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, the main challenges for a control system are identified, and a model predictive control algorithm is implemented and simulated on the process.

Keywords: Model predictive control, Gas turbines, 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. 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) recycle the exhaust gas (consisting of mainly CO_2 after water is removed) as working fluid in the gas turbine. We will investigate some operational challenges related to 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 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 Kim *et al.* (2001). 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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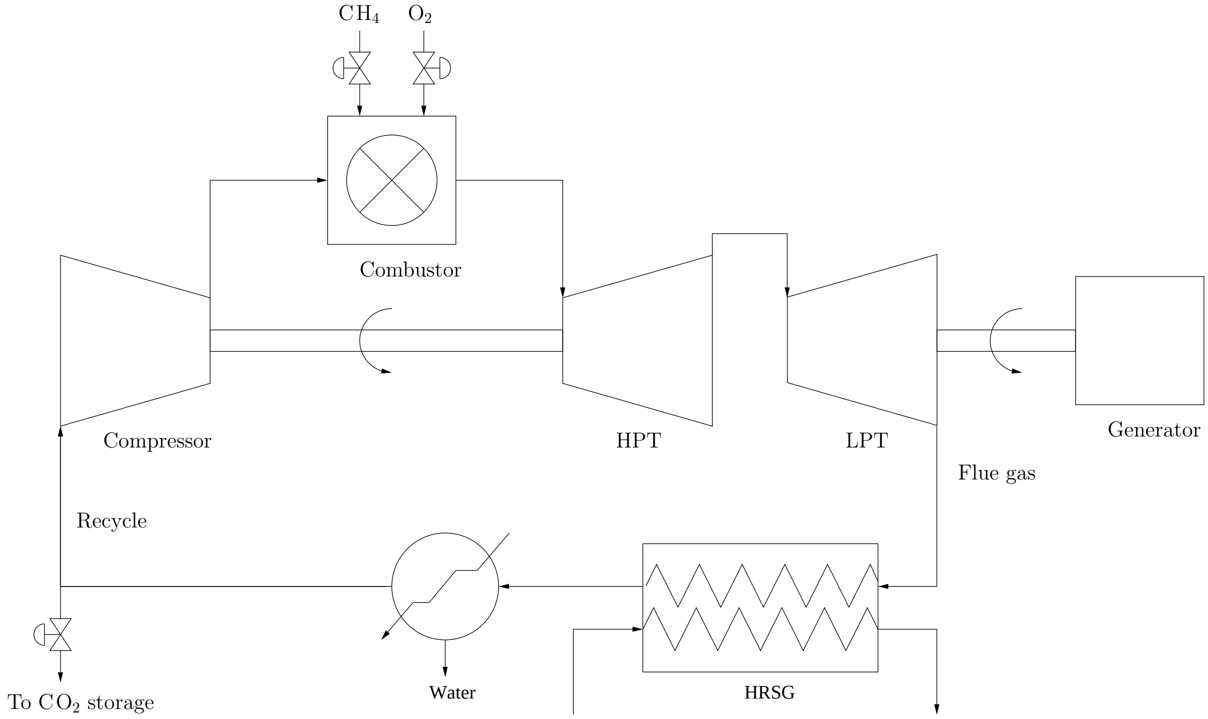

Fig. 1. Process layout

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). Some typical ("design") values for key variables are given in the table below:

Variable	Symbol	Typical value
LPT power output		100MW
Turbine inlet Temperature		1597K
Compressor mass flow	$\dot{m}_c$	173kg/s
Exhaust gas temperature		1095K
Massflow CO_2 to storage		16kg/s
Compressor inlet temperature		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 (see e.g. (Oh and Pantelides, 1996) and (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,

$$\dot{W}_c = \dot{m}_c \cdot \Delta h_c.$$

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

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

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):

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

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

The gas constant R is dependent on the molar weight M_c of the working fluid, $R = \bar{R}/M_c$.

In our case, we will assume that the compressor map given by that dimensionless "reduced speed" is proportional to dimensionless "reduced mass-flow":

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

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 (when compressor speed and mass flow are such that we stay far away from the compressor surge line).

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,

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

Similarly, the energy balance is given by [2]

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

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 the oxygen excess ratio

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

is larger than, say, 1.02. 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 setpoint $\dot{m}_{CH_4,ref}$ 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

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

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

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

3.3 Turbine

The power generated by the high pressure turbine is

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

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

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

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

3.4 Rotating shaft

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

$$I\frac{\mathrm{d}\omega}{\mathrm{d}t} = \frac{W_{HPT} - W_c}{\omega}$$

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.

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.

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

$$\dot{Q} = C_{p,wall}(T_{cold,avg} - T_{hot,avg}). \quad (1)$$

We have used the arithmetic mean, since the (more correct) logarithmic mean has a significant impact on computational performance. The PI-controller described above still makes sure the temperature drop on hot side is correct, but one should be careful when interpreting the physical significance of the values of the mass flow and temperature increase 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 (1). For the hot side, this is

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

and accordingly on the cold side.

In order to model pressure variations, a 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

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

where $0 \leq u_v \leq 1$ is the valve opening, and a control input.

4. CONTROL AND CLOSED LOOP SIMULATIONS

4.1 *Control challenges*

As the process is open loop stable, the control objective is to keep the process in a close-to-optimal operating point. 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.

Optimal operation: Expansion is more effective at high inlet temperatures, while compression is cheaper at low inlet temperatures. Thus, somewhat simplified and disregarding considerations related to operation of the steam bottoming cycle coupled to the HRSG, the efficiency of the cycle is high when the turbine inlet temperature is as high as possible, and the compressor inlet temperature is as low as possible.

The turbine inlet temperature is limited upwards for reasons related to safe operation of the turbine.

The capacity of the HRSG and the condenser limits compressor inlet temperature downwards. It is important to note that since the working fluid is recycled (as opposed to a "normal" gas turbine) the heat transfer in the HRSG has an important impact on the operation of the gas turbine cycle (in addition to the impact on a possible steam bottoming cycle).

Manipulated variables: Possible manipulated variables are fuel valve, O_2 valve, CO_2 valve, compressor variable inlet guiding vanes, 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 disregard any possible compressor variable inlet guiding vanes, as these are mainly used for start-up, and their use reduce compressor efficiency. Also, it will be seen that the CO_2 valve can have a similar effect on the massflow as such vanes have.

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, for the steam bottoming cycle, these manipulated variables should be used to operate the steam cycle as efficiently as possible, removing as much heat as possible. 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.

Controlled variables: The operational goal of the process is to deliver a desired power output as efficiently as possible. The main process constraint is the upper limit on turbine inlet temperature.

As we have not modeled a varying frequency on the power turbine (LPT), we control the power output directly instead of frequency. Hence, load changes will be treated as reference changes rather than as a disturbance.

Thus, in view of the discussion above, the controlled variables should be power output, turbine inlet temperature and compressor inlet temperature. However, since the compressor inlet temperature is mainly decided by the operation of the (unmodeled) condenser and steam bottoming cycle, we choose not to control this variable.

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,

$$x_{k+1} = Ax_k + Bu_k, \quad z_k = Cx_k + Du_k$$

where z_k are the controlled outputs. These might be different from any measured outputs, but in this paper we limit ourselves to the state feedback problem (the states and controlled outputs are available from gPROMS). 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.

We incorporate a simple integral control by assuming a "constant output disturbance" (the "DMC scheme", see e.g. Maciejowski (2002).)

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 15 sec). The horizon used, is $H_p = H_u = 30$s, with sample time 0.2s.

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, we "measure" (transfer) the state from gPROMS to Matlab, compute an optimal control trajectory, and return the manipulated variables for the next sample interval to gPROMS.

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

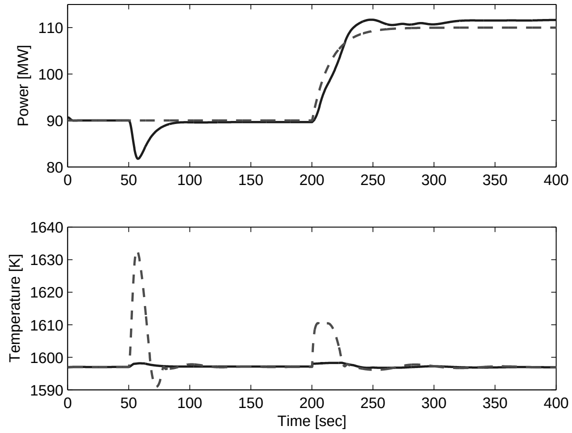

Fig. 2. Controlled var., MPC (-) and PI (- -).

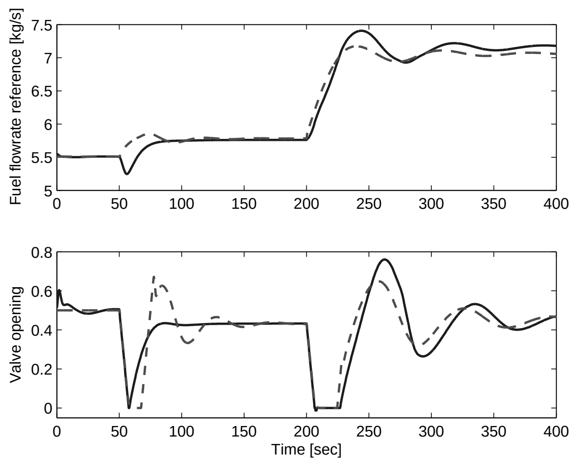

Fig. 3. Manipulated var., MPC (-) and PI (- -).

A simulation of the closed loop is shown in Figures 2 and 3. The first disturbance (at 50s) is an "event" in the HRSG/condenser that increases the compressor inlet temperature from 290K to 310K in less than 10s (the PI controller controlling the compressor inlet temperature has a 20K setpoint change). Secondly, after 200s, there is load increase (reference change) from 90MW to 110MW. Note that both these disturbances are rather large considering how fast they happen.

The MPC closed loop is compared to a well-tuned PI structure (with anti-windup) where the turbine inlet temperature is controlled by the CO_2 valve controller, while the flow of fuel controls the power output [4]. We see that the PI controller obtain good control of power, at the cost of significant temperature increase. Less deviations can be obtained by detuning the power controller. The MPC-controller obtain much better control of temperature, at the cost of having a dip in power output at the first disturbance. How large this dip must be, is a matter of tuning – if a larger temperature limit is used (or a larger deviation is allowed), then better power control can be achieved.

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

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 at 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.

State feedback: We have in this study assumed all states, as well as controlled variables, available to the controller. A next step will be to design a state observer (Kalman filter) to obtain an estimate of the states (and controlled variables) based on realistic measurements (note that it is not realistic to measure the turbine inlet temperature). In doing this, it is also natural to implement a better integral control than the "DMC scheme" used herein, by incorporating "integral states" into the observer. This should remove the steady state error shown in Figure 2.

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 2, the power controller actually contributes to the temperature rise after the first 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.

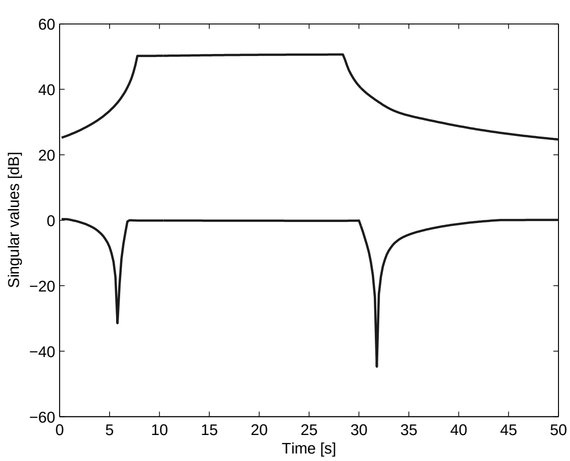

Fig. 4. Singular values of scaled $H(0)$ when load changes from 90MW to 110MW.

Nonlinearity: In Figure 4, we see a plot of the singular value of the scaled transfer function during a load change from 90MW to 110MW (similar to the change from 200s to 250s in Figure 2 and 3). We see that during this load change, the process gains changes considerably, thus the process exhibits significant nonlinearities. However, after the load change, the process gains are about as before. The nonlinearity is therefore not resulting from the actual load change, but from the operation of the CO_2 valve. The "corners" on the largest singular value corresponds to saturation of this valve. This nonlinearity also explains why the MPC controller only approximately respects the temperature constraint (a zoom on Figure 2 would reveal that the maximum temperature is 1598.3K); with a 25dB gain error in the model, some prediction error must be expected even with integral control.

6. CONCLUSION

A modular dynamic model of a semi-closed O_2/CO_2 gas turbine cycle for CO_2 capture was developed. The model is to a certain extent based on first principles and thermodynamics. A predictive controller, designed based on a linearization of the model, was shown to control the process satisfactorily. Nevertheless, several issues deserves further investigation.

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