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Optimal Operation of a Waste Incineration Plant for District Heating

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Abstract— **A case study of a waste incineration plant operating close to optimality by using simple feed-back control schemes is presented. Using off-line optimization the structure of the optimization problem is exploited and a set of variables is found, such that if the process is controlled with those variables are at their setpoints, operation is near-optimal.**

The procedure applied to the waste incineration plant is first to obtain a steady state plant model, which is optimized on grid points in the operating region in order to determine the set of active constraints and the optimally unconstrained variables and their optimal values. The variables assuming a constant optimal value are candidates for self-optimizing variables. This yields four operational regions, each with a set of corresponding self-optimizing variables.

For each region a simple control structure is defined to 1) satisfy constraints and 2) to control the self-optimizing variables to their optimal setpoints.

To be able to change between different regions, a switching table is set up. Using these switching rules, the plant can be controlled close to optimality within the different regions and when a disturbance causes the system to change from one region to another. Finally some dynamic simulation results are presented to show the control performance within the regions and across region boundaries.

I. INTRODUCTION

Rising energy prices, increasing competition and environmental demands make it increasingly necessary to operate plants as close to optimality as possible. In order to remain close to optimality in spite of disturbances, two approaches are usually considered [1]. The first paradigm is to obtain optimal operation via on-line optimization. This implies that the optimal setpoints of the controlled variables are computed on-line and are updated at certain time intervals based on the last available measurements. Setting up, solving and maintaining an RTO system can be a very time-consuming and complex task, as the uncertainty in the model and parameters can have a severe impact on the control performance, and the updated setpoints have to be available at the given sample times.

A second paradigm, which is very common in practice (although not always conscious of), is to identify appropriate “self-optimizing control” variables. Controlling these variables at their set-points keeps the process at or close to the optimal operating point in presence of disturbances without the need to re-optimize. Traditionally, such policies have been obtained by experience, nature or technical insight. The objective of our research is to find such policies in a systematic way, by performing the analysis and the calculations off-line. This systematic approach is here applied to a waste incineration plant for district heating.

In district heating networks, operators usually wish to obtain the lowest possible return temperature to the heat source, while the power plants are designed for providing heat at a certain temperature range. In this case study, the power plant is not owned by the district heating provider, which can lead to conflicts as the district heating provider attempts to draw more energy than what is produced, thus cooling down the plant. We design a control structure which prevents the plant from being cooled down while minimizing the operating cost. The example illustrates nicely the principles and benefits of self-optimizing control.

The structure of the paper is as follows: First the fundamental ideas of self-optimizing control are presented, then the waste incineration process is presented and explained together with the operating objectives. Next, the model is described and optimized. Based on the optimization results, a control structure is set up and is then tested for a dynamic model. After presenting and discussing representative results, the paper finishes with the drawn conclusions.

II. SELF-OPTIMIZING CONTROL

Performing the computations off-line and using the measurements to update the inputs using a feed-back scheme offers a very simple implementation and reduced cost to maintain. A concept within the second paradigm is self-optimizing control. The idea behind self-optimizing control defined in [1]:

> *Self-optimizing control is when we can achieve and acceptable loss with constant setpoint values for the controlled variables (without the need to reoptimize when disturbances occur)*

This means that for each region defined by the set of active constraints, we search for variables or variable combinations which are constant in presence of disturbances. If they are controlled at their optimal values, which is the same for all disturbances in that region, we indirectly obtain optimal operation, without having to reoptimize.

III. THE PROCESS

We consider a waste incineration plant with two production lines. The process flowsheet for one line is shown in Fig. 1. It is assumed that the lines are designed and operated symmetrically, such that it is sufficient to consider one line.

Cool water is flowing from the district heating network (DHN) and distributed equally onto the two production lines where is heated in the heat exchangers (HX) before it is returned to the network. Before the stream is split between

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TABLE I

MEASUREMENTS

y_1	Return temperature to furnace
y_2	Primary side heat exchanger exit temperature
y_3	Secondary side heat exchanger exit temperature
y_4	Cooler exit temperature (liquid)
y_5	Secondary side return temperature (to DHN)
y_6	Primary side flow rate

TABLE II

INPUTS

u_1	Bypass valve opening
u_2	Cooler valve opening
u_3	Primary side heat exchanger valve opening
u_4	Secondary side heat echanger valve opening
u_5	Secondary side bypass valve opening
u_6	Primary side flow pump duty
u_7	Cooling fan duty
u_8	Secondary side flow pump duty

the two plant lines, a bypass is installed to adjust the amount of water flowing thorough the heat exchangers.

In the two lines on the primary side, liquid water is heated to the desired temperature and transfers the heat to the secondary stream in the heat exchangers. The plant is equipped with an additional cooler, which is used when the DHN does not require all the produced heat. To prevent cooling down the plant, the exchanger can be bypassed.

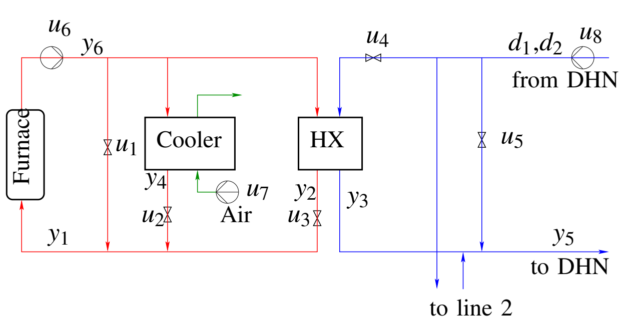

Fig. 1. Flowsheet of the incineration plant

In this study, the plant operator is interested in operating the plant to provide 16 MW per line, while minimizing energy consumption for pumps and fans and still satisfying temperature and flow constraints. The available measurements and inputs are listed in Tab. I and II.

The two lines are operated symmetrically and are subjected to operational *constraints*: The furnace entrance temperature is given as $y_1 = (126 \pm 1)°\text{C}$ and should not be violated to avoid condensing of fume gasses and boiling in the pipes. The primary side flow rate $y_6 = 250$ t/h and the return temperature to the district heating network y_5 must be in the interval from 90°C-150°C. In addition, the primary side heat exchanger exit temperature y_2 must not exceed 126°C.

TABLE III

PARAMETER VALUES FOR HEAT TRANSFER h

Unit	a_1(K^2/(W^{2}m^2)	a_2(K/(Wm^2)	a_3(/1(m^2)	a_4(W/(m^{2}K)
h_{HX}	$4 \cdot 10^{-4}$	-0.15	21.92	7615.8
h_{Cool}	0.42	-20.44	432.3	666.09

IV. STEADY STATE PLANT MODEL

The most important modelling assumptions are: Symmetric lines, non-compressible fluids, no pressure drop in heat exchanger and pipes and no heat losses.

A. Heat exchanger

Using the flow rates w and the specific heat capacities c_p, β is defined as the ratio between cold and hot heat capacity flow rates:

$$\beta = w^c c_p^c / w^h c_p^h \tag{1}$$

The number of transfer units is η:

$$\eta = UA/(w^c c_p^c). \tag{2}$$

Where $U = h^h h^c/(h^h + h^c)$ is the overall heat transfer coefficient and A is the total heat transfer area. The variables h^c and h^h are assumed equal, and their flow dependency in heat exchanger and cooler is found from fitting the steady state model to the dynamic model used to test the results,

$$h_{unit} = a_1(w^c)^3 + a_2(w^c)^2 + a_3 w^c + a_4 \tag{3}$$

where the subscript *unit* stands for either the heat exchanger (HX) or the cooler (Cool). The parameter values are listed in table III.

We now define α as

$$\alpha = UA\left[1/(w^h c_p^h) + \varepsilon/(w^c c_p^c)\right], \tag{4}$$

where ε is a parameter which is 1 for co-current and -1 for counter current heat exchangers. Further we define

$$\gamma = \exp(-\alpha), \tag{5}$$

and obtain the dimensionless gain matrix:

$$\mathbf{D} = \begin{bmatrix} \frac{\gamma(1-\beta)}{\gamma-\beta} & \frac{\beta(\gamma-1)}{\gamma-\beta} \\ \frac{\gamma-1}{\gamma-\beta} & \frac{1-\beta}{\gamma-\beta} \end{bmatrix} \tag{6}$$

The outlet temperatures can then be described by $\mathbf{T}_o = \mathbf{D}\mathbf{T}_i$, with $\mathbf{T}_i, \mathbf{T}_o$ as the vectors of input and output temperatures, respectively. The transfered heat is $q = w^h c_p^h (T_i^h - T_o^h)$.

For counter-current heat exchangers, this matrix becomes singular when $\beta = 1$ (parallel temperature profiles), which is the case for some operating conditions. In order to do simulations also cases where $1 - \delta < \beta < 1 + \delta$ for small δ, we expand the exponential term in (5) [2] and define

$$S = \sum_{i=1}^{\infty} (-\eta\delta)^i/(i+1)! \tag{7}$$

Using (7), we write $\gamma = -\eta\delta(1-S) + 1$ and for small values of δ we use

$$\mathbf{D} = \begin{bmatrix} \frac{\gamma}{1+\eta(1+S)} & \frac{\eta\beta(1+S)}{1+\eta(1+S)} \\ \frac{\eta(1+S}{1+\eta(1+S)} & \frac{1}{1+\eta(1+S)} \end{bmatrix} \tag{8}$$

TABLE IV

OPTIMAL INPUT VALUES

Region	u_1 [%]	u_2 [%]	u_3 [%]	u_4 [%]	u_5 [%]	u_6 [%]	u_7 [%]	u_8[a] [%]
α	92.6	0	100	x	100	6.78	0	x
β	92.6	0	100	100	x	6.78	0	x
γ	x	0	100	100	0	x	0	x
δ	0	x	100	100	0	x	x	x

[a] u_8 is not actually an input as it is used to set the disturbance flow rate

instead of (6). The series S is truncated after $i = 5$.

B. Pump, fan, valve and mixer modelling

The fan duty P is calculated by

$$P = \frac{1}{\eta}\left[\frac{w^3}{2\rho^2}\left(\frac{1}{A_2^2} - \frac{1}{A_1^2}\right) + \frac{w\Delta p}{\rho}\right], \tag{9}$$

where η, w ρ, Δp are the efficiency, flow rate, density and pressure difference, respectively, and A_1, A_2 denote cross sectional areas of the pipes before and after the fan, respectively.

Assuming equal pipe diameters before and after the pump and no elevation difference, the pump pressure outlet is calculated by

$$\Delta p = (\rho P \eta)/w. \tag{10}$$

The valves are modelled by

$$w = K_v\sqrt{(\rho_o/\rho)\Delta p}, \tag{11}$$

with K_v being constant on the primary side, and being a function of the valve opening in the secondary side. ρ_0 is a reference density.

The mixers are described by heat and mass balances:

$$w_{tot} = \sum_i w_i, \tag{12}$$

$$T_{out} = \sum_i (w_i/w_{tot})T_i \tag{13}$$

V. OPTIMIZATION

The optimization objective is to minimize the total work for the pumps and fans,

$$\min J = \sum w = u_6 + u_7 + u_8 \tag{14}$$

subject to the model equations and the operating constraints from section III.

To obtain an approximation of the operating regions, the disturbance space is discretized in two disturbance variables, namely flow and temperature coming from the district heating network. The temperature grid ranges from 65°C to 90°C and has a resolution of 0.1°C. The considered flow disturbance from the district heating network ranges from 500 t/h to 900 t/h and has a resolution of 1.6 t/h. The model is optimized for each of these grid points.

Evaluating the system for all grid points yields four regions, defined by constrained input variables. These regions are shown in figure 2.

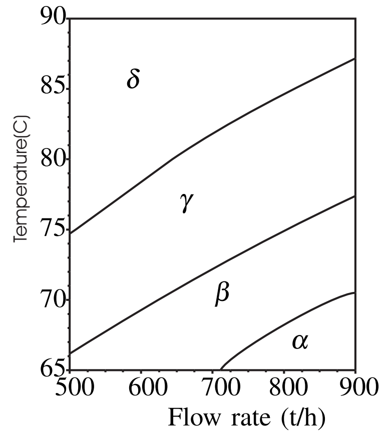

Fig. 2. Operating regions of the waste incineration plant

TABLE V

OPTIMAL OUTPUT VALUES

Region	y_1 °C	y_2 °C	y_3 °C	y_4 °C	y_5 °C	y_6 t/h
α	126	x	x	x	x	250
β	126	x	x	x	x	250
γ	126	x	x	x	x	250
δ	126	126	x	126	x	250

In table IV all inputs are given with the optimal values for each region. The x in the table indicate that the corresponding variable does not assume constant value throughout the region. In most cases when an input assumes a constant value, it is at a constraint, i.e. 0% or 100%. Table IV shows the optimal output values.

In the set of measured variables, only the furnace return temperature and the primary side heat exchanger and cooler temperature y_2 and y_4 assume constant values of 126 °C in Region δ. Otherwise all measurements are non-constant over the whole region.

VI. CONTROL STRUCTURE DESIGN

In each region, the degrees of freedom (DOF) available for optimization $N_{opt,free}$ are determined according to [3],

$$N_{opt}^{free} = N_m - N_0 - N_{active}, \tag{15}$$

where N_m is the number of control degrees of freedom, N_0 is the degrees of freedom without steady state effect (here $N_0 = 0$), and N_{active} is the number of active constraints.

If the number of DOF is zero, all inputs are used to satisfy the constraint, and no self-optimizing variable is required, as the optimum is at a constraint. Whenever the number of DOF is larger than zero we have a number of inputs which we do not need to satisfy a constraint, and we may use these inputs to minimize the operating cost. This is done by controlling a variable, which has an optimally invariant value and therefore is a self-optimizing control variable.

Table IV and V show the active input and output constraints for each region. All inputs except u_5 are present in both lines, so when calculating the DOF free for optimization, this has to be taken into account.

In region α,where the bypass u_5 is fully open, we have

$$N_{opt}^{free,\alpha} = 14 \underbrace{-2}_{u_2}\underbrace{-2}_{u_3}\underbrace{-1}_{u_5}\underbrace{-2}_{u_7}\underbrace{-1}_{u_8}\underbrace{-2}_{y_1}\underbrace{-2}_{y_6} = 2,$$

in region β,where u_4 is fully open, we have

$$N_{opt}^{free,\beta} = 14 \underbrace{-2}_{u_2}\underbrace{-2}_{u_3}\underbrace{-2}_{u_4}\underbrace{-2}_{u_7}\underbrace{-1}_{u_8}\underbrace{-2}_{y_1}\underbrace{-2}_{y_6} = 1,$$

in region γ,where u_4 is fully open and the bypass u_5 is fully closed, we have

$$N_{opt}^{free,\gamma} = 14 \underbrace{-2}_{u_2}\underbrace{-2}_{u_3}\underbrace{-2}_{u_4}\underbrace{-1}_{u_5}\underbrace{-2}_{u_7}\underbrace{-1}_{u_8}\underbrace{-2}_{y_1}\underbrace{-2}_{y_6} = 0,$$

and finally, in region δ where $u_1 = u_5 = 0$ and where u_3 and u_4 are fully open, we have

$$N_{opt}^{free,\delta} = 14 \underbrace{-2}_{u_1}\underbrace{-2}_{u_3}\underbrace{-2}_{u_4}\underbrace{-1}_{u_5}\underbrace{-1}_{u_8}\underbrace{-2}_{y_1}\underbrace{-2}_{y_4}\underbrace{-2}_{y_6} = 0.$$

In table IV we see that the primary side bypass valve opening u_1 is constant throughout regions α and β, and using the two DOF to set u_1 to its the optimal value in the two lines gives optimal operation.

The remaining two DOFs have to be used to satisfy the constraints on the furnace return temperature. One possibility would be to control y_1 using the secondary side heat exchanger valve u_4 in region α (and u_5 in region β), while keeping the bypass valve u_1 at a constant opening. However, this approach is not desirable from a dynamic point of view, because of the long time lag between the secondary side valves and the furnace inlet temperature y_1.

Therefore, in region α it is chosen to employ an input resetting structure, which utilizes the direct effect of bypass u_1 to control the furnace inlet temperature y_1, while the secondary side heat exchanger valve u_4 is used to reset the primary side bypass valve u_1 to the optimal value (Fig. 3).

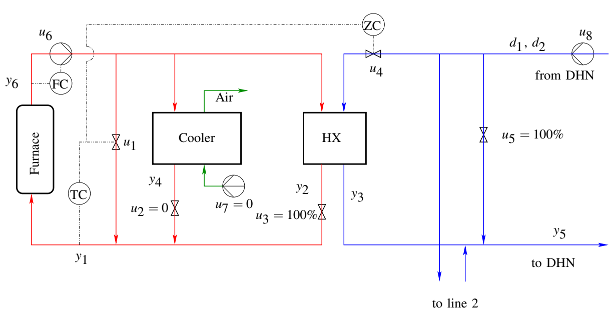

Fig. 3. Control structure for region α. (ZC: valve position controller for input u1)

In region β the bypass valve assumes the same constant value as in region α and is used as a self-optimizing variable as well. However, here the secondary side heat exchanger valve u_4 is in saturation, while u_5 may be use instead to reset the primary side bypass valve u_1 to its optimal value (Fig. 4).

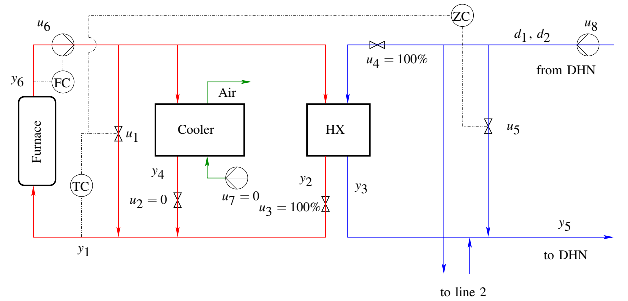

Fig. 4. Control structure for region β. (ZC: valve position controller for input u1)

Region γ does not have an unconstrained degree of freedom for optimization. This means that the system is operated optimally when all the (optimal) constraints are fulfilled. The usable manipulated variable in this case is the primary side bypass valve u_1, which is used to control the furnace return temperature y_1 (Fig. 5).

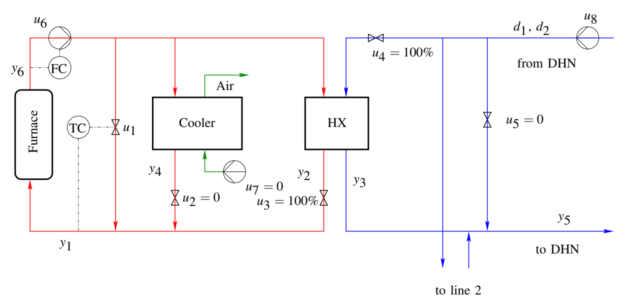

Fig. 5. Control structure for region γ

In region δ, the bypasses u_1 and u_5 are closed, while the heat exchanger valves u_3 and u_4, are fully opened. This region has two unconstrained inputs, the cooling fan duty u_7, and the cooler valve u_2. They are needed to control both, the furnace return temperature y_1 and the heat exchanger exit temperature y_2 to their setpoints at 126°C. This means that all three temperatures become $y_1 = y_2 = y_4 = 126$°C, because of the energy balance, the plant is operated optimally controlling any two temperatures of this set to 126 °C.

The relative gain array, [3], for a point in the middle of the region,

$$RGA = \begin{bmatrix} 1.1919 & -0.3854 \\ 0.1182 & 0.0758 \\ -0.3101 & 1.3096 \end{bmatrix}, \qquad (16)$$

suggests to pair u_2 with y_1 and u_7 with y_4. However this leads to a very poor dynamic performance, because in this pairing u_2 has very little initial gain on y_1 due to the equality of the exit temperatures of heat exchanger and cooler.

From the energy balance it is immediately clear that the heat has either to be removed in the heat exchanger or the cooler. Therefore opening the cooler valve u_2 alone will not have the desired effect on y_1. If the furnace return temperature y_1 becomes too hot and the cooler valve u_2 opens, it acts initially as a bypass and y_1 increases even further. However, as it closes, more water goes through the main heat exchanger, and the temperature y_1 increases as well. To effectively reduce the furnace return temperature y_1, it has to be controlled by the cooler duty u_4.

A set of pairings which gives good performance, is to use the cooling fan duty u_7 to control the furnace return temperature y_1 and to control the cooler exit temperature y_4 manipulating the cooler valve u_2 (Fig. 6).

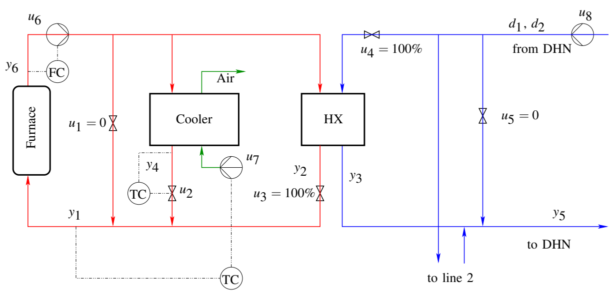

Fig. 6. Control structure for region δ

This pairing ensures that the fan duty u_7 increases before the disturbance coming from the district heating network affects the cooler exit temperature y_4, and avoids the bypass effect when the cooler valve u_2 starts opening.

VII. Dynamic model and simulations

In order to test the control structure and simulate the process, the model described in IV is extended to a dynamic model, where the heat exchangers are modelled as ideal tanks (Fig. 7). To add dynamics to pumps, valves and fans, a first order transfer function with $\tau = 1.5$ s was added. The mixers and splitters remain as previously described.

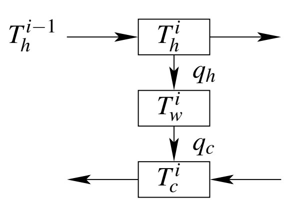

Fig. 7. Heat exchanger section

Each heat exchanger is modelled by 10 equal heat exchanger sections, with the governing equations [4]:

$$\frac{dT_h^i}{dt} = \left(T_h^{i+1} - T_h^i - \frac{h_h A}{w_h c_{p,h} N} \Delta T_h^i\right) \frac{w_h N}{\rho_h V_h} \tag{17}$$

$$\frac{T_w^i}{dt} = \left(h_h \Delta T_h^i - h_c \Delta T_c^i\right) \frac{A}{\rho_w c_{p,w} V_w} \tag{18}$$

$$\frac{dT_c^i}{dt} = \left(T_c^{i+1} - T_c^i - \frac{h_c A}{w_c c_{p,c} N} \Delta T_c^i\right) \frac{w_c N}{\rho_c V_c} \tag{19}$$

In the above equations T denotes the temperature, h the heat transfer coefficient, A the total heat transfer area, w the mass flow rate, c_p the heat capacity flow rate, N the number of sections, ρ the fluid density and V the volume. The superfix i denotes the compartment while the suffices h, c, and w denote the hot side, the cold side, and the wall element, respectively. The terms ΔT_h^i and ΔT_c^i express the signed difference between the wall and the hot and cold side of section i, respectively.

Modelling the heat exchangers discrete instead of continuous moves the regions up slightly, approximately 1 °C, but does not affect the structure of the problem. Using the dynamic model and the control structures developed above, the process was simulated for various scenarios in the different regions and for disturbances across region boundaries.

A. Control within Regions

As an example, the control performance in region α and δ is presented here. In region α, the control structure is well capable of keeping the variation in the furnace return temperature close to its desired value, while the self-optimizing control variable u_1 returns to its optimal value (Fig. 8). This reflects the control priorities: first, the active constraints are satisfied (y_1 close to 126), and second the system readjusts to optimal operation.

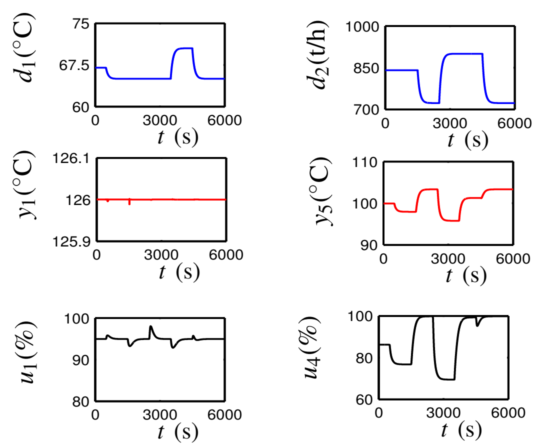

Fig. 8. Control performance in region α

In Fig. 9 the control performance for a disturbance in region δ is shown. It can be seen that the combination of cooling fan duty u_7 and cooler valve u_2 ensures deviation of less than 0.3°Cin y_1, even though the disturbances entering the plant are large. The control performance in the remaining regions, β and γ, is similar and is not shown in this paper.

B. Switching between regions

When the disturbances become so large that a region boundary is crossed, it is necessary to detect this event and to switch the control structure. For the switching strategy here it is assumed that the system cannot jump over a region, i.e. , the disturbances move the system gradually into the new region. Then, the switching logic can be based on monitoring the unconstrained and self-optimizing variables of the current and neighbouring regions. For example, in region α variable

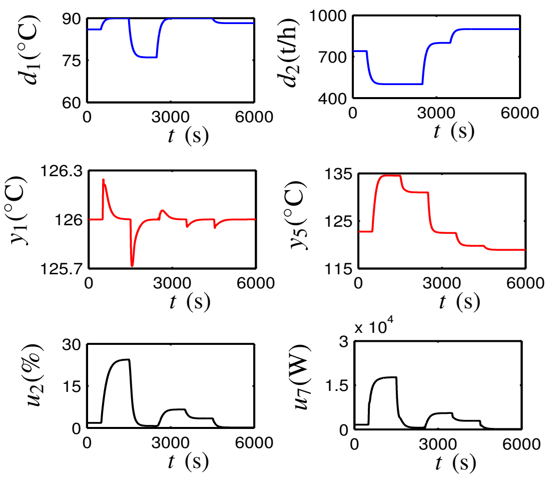

Fig. 9. Control for a disturbance in region δ

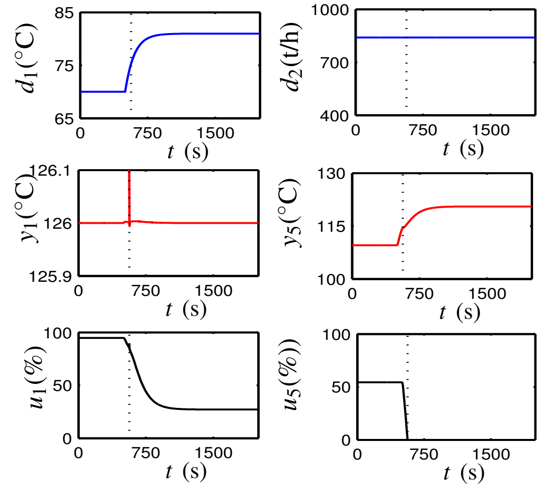

Fig. 10. Region switching from β to γ (dotted line shows switching instant)

TABLE VI

SWITCHING CONDITIONS

Transition	cond 1	cond 2	cond 3
$\alpha \rightarrow \beta$	$u_4 = 100\%$	$t < t_s$	Region = α
$\beta \rightarrow \gamma$	$u_5 = 0.00\%$	$t < t_s$	Region = β
$\gamma \rightarrow \delta$	$u_1 = 0.00\%$	$t < t_s$	Region = γ
$\delta \rightarrow \gamma$	$u_7 = 0.00\%$	$t < t_s$	Region = δ
$\gamma \rightarrow \beta$	$u_1 \leq 94.8\%$	$t < t_s$	Region = γ
$\beta \rightarrow \alpha$	$u_5 = 100\%$	$t < t_s$	Region = β

u_4 is unconstrained. If a disturbance enters such that u_4 goes into saturation, the unconstrained variable of region β is released and used for control. The same strategy is used for switching in the other regions. Switching from γ into the unconstrained region β is done when the self-optimizing variable of region β, u_1, reaches its optimal set-point. This is possible since in region γ the valve u_1 assumes a strictly smaller value than in region β.

To avoid chattering, the regions are switched when the corresponding variable has been in saturation or crossed its value for more than 2.5 minutes. Using this strategy we ensure that the control structure of one region is active long enough to realize its effects before switching to the next control system. The conditions for switching are listed in the switching table VI.

As an example, we consider a temperature rise in the district heating network, moving the system from region β to γ (Fig. 10). The variable u_1 is constant until u_5 goes into saturation. Then u_1 leaves the optimal point of region α to control the furnace inlet temperature y_1.

VIII. DISCUSSION

This case study shows the properties of self-optimizing control very nicely. The advantage is that for each region we have a very simple and easy to implement control structure. The procedure leads to a good understanding of the operating conditions and constraints. The knowledge from the degree of freedom analysis can be very beneficial for operation of the plant. In addition, it is easy to communicate to operators, as the control structures for each region are simple and easy to understand and maintain.

The challenge in handling several control structures is clearly tracking the operating regions and switching correctly. In this case study it has been found that monitoring the controlled variables of the four regions yields good results. The self-optimizing approach has been found to be a simple alternative to model predictive control (MPC), where the constraints are handled implicitly, i.e. the operating regions do not appear explicitly. Considering the simplicity of the control structure and the excellent control performance, it seems that the effort of maintaining and installing MPC may not be able to improve performance significantly. However this would has to be investigated in a separate study.

IX. CONCLUSION

This paper presents a case study of a waste incineration plant which is operated close to optimality using very simple control configurations and simple switching conditions for changing between them. The procedure applied reveals the different operation regions obtained from steady state operation explicitly, which makes the control structure more intuitive and understandable while still giving a very good performance. The switching rules are based on monitoring the constrained and self-optimizing variables and information about the system dynamics. For this process, the self-optimizing control approach seems to be an attractive alternative to MPC.

REFERENCES

[1] S. Skogestad, Plantwide control: the search for the self-optimizing control structure. *J. Proc. Cont., 10, 2000, 487-507*

[2] T. Hertzberg, Enkel varmaveksler, *Internal report* 2007, Department of chemical engineering NTNU, Trondheim

[3] S. Skogestad and I. Postlethwaite, Multivariable Feedback Control, 2^{nd} edition, John Wiley & Sons, England, 2005

[4] K.W.Mathisen, Integrated Design and Control of Heat Exchanger Networks, PhD thesis 1994, Norwegian University of Science and Technology, Trondheim