Astolfi (2010). The bed temperature is not an easy thing to measure but it can be done by infra-red cameras. These cameras are expensive and if they are not already installed it is hard to evaluate the return of investment of such sensor. A control strategy, using an infra-red camera filming the waste bed from above, has been proposed by Schuler et al. (1994). The conclusion of the study showed that this control strategy was effective for the reduction of pollutant emission (up to 30% for C_xH_y and up to 10% for CO) and unburned material (up to 10%). But the steam flow-rate and the concentration of oxygen γ_{O_2} were comparable to the control that does not use the camera. So instead of using the camera and the bed temperature, our control strategy for the waste bed is based on its height. This height can be estimated by using the fact that the height is a function of the pressure under the grate zone $h = f(\Delta P)$. So by putting pressure sensors under the grate in the primary air zone we can deduce the height. Indeed we can see in Fig. 3 that there are two important variables which are the height of the bed at the beginning of the combustion $h_1(t) = h(x = L_1, t)$ and the height of the bed at the end of the combustion $h_2(t) = h(x = L_2, t)$. The first variable is ruled by

$$\frac{dh_1}{dt} = \frac{(1 - w_{moisture}) \cdot q_{feed}}{\rho_w \cdot l_g \cdot L_1} - h_1 \cdot \frac{V_w}{L_1} \tag{1}$$

where q_{feed} is the feeding rate of the furnace kg/s, ρ_w is the mean density of the MSW flow rate in kg/m^3, l_g is the width of the grate in m, V_w is the average speed of the waste bed in m/s, $w_{moisture}$ is the moisture of MSW. We can consider that the total height of the waste bed is the sum of the height due to combustible matter h_c and the height due to inerts h_i. If h_2 is the total height at the end of combustion then $h_2 = h_i + h_c$. h_i is fixed by the initial composition of the waste bed and h_c will vary during the combustion. The differential equation of h_c is established by a spatial discretization of

$$\frac{\partial h_c(x,t)}{\partial t} = -V_w \cdot \frac{\partial h_c(x,t)}{\partial x} - \frac{q_{comb}}{\rho_c \cdot l_g \cdot (L_2 - L_1)} \tag{2}$$

Where : ρ_c is the density of the combustible part of the waste bed kg/m^3.

2.5 Combustion energy

As explained before, the incinerator smokes that are rejected to the atmosphere contain water vapour. Thus the latent heat of vaporization is not recovered in the process. The pertinent variable to use here is LHV. This value is fundamental for the combustion and with MSW its variability is large. An on-line calorific value sensor has been proposed by Kessel et al. (2002). This device needs the composition of flue gas which is not commonly measured in the combustion chamber. A quick evaluation of the LHV can be done by evaluating the energy received by the water (heating, evaporation and superheating), the bottom ashes (unburnt metals) and the energy not used in the flue gas. Once the LHV is known, the combustion can be modeled as shown in Fig. 4.

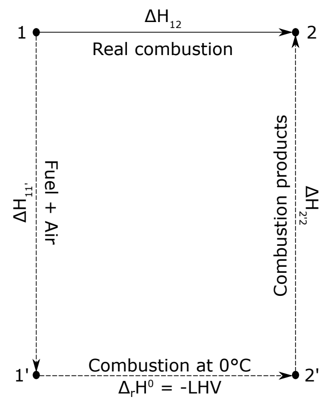

Fig. 4. Combustion cycle

The energy released by the combustion can therefore be written as

$$\Delta h_{1\to 2} = \Delta h_{1\to 1'} + \Delta h_{1'\to 2'} + \Delta h_{2'\to 2} \tag{3}$$

where $\Delta h_{i\to j}$ corresponds to the change of specific enthalpy during the transformation from i to j in J/kg.

2.6 Flame temperature

By assuming that the transformation is adiabatic (no heat transfer) and that there is no other energy applied to the system, we can estimate the flame temperature of the combustion gas. For this calculus we take into account the energy received by the inerts that do not participate to the combustion and the energy needed to elevate the water, contained in the MSW, to its boiling point:$\mathscr{P}_{inerts} = w_{inerts} \cdot q'_{comb} \cdot C_{p,inerts} \cdot (T_{flame} - T_{out})$ $\mathscr{P}_{water} = w_{moisture} \cdot q'_{comb} \cdot C_{p,water} \cdot (T_{sat} - T_{out})$ where $C_{p,inerts}$ is the average mass heat capacity of inerts J/kg/K, $w_{moisture}$ is the proportion of water in the fuel, $C_{p,water}$ is the average mass heat capacity of water J/kg/K, T_{flame} is the flame temperature K, T_{out} is the outside temperature in K, at which MSW is supposed to be in equilibrium with, T_{sat} is the evaporation temperature of water at the furnace temperature K. The flame temperature is calculated by considering $\Delta h_{1\to 2} = 0$ in (3) and by solving the equation $f(T_{flame}) = 0$.

2.7 Radiative heat transfer

In the combustion chamber radiation represents the major part of heat transfer. It can be assumed that the wall temperature corresponds to the vaporization temperature of the water at the circuit pressure ($P_{circuit}$). This temperature can be calculated thanks to a formula proposed by Osborne and Meyers (1934) that covers the range $0°$ to $374°$ corresponding to the critical temperature of water. Therefore we can use the Stefan-Boltzmann law of radiation which states that

$$\mathscr{P}_{gas} = \varepsilon \cdot S_{wall} \cdot \sigma \cdot (T_{gas}^4 - T_{wall}^4) \tag{4}$$

In order to calculate the emission coefficient Leckner (1972) proposed a method that presents a good accuracy at high temperatures, that is to say $T_{gas} > 400K$. The total emission of the gas flue is given by

$$\varepsilon_{H_2O-CO_2} = \varepsilon_{H_2O} + \varepsilon_{CO_2} - \Delta\varepsilon_{H_2O-CO_2} \tag{5}$$

where : $\varepsilon_{H_2O-CO_2}$ is the total gas mixture emissivity, ε_{CO_2} is the carbon dioxide emissivity, ε_{H_2O} is the water vapour emissivity, $\Delta\varepsilon_{H_2O-CO_2}$ is the correction term for the overlap. For these calculus, given that the radiation evolves with temperature, we will use the average temperature of the flue gas that is to say $T_{gas} = \frac{T_{flame}+T_{arch}}{2}$

2.8 Arch temperature

The arch temperature can then be found by considering an energy balance between the energy generated by the combustion, the energy lost by radiation, the energy taken by the inerts and the energy neededd to evaporate the water of the MSW. The energy balance will have the following form

$$\frac{d\mathscr{E}_{gas}}{dt} = \mathscr{P}_{radiation} + \mathscr{P}_{\text{combustion}} + \mathscr{P}_{inerts} + \mathscr{P}_{moisture}. \tag{6}$$

2.9 Steam flow-rate

The steam flow-rate can be estimated by different ways. Indeed we can suppose that the heat lost by the flue of gas is equivalent to the heat received by the water and therefore we can estimate the steam-flow rate. Another method will be to consider the total radiative heat transfer received by the walls. The thermal energy received by the water through the wall gives us an estimation of the steam flow-rate. The problem with these two methods is that the relationship between the output vector and the state space vector is non-linear. It is known that the arch temperature is closely linked to the steam produced, so by using the data available of the real plant a model has been found by using the least squares regression

$$Q_{steam} = a \cdot T_{arch} + b. \tag{7}$$

3. RESULTS

Once this model has been implemented in a simulation software the first thing that has been done was to compare it to the real process. Table 3 summarises the different characteristics of the simulations.

	Mean	Standard deviation
Measured arch temperature in $°C$	964	38
Simulated arch temperature in $°C$	948	42
Arch temperature error	3%	2%
Measured steam flow-rate in t/h	95	8
Simulated steam flow-rate in t/h	100	5
Steam flow-rate error	5%	6%

Table 3. Model and real values

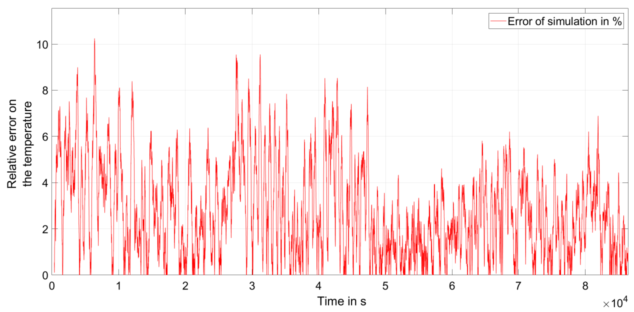

Fig. 5. Relative error for the temperature

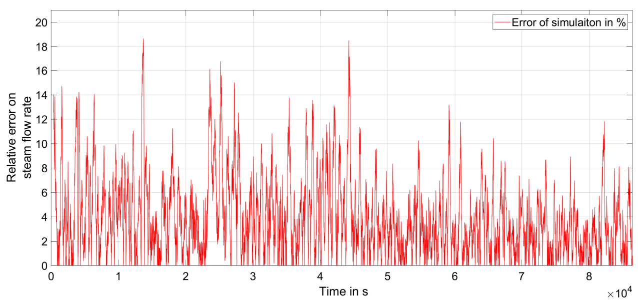

Fig. 6. Relative error for the steam flow

The high variability of the results is due to the fact that the most influencing factor, that is the LHV, is calculated by an indirect method. This parameter changes a lot with the composition of the waste and the error done in the exact calculation of this parameter is very influent on the accuracy of the prediction. Also the simplified model adds some error to the simulated values.

4. CONTROL STRATEGY

As mentioned before in this article, we are going to use a LQ regulator in order to perform the combustion control. The advantage of this technique in comparison to the basic PID controller is that the configuration of the latter is really hard for multivariable systems. The model used for the above simulations is non linear so we are going to linearize it around an operating point according to the value wanted by the client. The setting of the controller will be done with the linearized system and then applied to the real plant (non linear system).

4.1 Full-state feedback with integral loop

Let us consider the LTI system represented by :

$$\begin{cases} \dot{x}(t) = Ax(t) + B\big[u(t) + d_u\big] \\ \\ y(t) = Cx(t) + Du(t) + d_y \end{cases} \tag{8}$$

where $x(t)$ is the state vector $\in \mathbb{R}^n$, $y(t)$ is the output vector $\in \mathbb{R}^p$, $u(t)$ is the input vector $\in \mathbb{R}^m$, A is the state matrix $\in \mathbb{R}^{n\times n}$, B is the input matrix $\in \mathbb{R}^{n\times m}$, C is the output matrix $\in \mathbb{R}^{p\times n}$, D is the feedthrough matrix $\in \mathbb{R}^{p\times m}$, d_u is the constant disturbance vector at the input $\in \mathbb{R}^n$, d_y is the constant disturbance vector at the output $\in \mathbb{R}^p$. Matrix D is generally the null matrix (which is the case in our model) so we will consider for what follows that $D = 0_{p\times m}$.
A simple gain feedback is not feasible in real life because constant disturbances will lead to a static error. In order to cope with this problem an integral correction is applied to the simple gain feedback. By taking the derivative of equation (8) the disturbances d_u and d_y are eliminated. The resulting system is called the augmented system given by

$$\dot{x}_a(t) = A_a x_a(t) + B_a u_a(t) \tag{9}$$

where $x_a = \begin{bmatrix} \dot{x} \\ e \end{bmatrix}$ is the augmented state vector $\in \mathbb{R}^{(n+p)}$, $e = y - y_c$ is the error vector $\in \mathbb{R}^p$, u_a is the new input vector $\in \mathbb{R}^m$, $A_a = \begin{bmatrix} A & 0_{n\times p} \\ C & 0_{p\times p} \end{bmatrix}$ is the augmented state matrix $\in \mathbb{R}^{(n+p)\times(n+p)}$, $B_a = \begin{bmatrix} B \\ 0_{p\times m} \end{bmatrix}$ is the augmented input matrix $\in \mathbb{R}^{(n+p)\times m}$.

Control gain matrix $K_c = [K_p,\, K_i]$ with $K_p \in \mathbb{R}^{m\times n}$ and $K_i \in \mathbb{R}^{n\times p}$ defines the feedback $u_a(t) = -K_c \cdot x_a(t)$, so that the original control vector is equal to

$$u(t) = -K_p \cdot x(t) - K_i \cdot \int_0^t e(t)dt. \tag{10}$$

This control strategy can be summarize by Fig. 7

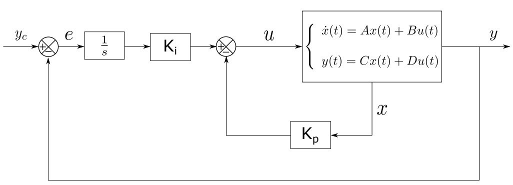

Fig. 7. Full-state feedback with integral loop

4.2 LQR principle

The LQR (infinite-horizon) method consists in minimizing a quadratic cost function according to the state-space representation of the plant. In our method, this function is based on the augmented system as follows :

$$J = \int_0^\infty e^{2\cdot\alpha_c\cdot t}\cdot(x_a(t)^T\cdot Q\cdot x_a(t)+u_a(t)^T\cdot R\cdot u_a(t))dt \quad (11)$$

where $Q \in \mathbb{R}^{n+p\times n+p}$ and $R \in \mathbb{R}^{m\times m}$ are weighting matrices, while $\alpha_c \in \mathbb{R}^+$ is a speed parameter. A feedback u_a minimizing cost function J can be computed via a Riccati algebraic equation. Then, modifying weighting matrices Q and R as well as parameter α_c easily leads to a controller satisfying some given specifications.

4.3 Command results

The command was tested for a typical case that is often encountered in incinerators. We consider a negative disturbance of the steam flow rate and we want to see the effect of the command on our system. As it is shown in Fig. 9 the disturbance rejection is done by a complementary work between high pressure air and primary air.

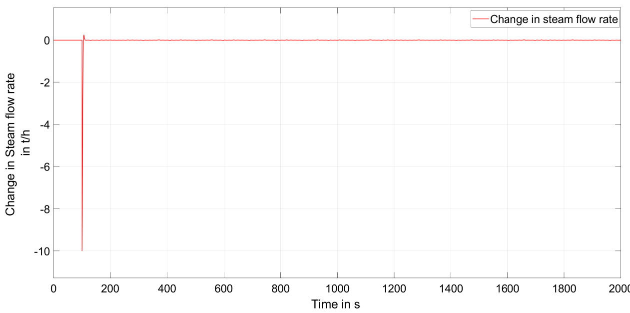

Fig. 8. Disturbance compensation of steam

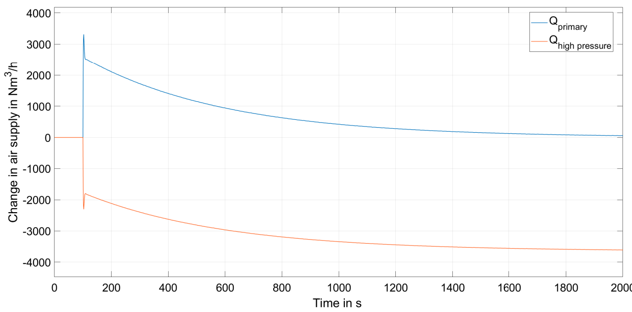

Fig. 9. Air supply command, primary air in blue and high pressure air in red

The matrices values for the simulation are :

$$A = \begin{bmatrix} -8.3333\cdot 10^{-4} & 0 & \\ 0 & -0.0042 & 0 \\ 0 & 0 & -0.4179 \end{bmatrix}$$

$$B = \begin{bmatrix} 5.33\cdot 10^{-5} & 0 & 0 \\ 0 & -4.7336\cdot 10^{-6} & 0 \\ 0 & 26.3468 & -32.2946 \end{bmatrix}$$

$$C = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 0.1136 \end{bmatrix}$$

5. CONCLUSION

This paper propose a new way to optimize the combustion of a WTE facility by setting new optimisation criteria. Indeed the control of the waste bed height, at the beginning and at the end of the process, ensures a good combustion quality. Also in order to implement our control strategy a new model of an incineration plant has been presented. Finally a multivariable control strategy has been proposed in order to cope with the insufficiency of a classical SISO (Single Input Single Output) control strategy.

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