MODELING AND CONTROL OF A O_2/CO_2 GAS TURBINE CYCLE FOR CO_2 CAPTURE

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Abstract: This article is concerned with the dynamics and control of a semi-closed O_2/CO_2 gas turbine cycle for CO_2 capture. In the first part the process is described and a model is developed. Thereafter, a control system structure is proposed, and an output feedback model predictive control algorithm is implemented and simulated on the process.

Keywords: Model predictive control, gas turbines, semi-closed cycle, CO_2 capture

1. 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 todays 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). 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 2) recycles the exhaust gas, consisting mainly of CO_2 after water is removed, as working fluid in the gas turbine. We will investigate dynamics and control of this semi-closed O_2/CO_2 gas turbine cycle, where CO_2 capture is achieved since some CO_2 must be removed from the cycle to avoid accumulation. In particular, we look at the design of a predictive controller that aims at achieving close-to-optimal load control operation, despite disturbances that inevitably will excite the system, while handling important process constraints explicitly.

The literature on this specific process is scarce, at least as far as dynamics and control are concerned. On conventional (open) gas turbine processes, there are 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). The modeling in this work is based on Ulfsnes *et al.* (2003).

In the first part of the paper, the process is described and a model is developed. Thereafter, the main challenges for a control system are discussed, and closed-loop simulations using a model predictive control algorithm is compared to a "conventional" approach using PI controllers. A brief discussion ends the paper.

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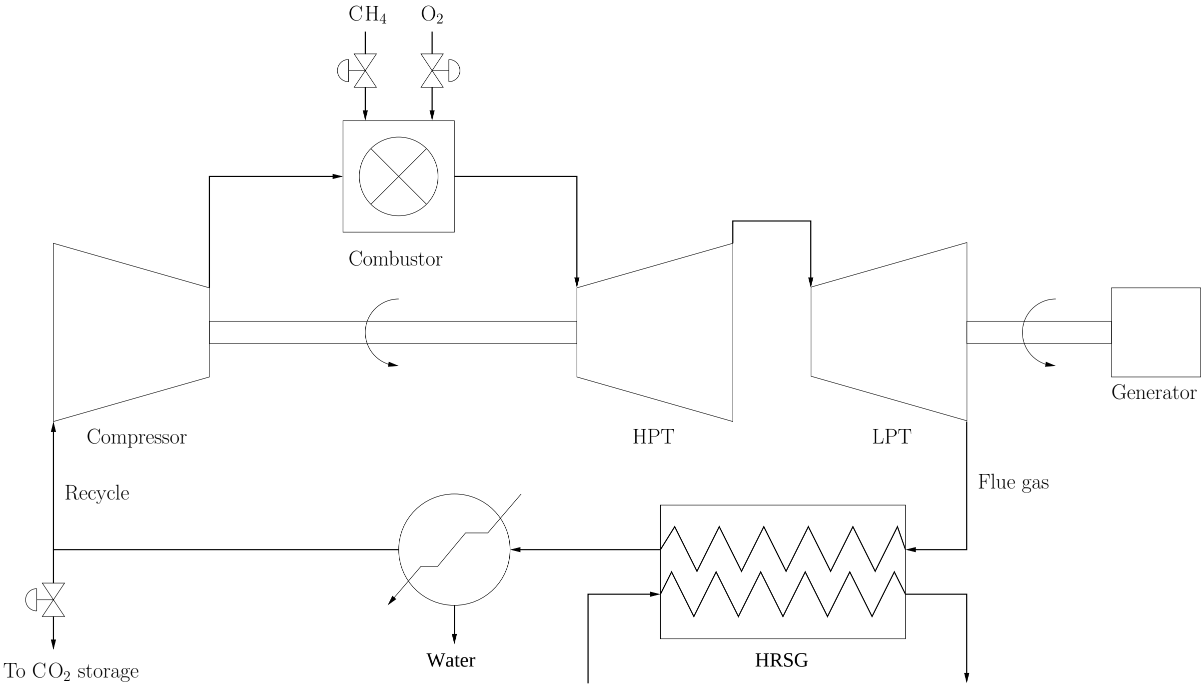

Fig. 1. Process layout

2. 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 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; the other stream is removed from the cycle for storage.

For space reasons, we will not give the values for all parameters, or provide a complete nomenclature. 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 below:

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

3. MODELING

The dynamic process model is based on Ulfsnes *et al.* (2003). Some simplifications are made, mainly for computational efficiency reasons. The modeling is performed using the modeling environment gPROMS (gPROMS, 2003). Thermodynamic properties have been determined with Multiflash, a physical property package.

3.1 Compressor

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

$$\dot{W}_{\mathrm{c}} = \dot{m}_{\mathrm{c}} \cdot \Delta h_{\mathrm{c}}.$$

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

$$\Delta h_{\mathrm{c}} \cdot \eta_{\mathrm{c,s}} = \Delta h_{\mathrm{c,s}}.$$

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

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):

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

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

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" is proportional to dimensionless "reduced mass flow":

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

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.

3.2 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,

$$\dot{m}_{\mathrm{out}} = \dot{m}_{\mathrm{c}} + \dot{m}_{\mathrm{CH_4}} + \dot{m}_{\mathrm{O_2}}.$$

Similarly, the energy balance is given by [2]

$$\dot{m}_{\mathrm{CH_4}}\Delta h_{\mathrm{CH_4}} + \dot{m}_{\mathrm{O_2}}\Delta h_{\mathrm{O_2}} + \dot{m}_{\mathrm{c}}\Delta h_{\mathrm{cc}} + \dot{m}_{\mathrm{out}}\Delta h_{\mathrm{rx}} = 0,$$

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

$$\mathrm{CH_4} + 2\mathrm{O_2} \rightarrow \mathrm{CO_2} + 2\mathrm{H_2O}.$$

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

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

is larger than, say, 1.02. Furthermore, we assume a fixed percentage pressure drop over the combustion chamber.

The fuel ($\mathrm{CH_4}$) and $\mathrm{O_2}$ 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 $\mathrm{O_2}$, such that a constant oxygen excess ratio is maintained. The setpoint $\dot{m}_{\mathrm{CH_4,ref}}$ to the flow controller for the $\mathrm{CH_4}$ 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

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

where $\tau_{\mathrm{CH_4}}$ is given by the bandwidth of the flow controller. Further, $\dot{m}_{\mathrm{O_2}}$ is a fixed ratio of $\dot{m}_{\mathrm{CH_4}}$ given by $\lambda_{\mathrm{O_2}}$ and the molar masses.

3.3 Turbine

The power generated by the high pressure turbine is

$$\dot{W}_{\mathrm{HPT}} = \dot{m}_{\mathrm{HPT}} \cdot \Delta h_{\mathrm{HPT}}$$

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

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

given by the (assumed constant) isentropic turbine efficiency $\eta_{\mathrm{HPT,s}}$.

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, when these differ from the design values ("off-design calculations"). The choked nozzle equation used here is given as

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

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

3.4 Rotating shaft

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

$$I\frac{\mathrm{d}\omega}{\mathrm{d}t} = \frac{\dot{W}_{\mathrm{HPT}} - \dot{W}_{\mathrm{c}}}{\omega}$$

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.

3.5 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 larger inlet mass flow, at about equal temperature. 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.

[2] The symbol Δh_{cc} is used to differentiate this enthalpy from the enthalpy increase in the compressor.

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

$$\dot{Q} = U_{\text{wall}} A_{\text{wall}} (T_{\text{cold,avg}} - T_{\text{hot,avg}}), \qquad (1)$$

where $U_{\text{wall}} A_{\text{wall}}$ 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.

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 (1). For the hot side, this is

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

and accordingly on the cold side.

In order to model pressure variations, a mass balance together with the ideal gas law is used.

3.6 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, using the valve equation

$$\dot{m}_{CO_2} = K_{\text{v}} \sqrt{\Delta p} u_{\text{v}},$$

where $0 \leq u_{\text{v}} \leq 1$ is the (rate constrained) valve opening, a control input.

4. CONTROL AND CLOSED LOOP SIMULATIONS

4.1 Control structure

The control problem we consider, is that of load control: 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. This study does not include start-up and shutdown of the system.

Manipulated variables: Possible manipulated variables are fuel valve, O_2 valve, CO_2 valve, compressor variable inlet guiding vanes (VIGV), 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 disregard any compressor VIGVs. This is a limitation that will be discussed in Section 5.

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. The impact of this is modeled by the PI-controller controlling the mass flow on the cold side of the heat exchanger.

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

Controlled variables: For the semi-closed gas turbine cycle alone, the Carnot efficiency is maximized by keeping the turbine inlet temperature (TIT) as high as possible (limited by turbine material 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) to maximize energy flow to the HRSG, and we have adopted this philosophy and choose to control TET. In a real application, the optimum TET setpoint will vary with load changes, but for simplicity we have chosen to keep it fixed. Note that for a given load, maximizing TET is close to maximizing TIT.

Since this study considers the load control problem, the controlled variables (z) will be LPT power output, $\dot{W}_{\text{LPT}}$ in addition to TET.

Measured variables: Although TIT imposes an important constraint, is not possible to measure this variable, and 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, $\dot{W}_{\text{LPT}}$ 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).

4.2 Predictive control

Linear MPC 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 inputs, 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,

$$x_{k+1} = Ax_k + Bu_k + Ed_k,$$
$$z_k = C_z x_k + D_z u_k + F_z d_k,$$
$$y_k = C_y x_k + D_y u_k + F_y d_k$$

where z_k are the controlled outputs, and y_k are the measured outputs. A standard linear Kalman filter is used to estimate the state (x_k) from the measured variables and the inputs (u_k). The d_k is a disturbance state used in the Kalman-filter to obtain integral control. The linear model is obtained using the `LINEARIZE`-function of gPROMS. We assume linear constraints on states (or controlled outputs), input and input rate,

$$Ex_k \leq 0, \quad Fu_k \leq 0, \quad G(u_k - u_{k-1}) \leq 0.$$

We choose to minimize a quadratic objective function of the following form [3],

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

where $\hat{z}(k+i|k)$ and $\hat{u}(k+i|k)$ are predicted variables at time k (with $\hat{u}(k-1|k) = u(k-1)$), and $r(k)$ is a reference trajectory for the controlled variables.

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 used $H_p = H_u = 50$, with sample time 0.5s.

4.3 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 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 [4]. 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 VIGV to keep high TIT/TET at part-load, but this is obtained in this process since the

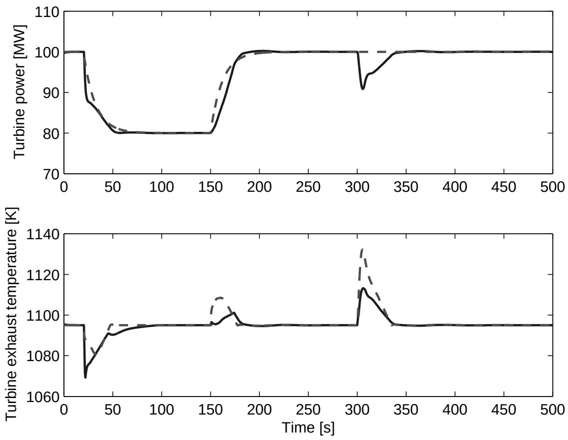

Fig. 2. Controlled var., MPC (-) and PI (- -).

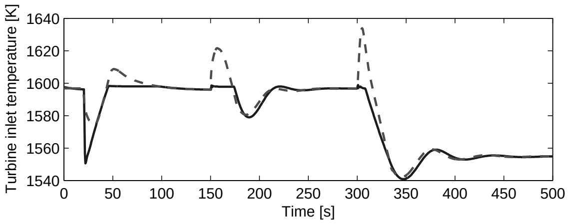

Fig. 3. TIT, MPC (-) and PI (- -).

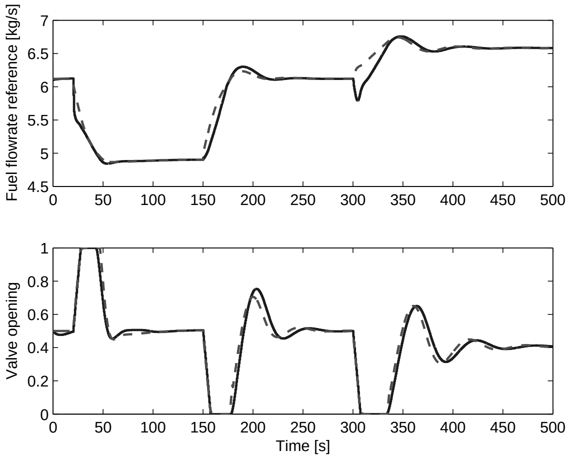

Fig. 4. Manipulated var., MPC (-) and PI (- -).

controller changes the total mass in the loop when operating the CO_2 valve.

A simulation of the closed loop is shown in Figures 2-4. The first disturbance (at 20s) is a change in power setpoint from 100MW to 80MW, and at 150s the setpoint goes back 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 (the PI controller controlling the compressor inlet temperature has a 20K setpoint change). 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 deviations can be obtained by detuning the power controller. The MPC-controller obtain

[3] The norm $\|\cdot\|_H$ is defined by $\|z\|_H = \sqrt{z^{\mathsf{T}} H z}$, $H > 0$.
[4] The setpoint to this controller is filtered to allow tighter control of the power subject to other disturbances.

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

5. DISCUSSION

Modeling: The developed model is mainly based on first principles and thermodynamics. However, as no such process exists today, there is considerable uncertainty related to several dimensions and characteristics. As we see it, the main uncertainty factor related to the dynamics is the modeling of the HRSG/condenser. Other issues that will be looked upon are a more realistic compressor map, and using a single shaft gas turbine. Introducing isentropic efficiency maps for the compressor and the turbines will also increase model confidence, but we believe that this will not have a significant influence on the dynamic properties of the process.

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 input 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.

Limitations: The control setup considered in the previous section is rather simple. Most importantly, compressor VIGV are not used as an input for limiting mass flows at part-load. However, the simulations showed that since this process is operated in semi-closed cycle, it is possible to obtain this effect using the CO_2-valve, and hence (to a certain extent) it is viable to operate the cycle at part-load without using VIGV. Nevertheless, in future work, we aim at including VIGV as control input, to obtain more flexibility.

There are also some important constraints that are not taken into account in this paper:

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

Including VIGV as input should facilitate handling these constraints.

Obtaining models: In this work, we obtained linear models at the operating point from gPROMS, using the `LINEARIZE`-function. Due to the nonlinearities (mainly the CO_2-valve), the prediction became bad when far from the operating point. It was experimented with obtaining linear models from several operating points, and use them in the prediction in a gain-scheduled manner. The models were "scheduled" over the prediction horizon using a process variable (in our case, a manipulated variable) from the previous time-step. In doing this, the prediction was significantly improved, resulting in better control (especially, better constraint handling)

6. CONCLUSION

A modular dynamic model of a semi-closed O_2/CO_2 gas turbine cycle for CO_2 capture was developed. A predictive controller, designed based on a linearization of the model, was shown to control the process satisfactorily. In particular, the controller keeps temperature constraints without resorting to the logic normally used by gas turbine load controllers, and is able to maintain reasonably high efficiency at part-load without using variable inlet guiding vanes. In future work, we will study how efficiency at part-load should be optimized using the increased flexibility of variable inlet guiding vanes, subject to more realistic constraints.

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