Control Design for a Gas Turbine Cycle with CO_2 Capture Capabilities¶

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The semi-closed oxy-fuel gas turbine cycle has been suggested in (Ulizar and Pilidis, 1997) as an alternative for power production with CO_2 capture capabilities. This article is concerned with two critical design decisions for a similar process. Optimizations are used on a simulation model to evaluate part load performance for four different control structures. Finally a model predictive controller is implemented for each of the structures, and closed loop simulations are used to assess the different control structures.

1. Introduction

Fossil fuels are today a major source of power, and are likely to remain so in foreseeable future. At the same time, there is today an increasing concern about global warming and climate changes resulting from emission of CO_2 to the atmosphere. This gives a clear incentive to investigate fossil fuel-based power processes where the produced CO_2 is *captured*.

It is generally acknowledged (see e.g. Bolland and Undrum (2003)) that there are three main concepts for CO_2 capture from combustion of fossil fuels: (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). All these concepts have in common that they thermodynamically are less efficient compared to alternatives that emit (all) CO_2.

These processes are generally highly integrated, involving energy and mass recycle, and optimizing efficiency might lead to operational (control) challenges. Therefore it is important to look at the interplay between process design and control.

The process we study herein (described in more detail in Section 2), is based on concept (c) above. The exhaust gas from a gas turbine with CO_2 as working fluid, is used as heating medium for a steam cycle (similarly to a conventional "combined cycle"), before water is removed and the CO_2 is recycled as working fluid in the gas turbine.

The purpose of this paper is to investigate two critical design decisions: that is use of compressor variable guide vanes, and constant vs. "floating" HRSG pressure, and their effect on controllability and part-load efficiency.

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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 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). The modeling in this work is based on Ulfsnes *et al.* (2003).

2. Process description

A sketch of the process is shown in Figure 1. In the combustion chamber natural gas (here assumed to consist of methane (CH_4) only) burns in pure oxygen (O_2). Complete combustion is assumed and a perfect flow/ratio controller is assumed to provide oxygen at a ratio slightly above the stoichiometric ratio. Recycled gas, mainly consisting of CO_2, is compressed (in the compressor) and used as an inert in the combustion to limit temperatures in the combustion chamber and turbine inlet. Variable guide vanes (VGV) in the compressor can be used to reduce the mass flow through the compressor (and thus be used to control the high pressure turbine inlet temperature (TIT)). The gas leaving the combustor expands through two turbines. The high pressure turbine (HPT) is used to drive the compressor, while the low pressure turbine (LPT) is connected to a generator. The temperature at the inlet of the HPT is very high, and therefore some of the compressed CO_2 from the compressor is used as a cooling medium for the HPT. This twin shaft set up gives the process more flexibility (than a single shaft turbine) because the compressor does not have to operate at a fixed speed (the study of a single shaft turbine is ongoing work). The exhaust gas leaves the LPT with a temperature well suited to deliver heat to a steam bottoming cycle. In the heat recovery steam generator (HRSG), as much heat as possible is removed from the exhaust gas to produce power in a steam
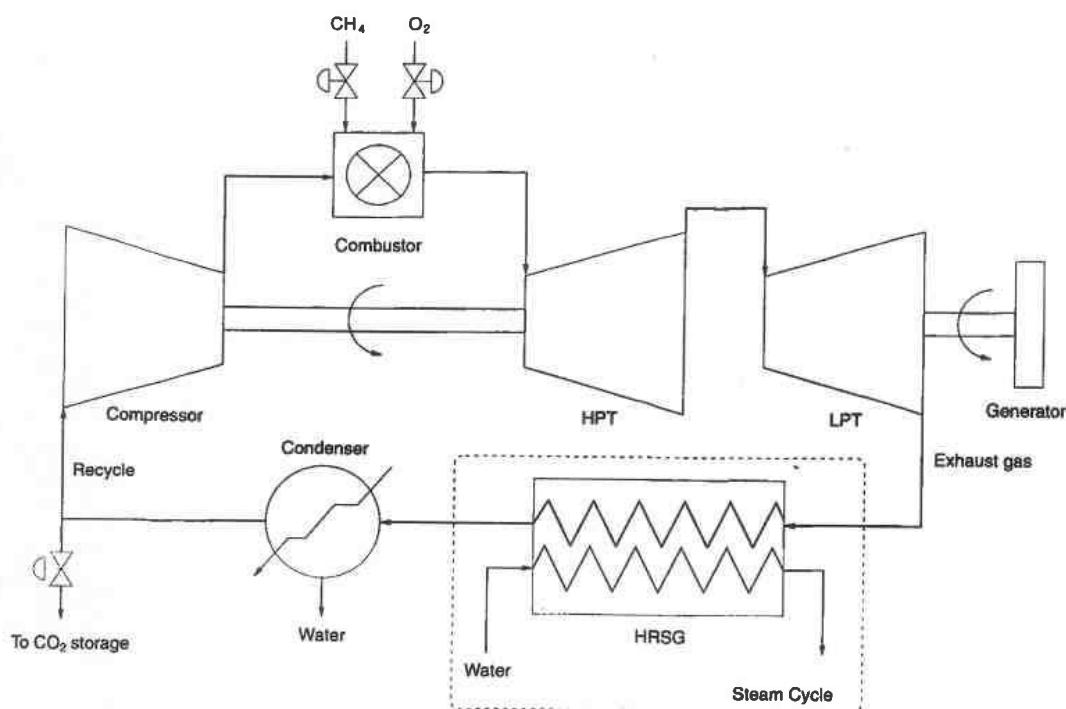

Figure 1.   Process layout.

turbine. After the HRSG the gas is further cooled in a condenser to remove water from the cycle. A controller is assumed to keep the temperature out of the condenser constant by manipulating the cooling water stream in the condenser. 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. The CO_2-valve controls the ratio between the streams. Some typical ("design") values for key variables are given in the table below:

Variable	Symbol	Typical value
LPT power output	$\dot{W}_{LPT}$	92.9MW
Turbine inlet temperature	TIT	1600K
Compressor mass flow	$\dot{m}_c$	227kg/s
Exhaust gas temperature	TET	982K
Mass flow CO_2 to storage	$\dot{m}_{CO_2}$	16.7kg/s
Fuel mass flow	$\dot{m}_{CH_4}$	5.8kg/s
O_2 mass flow	$\dot{m}_{O_2}$	23.6kg/s
Compressor inlet temp.	T_{in}	290K
Compressor pressure ratio	π_c	18.5

For space reasons we will not provide the complete modeling of the process. Instead the interested reader is referred to Imsland *et al.* (2004) or Ulfsnes *et al.* (2003) where the modeling is presented. Some few additions have been made; VGV have been implemented in the compressor (by parametrization of the compressor map along the VGV position), and we now also use turbine cooling (some of the compressed CO_2 is mixed with the combustion exhaust before entering the HPT).

In a conventional combined cycle VGV are used to increase heat flow to the HRSG, when operating at part load. When VGV are used, the gas turbine becomes less effective, but as more heat is transferred to the HRSG, the combined cycle efficiency increases. This means that to be able to study use of VGV for this process, it is necessary to calculate the power production in the HRSG. As it was chosen to not create a detailed model of the cold side of the HRSG, the power production must be estimated. To do this the gas turbine simulation tool GTPRO was used. Using this program we simulated a conventional HRSG in an operating range similar to "our" process. From this data we used curve fitting to generate a function for the power output as a function of TET and exhaust gas mass flow rate.

The complete simulation model was implemented in gPROMS (gPROMS, 2003). Thermodynamic properties have been determined with Multiflash©, a physical property package.

3. Control structure

The control problem we consider, is that of load control: Operate the process such that the overall power demand is met, and such that the power output responds fast enough to disturbances. 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. This study does not include start-up and shutdown of the system.

We will first briefly explain the manipulated variables available, before we discuss the different control structures that are investigated.

Manipulated variables: The manipulated variables available are the fuel valve, O_2 valve, CO_2 storage valve, compressor variable guide vanes (VGV), and a number of variables affecting the operation of the HRSG and the condenser. As explained in Section 2, a perfect ratio controller has been assumed to manipulate the O_2 valve to obtain a constant ratio of inflow of CH_4 and O_2. A well-tuned controller is also assumed to control the fuel valve, leaving the fuel flow reference value as a manipulated variable. As the cold side of the HRSG has not been modeled in any detail, the manipulated variables in the HRSG are not available. However, according to (Kehlhofer *et al.*, 1999) these variables 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 manipulated variables used for control are thus the opening of the CO_2 storage valve, the fuel flow reference value and the variable guide vanes.

Controlled variables: To make sure that the correct load is delivered to the grid, we need to control the overall combined cycle power output. But as the dynamics in the steam turbine in the HRSG are much slower than the gas turbine, faster response can be achieved by controlling the LPT power output. The setpoint for W_{LPT} must be set such that the combined cycle power output becomes as desired.

The remaining degrees of freedom should be used to keep the power production as efficient as possible while respecting important process constraints. Among the many constraints in this process, the TIT and the pressure in the HRSG are the most important ones. Violation of these constraints can cause severe damage on the process equipment, but at the same time the process needs to be operated close to these constraints to keep the cycle efficiency high. In a conventional combined cycle gas turbine, VGV are used to keep TIT high to increase heat flow to the HRSG when operating at part load. The main difference between this process and a conventional gas turbine is that the exhaust gas is recycled to the process and the working medium is CO_2. It is therefore not given beforehand that use of VGV are beneficial for this process, and this is one of the issues we want to investigate.

As this process operates in a semi-closed cycle it is necessary to control the pressure in the HRSG. If a conventional HRSG is used, this pressure must be kept constant at about atmospheric pressure, due to wall thickness in the heat exchangers in the HRSG. However, if this pressure can be allowed to vary, greater combined cycle efficiency can be achieved when the power plant is operating at part load. This is because if the pressure on the low pressure side of the compressor and turbines can be allowed to change, the pressure ratio in these components can remain almost constant, even though the mass flow is reduced.

Four different control structures were compared. In structure S1 and S2 the pressure in the HRSG is allowed to vary and the pressure setpoint depends on the desired power production (and is found by use of off-line optimizations). The pressure is still constrained between 0.8 bar and 1.2 bar. In control structure S3 and S4 the pressure is kept constant at 1 bar. In structure S1 and S3 VGV are used, which means that it is

possible to keep the temperature at a setpoint (1600K) at part load. These control structures are summarized in the table below.

	Contr. var.	VGV	Pressure	TIT
S1	$\dot{W}$,p,TIT	yes	$p_d = p_d(\dot{W})$	$T_d = 1600\text{K}$
S2	$\dot{W}$,p	no	$p_d = p_d(\dot{W})$	—
S3	$\dot{W}$,p,TIT	yes	$p_d = 1$ bar	$T_d = 1600\text{K}$
S4	$\dot{W}$,p	no	$p_d = 1$ bar	—

To investigate the impact of using VGV and varying pressure in the HRSG, the optimization tool in gPROMS was used. At different part loads an optimization problem was solved to minimize the use of methane feed when operating at the desired load, subject to constraints. The constraints used were TIT < 1600K, the pressure constraint for the given control structure and that the solution had to be a steady state solution. The results from the optimizations are shown in Figure 2, where a relative value means the actual value divided by the value when the process is operating at its design point (i.e relative power means $\frac{\dot{W}}{\dot{W}_{design}}$). Efficiency is calculated as

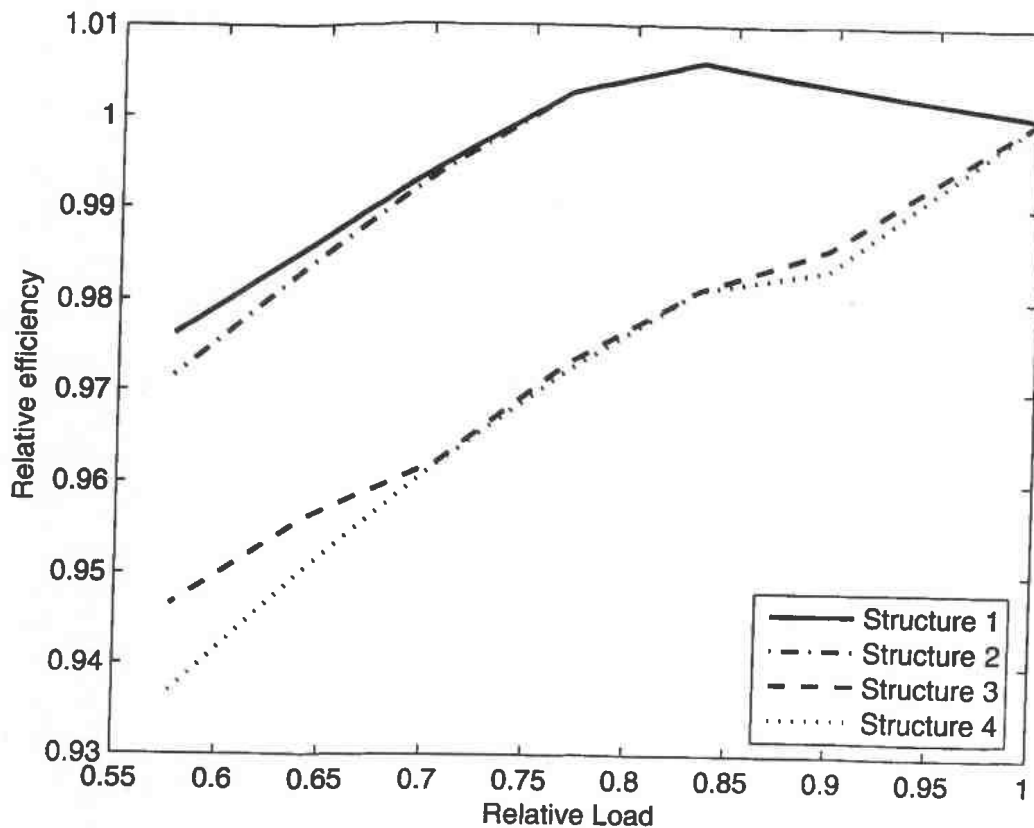$$\eta = \frac{\dot{W}_{out}}{\dot{m}_{CH_4} \cdot \text{LHV}},$$

Figure 2. Optimal steady state combined cycle efficiency for the four different control structures when operating at part load.

where LHV is the lower heating value of CH_4 and $\dot{W}_{out}$ is the total power production from both the low pressure turbine and the steam bottoming cycle. This means that losses in efficiency due to O_2-production and compression of CO_2 before storage are not considered.

For each of the control structures, Figure 2 shows the optimal steady state operating point for a given part load. As one can see the efficiency at part load is higher if one can vary the pressure in the HRSG. It can be seen that with control structure S1 and S2 the process can be operated at its optimal efficiency down to about 75% part load (the relative efficiency is larger than one in this region because there has not been put much effort on optimization of the design point). In fact these solutions are identical in this region because it is not optimal for structure S1 to use VGV. At 75% part load the 0.8 bar constraint becomes active. Then the pressure ratio in the compressor and turbines decreases, which results in a less efficient cycle. If the pressure constraint can be reduced, the range of optimal relative efficiency can be widened. The figure also shows that there is a small gain in efficiency when VGV are used. This gain is very dependent on the relation between TET and steam turbine power output in the HRSG. Due to the uncertainty related to the estimation of the power production in the HRSG, it is difficult to arrive at a reliable conclusion regarding whether use of VGV is beneficial or not. However, keeping TIT constant with use of VGV also has the benefit that large temperature gradients in the system can be avoided.

4. Control and closed loop simulations

The four control structures are implemented with linear MPC. We will first give a short introduction to the MPC used, and then present closed loop simulations.

4.1. *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 main advantages of MPC is that it handles multivariable control problems, and that the controller can take actuator limitations and process constraints into account in a simple and structured manner. This makes predictive control well suited for this process.

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. We have used TET, rotational speed of the compressor, $\dot{W}_{LPT}$, temperature and pressure at outlet of the HRSG, the position of VGV and the state of the condenser (the integral error of the PI controller controlling the outlet temperature of the condenser) as measured variables (y). 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 compensate for model mismatch when operating far from the linearization point. The

linear model is obtained using the LINEARIZE-function of gPROMS at the design operating point. We assume linear constraints on states (or controlled outputs), input and input rate,

$$Ex_k \le b_z, \quad Fu_k \le b_u, \quad G(u_k - u_{k-1}) \le b_{\Delta u}.$$

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

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

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 TIT (1600K), the limits on pressure in HRSG and the constraints on valve operation (opening between 0 and 1, stroke time 15s). We used $H_p = H_u = 50$, with sample time 0.5s.

4.2. *Closed loop simulations*

To compare the dynamic properties of the control structures proposed in Section 3. closed loop simulations using the MPC in Section 4.1 were performed for each structure.
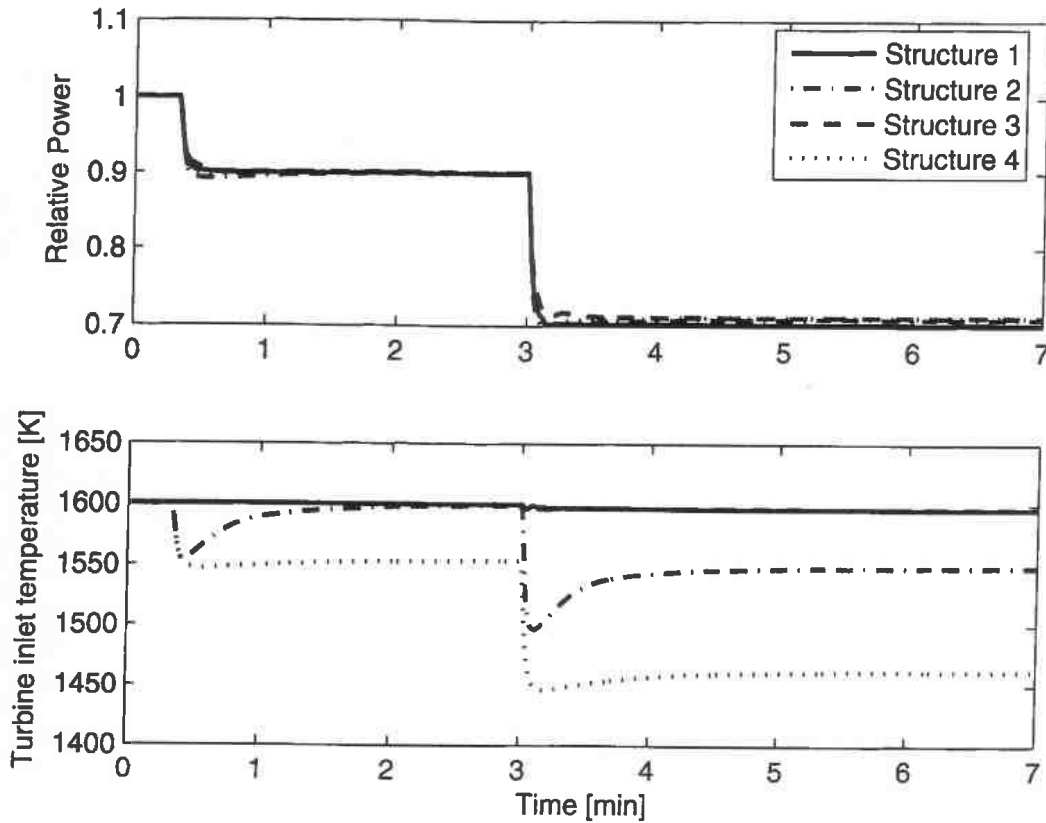

Figure 3.   Relative Combined cycle power and TIT for the four different control structures during load changes.

[1]The norm $\| \cdot \|_H$ is defined by $\| z \|_H = \sqrt{z^T H z}$, $H > 0$.
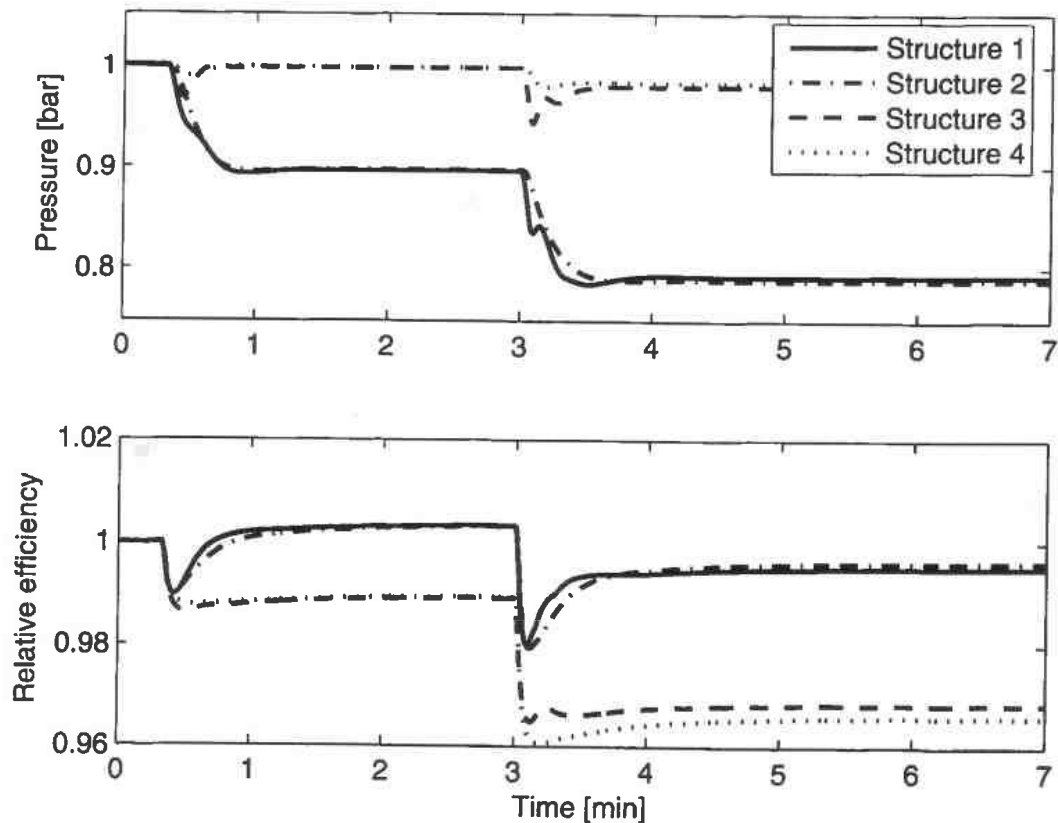
Figure 4.   Pressure in HRSG and Combined cycle efficiency for the four different control structures when operating at part load.

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 closed loop simulations are shown in Figures 3–4. The first disturbance (at 20s) is a change in relative power from 1 to 0.9, and at three minutes there is a new load change from 0.9 to 0.7. Note that all the different control structures distribute the load between the steam cycle and LPT differently, and it is necessary to calculate the setpoint for the LPT load (which is the controlled variable) for each structure independently. The power control is very fast and load changes are done in a few seconds. Also note the large differences in TIT between the different structures. A combined cycle efficiency plot shows that after the second load change, the efficiencies are higher than what was found in Figure 2. The reason for this is the steady state offset in pressure (the constraints are violated by a small margin) which causes a more efficient cycle. As long as we cannot measure TIT, there will be steady state offsets when operating this far from the linearization point. Remedies for this might be to use a nonlinear model or to include constraints in the state estimator.

5. Discussion

Modeling: The developed model (see Imsland *et al.* (2004)) 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.

VGV: From the results presented it is difficult to conclude whether VGV should be used or not. The use of VGV is very dependent on the relation between TET and the steam turbine power output. In (Kehlhofer *et al.*, 1999) there is a plot relating steam turbine output to the gas turbine exhaust temperature. In this plot the relations between the steam turbine power and TET is much steeper than what we found using GTPRO. If we had chosen to use this relation instead, the benefits of using VGV would have been much larger.

HRSG pressure: In the results presented, better part load performance was achieved when the pressure in the HRSG was allowed to go below atmospheric pressure. However, if underpressure is undesirable, it should be possible to design the process such that the HRSG design must deal with over pressure instead. The process can be designed to have a higher pressure in the HRSG when operating at the design point. Then when the power plant operates at part load, the pressure can be allowed to decrease till atmospheric pressure is reached. The cost of this is that the oxygen and natural gas must be compressed to a higher pressure before entering the combustion chamber. At the same time less energy is needed to compress CO$_2$ for storage, because the exhaust will be at a higher pressure.

6. Conclusion

The results show that if the pressure in the HRSG can be allowed to vary, a large gain in combined cycle efficiency can be achieved when operating at part load. But to allow for this kind of control, wall thickness in the heat exchangers must be increased, which will increase the development costs for the plant. When deciding whether to keep the low pressure part of the power plant constant at atmospheric pressure or not, it is necessary to consider how the power plant will be operated. Many gas turbines are used to manage peaks in the power demand, and operate at part load for a considerable amount of their lifetime. In such cases the extra costs should at least be considered.

For the two control structures that used VGV to keep a constant TIT, there was not a significant gain in efficiency. Thus, to be able to draw more firm conclusions, it is necessary with a more accurate model of the HRSG.

The closed loop simulations showed that all the different control structures are feasible, and confirm the results found from the off-line optimizations. The control structures that use VGV have the benefit that large temperature changes in the cycle can be avoided.

Acknowledgements

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