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CONTROL ISSUES IN THE DESIGN OF A GAS TURBINE CYCLE FOR CO_2 CAPTURE

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This article is concerned with control issues related to the design of a semi-closed O_2/CO_2 gas turbine cycle for CO_2 capture. Some control strategies and their interaction with the process design are discussed. One control structure is implemented on a dynamic simulation model using a predictive controller, and simulations assess the performance and compare its merits with a conventional PI structure. The results indicate that it can be advantageous for operability to allow a varying (as opposed to fixed) compressor inlet pressure, at the cost of a more expensive design. Furthermore, the results show that a predictive controller has some advantages with respect to the simpler conventional PI control structure, in particular in terms of constraint handling.

Keywords: ***CO_2 capture; Control design; Control-integrated process design; Semi-closed gas turbine cycle; Predictive control***

INTRODUCTION

Gas turbines are widely used for power production from gaseous fossil fuels. Although gas turbine engines are relatively clean burning, there is inevitably a production of CO_2 from combustion of fossil fuels. Thus, with today's increasing concern about global warming and climate change, there is an incentive to investigate gas turbine processes with CO_2 capture.

Focusing on gas turbines, it is generally acknowledged (see e.g. Bolland and Undrum (2003)) that there are three main concepts for CO_2 capture:

(a) Conventional power cycles where CO_2 is removed from the exhaust (post-combustion removal),
(b) Removal of carbon from fuel (pre-combustion removal), and
(c) Combustion with pure oxygen (instead of air), which leaves the exhaust consisting of CO_2 and water (easily condensed to obtain pure CO_2).

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While all these concepts have their pros and cons, we will in this paper concentrate on a process based on concept c).

The process we study (described in more detail in Section 4) recycles the exhaust gas, consisting mainly of CO_2 after water is removed, as working fluid in the gas turbine. CO_2 capture is achieved since some CO_2 must be removed from the cycle to avoid accumulation. The scope of the paper is to provide a limited assessment of the operational capabilities of this gas turbine cycle. In particular, we look at the design of control strategies to achieve close-to-optimal load control operation, despite disturbances that inevitably will excite the system. Therefore, we study both a conventional approach, as well as an approach based on online optimization (often denoted predictive control).

The process is studied in e.g. Bolland and Saether (1992), Bolland and Mathieu (1998), and Ulizar and Pilidis (1997, 2000), but must still be considered to be at the design stage. Apparently, the only analysis of transient performance is reported in Ulfsnes et al. (2003). On conventional (open) gas turbine processes, there is considerably more, for instance Rowen (1983) and Ordys et al. (1994). Predictive control of conventional gas turbines is suggested in Vroemen et al. (1999) with experiments in van Essen and de Lange (2001).

Section 2 of the paper provides some background as well as motivation for the integrated process design approach taken in this paper. Thereafter, the process is described, and a dynamic model is developed. In Section 6 the main challenges for a control system are discussed, and two different control methods are assessed. This is done by simulations of the dynamic model of the gas turbine cycle as well as its control system. A brief discussion and some conclusions end the paper.

A RATIONALE FOR INTEGRATED PROCESS DESIGN

As the focus of this paper is a process that has not been built yet, one might question why it is appropriate to look at dynamics, operability and control at this stage. We argue that for new process designs, it is advantageous to address issues related to operations at an early stage, even though this might not be usual in "traditional" sequential process design practice. The obvious reason is that an integrated process design practice—letting operational concerns have an early influence on the chosen process design—may lead to designs that are easier to operate than otherwise (van Schijndel and Pistikopoulos, 1999). The price to pay for this advantage is increased design complexity by the need to include dynamics at an early design stage.

Another (related) reason stems from the fact that power processes with CO_2 separation/capture thermodynamically has a disadvantage compared to processes without capture capabilities. A 10 percentage point efficiency loss is typical (Bolland and Undrum, 2003). Thus, to compete, it is imperative that these processes are optimized with respect to efficiency. This inevitably results in a strong focus on energy (and mass) recycle. Such designs generally lead to strong dynamic interaction between sub-processes, which typically is a challenge when it comes to operability and control. Therefore, it is important that there is a strong focus on dynamics already at early design stages, to avoid process layouts that are unnecessarily hard to operate.

To elaborate further on dynamic behavior, a power plant that delivers power to a grid needs to respond to changes in load demand in an acceptable manner, quickly and without too much overshoot. Good transient performance is more important than earlier since deregulated power markets implies more frequent power demand changes. Other situations where dynamics come into play are startup and shutdown of a plant, and different fault conditions. The latter may include the loss of sensor signals or an error in some auxiliary system.

CONTROL DESIGN

Control design amounts to two critical choices. First, it is necessary to define the input and output signals of the controller. The inputs are mainly online measurements, e.g., a frequency signal or a fuel flow measurement in a power plant. These input signals may include variables that are controlled as well as auxiliary measurements used to infer non-measurable variables. Input signals may, however, also be generated by plant operators, for instance a request for a load change. The output signals of the controller define the value of the plant's manipulated variables such as the position of a fuel valve. Second, the control algorithm itself must be defined. In general the controller will have several input signals and several manipulated variables, thus it is a multivariable controller.

In this work we will base the choice of control inputs and outputs (manipulated variables) on inspection of the semi-closed CO_2/O_2 gas turbine cycle and on the performance specifications of the controlled cycle. The latter implies that if for instance there are limits on the turbine inlet temperature (TIT) the controller needs some input signal that is related to the TIT.

Two control design methods are applied in this work. First, we design a conventional controller using single input single output PID-controllers. Second, we apply a model predictive control strategy implying that the controller at each sampling instant solves an optimization problem to compute the output signal. This method, which has gained a high reputation in the process industries, is briefly explained in the following section.

Predictive Control

Linear MPC (Model Predictive Control) refers to an online optimization where, at each sample instant, the control is determined by optimizing future behavior as predicted by a linear process model, subject to constraints on states (or controlled variables) and manipulated variables, then applying the first part of the computed control on the process (Maciejowski, 2002).

The linear discrete-time process model used for prediction is on standard state-space form,

$$x(k+1) = Ax(k) + Bu(k) + Ed(k),$$
$$z(k) = C_z x(k) + D_z u(k) + F_z d(k)$$

where $z(k)$ are the controlled variables. A standard linear Kalman filter (Gelb, 1974) is used to estimate the state ($x(k)$) from the measured variables and the manipulated variables ($u(k)$). The $d(k)$ is a disturbance state used in the Kalman filter to compensate steady state error in the controlled variables (i.e., integral control). Symbol k denotes the discrete time index.

We assume linear constraints on states (or controlled variables), manipulated variables and rate of variation on manipulated variables,

$$G_x x(k) \le g_x, \quad G_u u(k) \le g_u, \quad G_{\Delta u}(u(k) - u(k-1)) \le g_{\Delta u}.$$

The most important constraints that are imposed here, are the upper limit on turbine inlet temperature (1598K) and the constraint on valve operation (opening between 0 and 1, stroke time 15s).

We choose to minimize a quadratic objective function of the following form, which optimizes future behavior[1],

$$V(k) = \sum_{i=1}^{H_p} \left\| \hat{z}(k+i \mid k) - r(k+i) \right\|_Q^2 + \sum_{i=0}^{H_u} \left\| \hat{u}(k+i \mid k) - \hat{u}(k+i-1 \mid k) \right\|_R^2$$

where $\hat{z}(k+i \mid k)$ and $\hat{u}(k+i \mid k)$ are predicted variables at future time $k+i$, given information at time k (with $\hat{u}(k-1 \mid k) = u(k-1)$), and $r(k)$ is the reference (desired) trajectory for the controlled variables. We use $H_p = H_u = 50$, with sample time 0.5s, which means that we predict and optimize behavior over a horizon of 25s. The quadratic objective function together with the linear process model and linear constraints, imply that the problem can be formulated as a convex quadratic program, which can be solved efficiently.

PROCESS DESCRIPTION

A sketch of the process is shown in Figure 1. In the combustion chamber, methane (CH_4) and oxygen (O_2) react at a ratio slightly above the stoichiometric ratio. Recycled gas, mainly consisting of (CO_2), is compressed and used as an inert in the combustion to

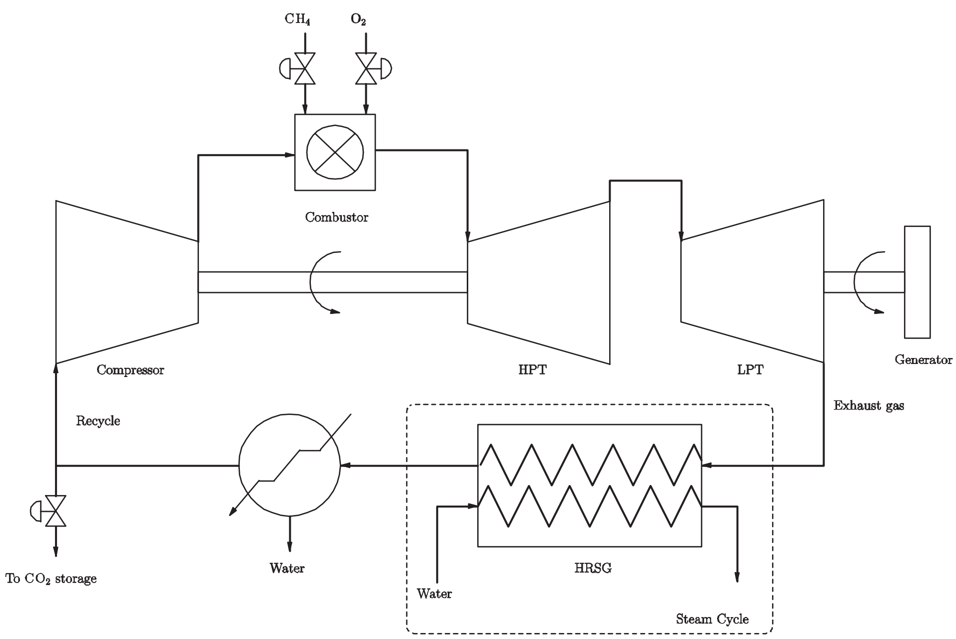

Figure 1 Semi-closed CO_2/O_2 gas turbine cycle – process layout.

[1]The norm $\|\cdot\|_H$ is defined by $\|z\|_H = \sqrt{z^T H z}$, $H > 0$.

Table 1 Typical values for some key variables.

Variable	*Symbol*	*Typical value*
LPT power output	$\dot{W}_{LPT}$	100MW
Turbine inlet temperature	TIT	1597K
Compressor mass flow	$\dot{m}_c$	173kg/s
Exhaust gas temperature	TET	1095K
Mass flow CO_2 to storage	m_{CO_2}	16kg/s
Fuel mass flow	$\dot{m}_{CH_4}$	6.1kg/s
O_2 mass flow	m_{O_2}	25kg/s
Compressor inlet temp.	T_{in}	290K
Compressor pressure ratio		19.3

limit temperatures in the combustion chamber and turbine inlet. The gas leaving the combustor is expanded in two turbines. The high pressure turbine (HPT) drives the compressor, while the low pressure turbine (LPT) is connected to a generator. The exhaust gas leaves the power turbine with a temperature well suited to deliver heat to a steam bottoming cycle. After the heat recovery steam generator (HRSG) the gas has to be cooled in a condenser, and condensed water is removed from the cycle. The exhaust gas, now mainly consisting of CO_2 is split into two streams; one stream is recycled to the compressor and the other stream is removed from the cycle for storage. We have tried to keep the notation standard; see also the nomenclature of Ulfsnes et al. (2003). Some typical design values for key variables are given in the Table 1.

MODELING

Choosing an Appropriate Model

In order to make decisions related to control and operability, it is crucial to have a dynamic model that represents the relevant dynamics well. It is important to be aware that these models have a different focus than the steady state models usually used for process design. The dynamic models used for developing and analyzing control structures do not need to be accurate when it comes to steady state, but needs to capture the dominant dynamic phenomena (as for instance dynamic interaction between different process parts). In the following we try to separate, conceptually, the different types of models that might be used towards commissioning of a plant:

1. Steady-state models. These are used for evaluation of “thermodynamic feasibility” and efficiency of a design, and optimization of steady state efficiency.
2. Advanced dynamic models. These are used for detailed study of the process. The complexity is high. The models may be defined by nonlinear partial differential equations.
3. Simplified dynamic models for control analysis and design. They may also be used for online purposes such as in a predictive controller.

As a side remark, an integrated process design procedure will usually imply the use of both a steady state model and a dynamic model in parallel.

The dynamic process model used in this work is based on Ulfsnes et al. (2003), and can be categorized as a simplified dynamic model, cf. the numbered list above. Some simplifications are made, mainly for computational efficiency reasons. The modeling is performed using the modeling environment gPROMS (Process Systems Enterprise Ltd., 2003). Thermodynamic properties have been determined with the physical property package Multiflash. A brief presentation of the model is given below.

Compressor

The power required for compression is equal to the increase in enthalpy,

$$\dot{W}_{\mathrm{c}} = \dot{m}_c \Delta h_c$$

The increase in specific enthalpy will be calculated by assuming it being somewhat larger (given by the efficiency) than the isentropic enthalpy increase $\Delta h_{c,s}$,

$$\Delta h_c \eta_{c,s} = \Delta h_{c,s}$$

We have assumed a constant isentropic efficiency $\eta_{c,s}$.

For a given compressor, the static relation between (dimensionless) compressor speed, compressor mass flow and compressor pressure ratio is usually called the *compressor map*. The "reduced" quantities are the standard quantities used for compressors with air as the working fluid (Saravanamuttoo et al., 2001):

$$N_{red} = \frac{N}{\sqrt{T_1}}, \qquad N_{\mathrm{dim}} = \frac{N_{red}}{N_{red,design}}.$$

$$\dot{m}_{red} = \frac{\dot{m}_c \sqrt{T_1}}{p_1} \sqrt{\frac{R}{\gamma_1}}, \qquad \dot{m}_{\mathrm{dim}} = \frac{\dot{m}_{red}}{\dot{m}_{red,design}}.$$

The gas constant R is dependent on the molar weight M_{c} of the working fluid, $R = \bar{R}/M_{\mathrm{c}}$. In our case, we will assume that the compressor map given by that dimensionless "reduced speed" N_{dim} is proportional to dimensionless "reduced mass flow" $\dot{m}_{\mathrm{dim}}$:

$$N_{\mathrm{dim}} = K\dot{m}_{\mathrm{dim}}$$

This corresponds to having vertical lines in the compressor map. This can be a good approximation in the normal operating range of a gas turbine cycle.

Combustion

Due to the rapid response of the combustion process, we have assumed an instantaneous mass balance, which gives the following mass flow leaving the combustion chamber,

$$\dot{m}_{out} = \dot{m}_c + \dot{m}_{CH_4} + \dot{m}_{CO_2}$$

Similarly, the energy balance is given by

$$\dot{m}_{CH_4}\Delta h_{CH_4} + \dot{m}_{O_2}\Delta h_{O_2} + \dot{m}_c\Delta h_{cc} + m_{out}\Delta h_{rx} = 0$$

where Δh_{rx} is the enthalpy of reaction, assuming all fuel reacts according to

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$$

Compared to using an equilibrium reactor as in Ulfsnes et al. (2003), this is a good approximation assuming that the oxygen excess ratio

$$\lambda_{O_2} = \frac{\dot{n}_{O_2}}{2\dot{n}_{CH_4}}$$

is large enough to achieve complete combustion. Furthermore, we assume a fixed percentage pressure drop over the combustion chamber.

The fuel (CH_4) and O_2 streams enter the combustion chamber through two valves. We assume both of these are controlled with flow controllers, and we assume that a perfect ratio controller controls the inflow of O_2 , such that a constant oxygen excess ratio is maintained. The reference $\dot{m}_{CH_4,ref}$ to the flow controller for the CH_4 stream is the manipulated variable for the controller to be designed. If we assume that this flow controller is well tuned, then we can write

$$\frac{d\dot{m}_{CH_4}}{dt} = \frac{1}{\tau_{CH_4}}(\dot{m}_{CH_4,ref} - \dot{m}_{CH_4})$$

where τ_{CH_4} is given by the bandwidth of the flow controller. Further, $\dot{m}_{O_2}$ is set to a fixed ratio of $\dot{m}_{CH_4}$ given by λ_{O_2} and the molar masses.

Turbine

The power generated by the high pressure turbine is

$$W_{HPT} = \dot{m}_{HPT}\Delta h_{HPT}$$

where the enthalpy drop is less than the isentropic enthalpy drop,

$$\Delta h_{HPT} = \eta_{HPT,s}\Delta h_{HPT,s}$$

given by the (assumed constant) isentropic turbine efficiency $\eta_{HPT,s}$. The same relations are used for the low pressure turbine (exchange HPT with LPT).

Moreover, we assume that both turbines can be regarded as “choked nozzles”, which is used to calculate the relationship between pressure drop, temperature and mass flow (and molar weight), when these differ from the design values (“off-design calculations”). The choked nozzle equation used here is given as,

$$\tilde{p}_{in} = \tilde{\dot{m}}\sqrt{\frac{\tilde{T}_{in}}{\tilde{M}}},$$

where ~ denotes the ratio to the design value, e.g. for the molar weight, $\tilde{M} = M/M_{design}$.

Rotating Shaft

The high pressure turbine drives the compressor via a rotating shaft. Newton’s second law gives

$$I\frac{d\omega}{dt} = \frac{\dot{W}_{HPT} - \dot{W}_c}{\omega}$$

where $\omega = \pi N/30$.

The low pressure turbine drives the generator via another rotating shaft. We assume that the generator delivers its power to an infinite bus, thus the rotating speed of the low pressure turbine will be fixed.

Heat Recovery Steam Generator and Condenser

In this work, we look at the heat recovery steam generator and condenser as a single counter flow heat exchanger. We do not model in any detail anything on the cold side of the heat exchanger. However, as the load of the plant varies, the amount of removed heat varies. For instance, a load increase will give a heat exchanger inlet mass flow with higher heat content. If the additional heat is not removed, then the compressor inlet temperature will inevitably increase, which will have a severe effect on the overall efficiency of the cycle. Thus, in a real plant, the steam bottoming cycle and condenser must be operated such that the changes in the compressor inlet temperature are suppressed. We have chosen to model this by letting a PI-controller decide the flow on the cold side of the heat exchanger such that the outlet temperature is kept constant. A suitable tuning of this controller represents the dynamics of the change in operating point for the steam cycle and condenser.

The heat transferred in the heat exchanger is modeled as proportional to the difference in average temperature between cold and hot side,

$$\dot{Q} = U_{wall}A_{wall}(T_{cold,avg} - T_{hot,avg})$$

where $U_{wall}A_{wall}$ is the heat transfer coefficient for the whole wall. We have used the arithmetic mean when calculating average temperature, since the (more correct) logarithmic mean proved to have a significant impact on computational performance, and because it is not important to have accurate temperatures on the cold side.

The heat exchanger will not react instantly to changes in the inflow. We thus model the "real" outlet temperature as a first order lag of the outlet temperature given from the above equation. For the hot side, this is

$$\frac{dT_{hot,out}}{dt} = \frac{1}{\tau_{HX}}\left(\frac{\dot{Q}}{\dot{m}_{hot}c_{p,hot}} + T_{hot,in} - T_{hot,out}\right)$$

and accordingly on the cold side.

In order to model pressure variations, a dynamic "overall" mass balance together with the ideal gas law is used. The outlet composition is set equal to inlet composition at all times.

Valve and Splitter

After most of the water is removed in the condenser, some of the CO_2 leaves the cycle through a valve. The flow through this valve is mainly determined by the pressure difference over the valve, using the valve equation

$$\dot{m}_{CO_2} = K_v\sqrt{\Delta p}u_v$$

where $0 \le u_v \le 1$ is the (rate constrained) valve opening, a control manipulated variable.

CONTROL STRUCTURE AND CLOSED LOOP SIMULATIONS

The control problem we consider is that of load control, i.e., operate the process so it supplies a specified load to the grid. As the process is open loop stable, the control objective is to operate the process as efficiently as possible, under varying disturbances. The major disturbances that affect the operation and are considered herein are load changes and disturbances affecting the heat transfer in the HRSG.

The first part of this section is devoted to the choice of the variables used for affecting process behavior (the manipulated variables), the variables the controller will try to influence (the controlled variables) and the variables used to get information about the state of the process (the measured variables). Thereafter, simulations show the performance of the chosen control structure, for both a simple PI controller, and a MPC controller.

Manipulated Variables

Possible manipulated variables are fuel valve, O_2 valve, CO_2 valve, compressor variable guiding vanes (VGV), and a number of variables affecting the operation of the HRSG and the condenser.

As explained above, we assume a perfect ratio controller to manipulate the O_2 valve to obtain a constant ratio of inflow of CH_4 and O_2. We also assume a well-tuned controller controlling the fuel valve, leaving us with the reference value as a manipulated variable. Furthermore, we have chosen to not include compressor VGVs. This is a limitation that will be discussed in Section 7.

We have not developed a detailed model of the cold side of the HRSG and the condenser, thus any manipulated variables related to these systems are not available to us. However, according to Kehlhofer et al. (1999), these are not normally used for load control in a conventional combined cycle. Thus, for the steam bottoming cycle, these manipulated variables should be used to operate the steam cycle as efficiently as possible for varying loads, removing as much heat as possible from the turbine exhaust.

This leaves us with opening of CO_2 valve and fuel inflow controller reference as manipulated variables (u) for this study.

Controlled Variables

With two manipulated variables, we can have two controlled variables. However, there are three natural selections of controlled variables.

Power: As this study considers the load control problem, the obvious controlled variable is the power output. The overall power output is the sum of the LPT power output, $\dot{W}_{LPT}$, and the steam turbine power output. Since the response in LPT power output is much faster than the steam turbine power output, we assume the LPT power output is controlled to achieve the total desired power output. In effect, this is not very different from how power control for the overall plant would work.

Temperature: For the semi-closed gas turbine cycle considered alone (without the steam cycle), the efficiency is mainly decided by pressure and temperature ratios. The pressure ratio is mainly decided from the design stage, but temperatures are subject to online variations. The closed cycle Carnot efficiency is maximized by keeping the turbine inlet temperature (TIT) as high as possible (limited by turbine material temperature constraints), and keeping the compressor inlet temperature as low as possible. Combined cycle considerations clutter the picture slightly, since the efficiency of the steam cycle must also be considered. However, Ordys et al. (1994) recommends for conventional combined cycles to keep the turbine exhaust temperature (TET) as high as possible (subject to constraints on TIT) to maximize energy flow to the HRSG, and we have adopted this philosophy in this paper and choose to control TET[2]. Note that for a given load, maximizing TET is close to maximizing TIT.

Pressure: The third variable, which is natural to consider for control, is the pressure at the low pressure side. In contrast to the conventional open gas turbine cycle, the semi-closed gas turbine cycle is not dependent on atmospheric conditions. Thus, the pressure at the compressor inlet may be different from the atmospheric pressure. An important design issue is whether this pressure should be allowed to vary (and in that case, how much it can vary), as this has a major impact on the physical design of the HRSG. If it is allowed to vary, the HRSG must be built with wall thickness that can handle the allowed pressure variations. On the other hand, if this pressure is specified to remain at atmospheric pressure,

[2]Nevertheless, we would like to note that several aspects also points in favor of controlling TIT, as recommended by Kehlhofer et al. (1999).

then conventional HRSGs can be used. A varying pressure can be beneficial both from an efficiency point of view, but also for handling disturbances.

In our simulations, we have chosen to not control the pressure, and hence let the pressure “float”.

For optimum part load efficiency, the optimum TET (and if pressure is controlled, the optimum pressure) may vary with load changes, but for simplicity we have chosen to keep the desired TET fixed in this study.

In summary, the controlled variables (z) will be LPT power output ($\dot{W}_{LPT}$) and turbine exhaust temperature (TET).

Measured Variables

Although the TIT imposes an important constraint, it is not possible to measure this variable, and the TIT must be inferred from other measurements. In our case, this is done through a Kalman filter, which is also needed to obtain the states for MPC prediction. We have used TET, N , $\dot{W}_{LPT}$ and the state of the steam cycle (the integral error of the PI controller controlling mass flow on cold side of HRSG) as measured variables (y) for the Kalman filter.

Closed Loop Simulations

The simulations are performed in gPROMS, while the controller calculations are done in Matlab. gPROMS communicates with Matlab via gPROMS’ Foreign Process Interface. The QP-problem is solved using quadprog from the Optimization Toolbox in Matlab. At each sample instant, the measurements are transferred from gPROMS to Matlab, where an optimal control trajectory is computed, and the manipulated variables for the next sample interval are returned to gPROMS. The linear models used in the MPC (cf. Section 3.1) were obtained from gPROMS, using the LINEARIZE-function in the full-load operating point.

The MPC closed loop trajectories are compared to trajectories from a well-tuned PI control structure (with anti-windup) where the turbine exhaust temperature is controlled by the CO_2 valve controller, while the flow of fuel controls the power output[3]. In a conventional gas turbine, the power loop would incorporate logic to avoid too high TIT, but this is not implemented here. A conventional process would also reduce mass flow by using VGV to keep high TIT/TET at part-load, but this is obtained in this process since the controller changes the total mass in the loop when operating the CO_2 valve.

A simulation of the closed loop is shown in Figures 1-3. The first disturbance (at 20s) is a change in power reference from 100MW to 80MW, and at 150s the reference returns to 100MW. At 300s, an “event” in the HRSG/condenser causes an increase in compressor inlet temperature from 290K to 310K in less than 10s. Note that both these disturbances are rather large considering how fast they happen.

We see that the PI controller obtains good control of power, at the cost of TIT constraint violations. Less deviation can be obtained by detuning the power controller. The MPC-controller obtains better control of temperature, at the cost of having a dip in power output at the last disturbance. How large this dip must be, is a matter of tuning – if a higher

[3] The reference to this controller is filtered to allow tighter control of the power subject to other disturbances.

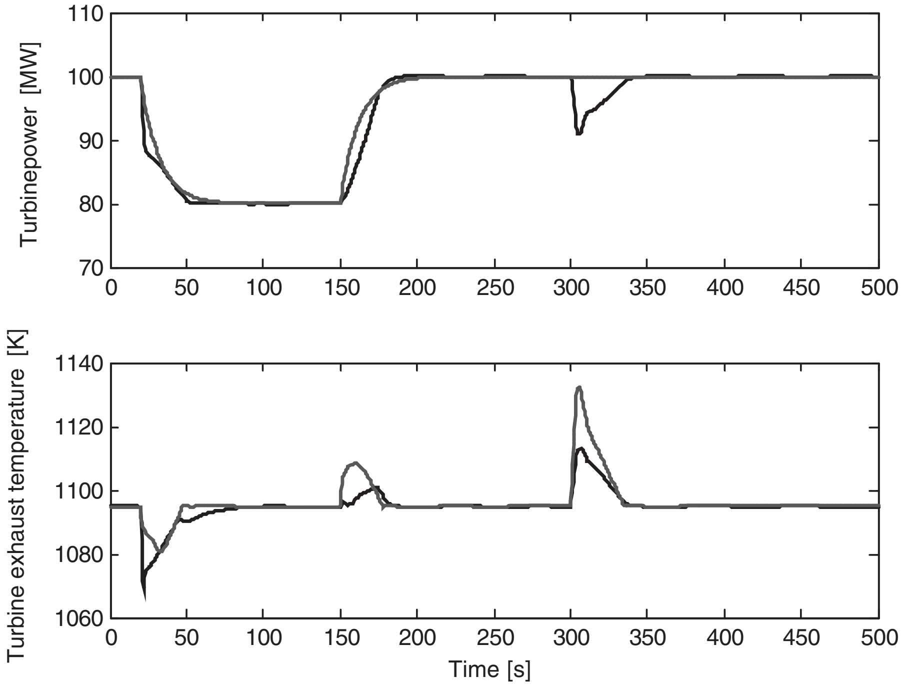

Figure 2 Controlled variables, using MPC (solid) and PI (dashed).

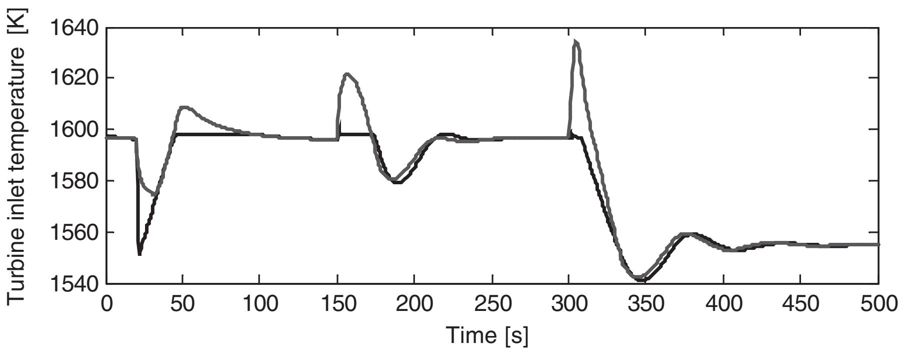

Figure 3 Turbine inlet temperature, using MPC (solid) and PI (dashed).

temperature limit is used (or the hard TIT constraint is replaced with a soft constraint), then better power control can be achieved. 370

DISCUSSION

Control structure: Since we have not considered compressor VGV as a manipulated variable, we have only two degrees of freedom for control. The power output has to be controlled, thus we must choose to use the second degree for controlling either temperature or pressure[4]. In this paper we have chosen to control temperature, which means that the pressure in the cycle will float, and the process equipment must hence be designed to tolerate this. However, a varying pressure has advantages both with respect to part load efficiency and disturbance handling, for instance, the CO_2-valve can be used for varying compressor mass flow during load changes.

If the HRSG pressure is constrained to be at atmospheric pressure, then the CO_2-valve is more or less assigned to this task, and there is no degree of freedom left for varying the compressor mass flow with this variable. Therefore, in this case, the use of compressor VGV is required for good part load performance.

Thus, our results show that for this process it is possible to vary compressor massflow using the CO_2-valve, and hence if the pressure is allowed to vary, it is viable to operate the cycle at part-load without using VGV. Nevertheless, in future work, we aim at including VGV as a manipulated variable, to obtain more control flexibility as well as possibly better part-load performance.

There are also some important constraints that are not taken into account in this paper, which might to some degree necessitate including VGV:

- Compressor surge constraints (operate the compressor to avoid surge).
- Pressure constraints, especially related to the HRSG.
- Constraints related to processing (compressing) downstream the CO_2-valve.

Some of these issues are addressed in Snarheim (2004a, 2004b). In particular, Snarheim (2004b) discusses optimal part-load operation using VGVs, for both fixed and varying HRSG pressure.

Control performance trade-off: As can be suspected, there is a trade-off between good control of temperature and good power control. The PI-controller can have considerably smaller maximum turbine inlet temperature if the power controller is detuned. As we can see in Figure 4, the PI power controller actually contributes to the temperature rise after the last disturbance. For the MPC controller, we see that the fuel manipulated variable is used to keep the temperature below the limit, but we must pay with a dip in power output. If we can allow a higher temperature limit (or allow the temperature limit to be a soft constraint), this dip can be considerably reduced.

Control complexity trade-off: There is a considerable difference in complexity between the PI control structure, implemented by simple algorithms, and the MPC control structure, consisting of online mathematical programming as well as a Kalman filter. The increase in complexity is rewarded by better control, and in particular better handling of constraints. As mentioned above, the PI-controller implemented in the simulations does not take constraints (apart from the constraints on manipulated variables) into consideration. In a real implementation, the PI-controller would have to be augmented with logic to achieve this, which in practice can be a tedious procedure involving tuning and iterations. On the other hand, constraints in MPC are specified in a straightforward manner, and handled in a structured way. There is also the possibility to include “soft constraints” (constraints that may be broken), and prioritize between different constraints.

[4]Note that even if we control pressure, we must make sure that the turbine inlet temperature constraint is not violated.

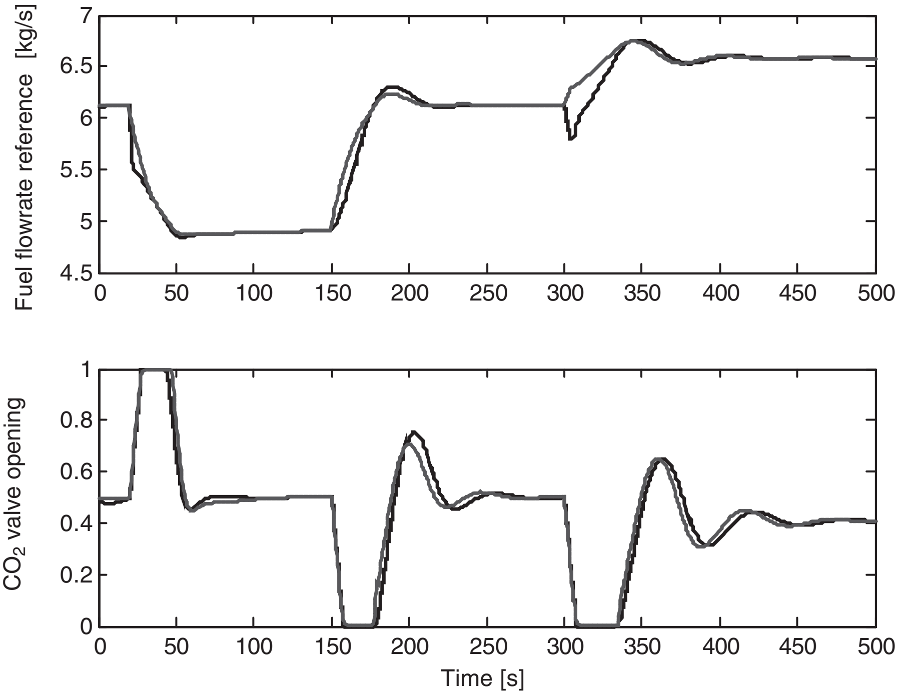

Figure 4 Manipulated variables, using MPC (solid) and PI (dashed).

CONCLUSION

In this paper we have argued that it is beneficial to integrate control issues early in design procedures. In particular, for the process considered in this paper, we claim that allowing pressure variations in the cycle allows better process control. These advantages must be weighed against the increased design complexity of a semi-closed gas turbine cycle without fixed pressure at the compressor inlet.

Furthermore, we show that using a rather complex control algorithm has several advantages, especially related to handling of process constraints. Again, these advantages must be weighed against the cost of implementation. However, as such controllers are becoming standard in other parts of industry, the cost and uncertainty of implementation of such controllers should be manageable.

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