Design and off-design analyses of a pre-combustion CO_2 capture process in a natural gas combined cycle power plant

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Abstract

In this study, a cycle designed for capturing the greenhouse gas CO_2 in a natural gas combined cycle power plant has been analyzed. The process is a pre-combustion CO_2 capture cycle utilizing reforming of natural gas and removal of the carbon in the fuel prior to combustion in the gas turbine. The power cycle consists of a H_2–fired gas turbine and a triple pressure steam cycle. Nitrogen is used as fuel diluent and steam is injected into the flame for additional NO_x control. The heat recovery steam generator includes pre-heating for the various process streams. The pre-combustion cycle consists of an air-blown auto thermal reformer, water-gas shift reactors, an amine absorption system to separate out the CO_2, as well as a CO_2 compression block. Included in the thermodynamic analysis are design calculations, as well as steady-state off-design calculations. Even though the aim is to operate a plant, as the one in this study, at full load there is also a need to be able to operate at part load, meaning off-design analysis is important. A reference case which excludes the pre-combustion cycle and only consists of the power cycle without CO_2 capture was analyzed at both design and off-design conditions for comparison. A high degree of process integration is present in the cycle studied. This can be advantageous from an efficiency stand-point but the complexity of the plant increases. The part load calculations is one way of investigating how flexible the plant is to off-design conditions. In the analysis performed, part load behavior is rather good with efficiency reductions from base load operation comparable to the reference combined cycle plant.

Key words: Carbon capture and storage (CCS), CO_2 capture, Pre combustion capture, Off-design analysis, Process simulation

1 Introduction

Levels of atmospheric carbon dioxide, methane, and other greenhouse gases are on the rise and are contributing to the warming of the atmosphere due to the greenhouse effect. Natural causes can only explain part of this global warming. Fossil fueled power generation, transportation, industrial processes, and other man-made greenhouse gas emission sources add to the picture, mainly because of CO_2 emissions. Out of the energy related carbon dioxide emission sources, the power generation sector is the largest emitter (International Energy Agency, 2006). Thus, if one tries to control and limit the emission of greenhouse gases and thereby attenuating the rise in atmospheric temperature, CO_2 capture from fossil fuel power plants can be a viable path. Among the fossil fuels, the capture of the carbon from coal is attracting the main attention because of the high carbon dioxide emissions per kilowatt hour of electricity and the abundance of coal-fired plants in the world. However, for

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Norway, with large natural gas reserves and the planned and already built gas-fired power plants in the country, CO_2 capture from such plants will be important.

The methods for capture of CO_2 from fossil fuel power generation sources can be divided into three main categories:

1) Post-combustion capture, where the CO_2 is captured at the tail end of the plant from the flue gases, i.e., after the combustion (Chapel and Mariz, 1999). Capture of CO_2 from the flue gases of a power plant may be the best option for capture retrofits of existing power plants. It is also a viable option for new plants. The currently preferred option is capture by absorption processes based on chemical solvents and have been implemented in a number of pilot projects world-wide for CO_2 capture purposes, for example, the Castor pilot project in Denmark (Le Thiez et al., 2004; Knudsen et al., 2006), and the Boundary Dam pilot plant in Canada (Wilson et al., 2004).

2) Pre-combustion capture, where the fossil fuel is used for producing a syngas and the carbon (as CO_2) is separated out before the combustion takes place. The fuel for the combustion mainly consists of hydrogen mixed with a diluent, such as, nitrogen or steam. An existing technology for power plant applications, the integrated gasification combined cycle (IGCC), could be attractive as part of a coal based pre-combustion CO_2 capture method (Bohm et al., 2007).

3) Oxy-fuel combustion, where the oxidizer for the combustion is oxygen instead of air. The combustion products are mainly carbon dioxide and steam, and the CO_2 can be separated out by condensing the steam. Many proposals for cycle configurations have been suggested in the oxy-fuel category. Exam-

ples include the Graz cycle (Jericha et al., 2004), the Matiant cycle (Mathieu and Nihart, 1999), the advanced zero emissions power plant (Griffin et al., 2005), and chemical looping combustion (Richter and Knoche, 1983; Ishida and Jin, 1994).

This study focuses on the pre-combustion approach. More specifically, pre-combustion capture utilizing an air-blown auto thermal reformer (ATR) in a natural gas fueled combined cycle (NGCC) plant. Similar process configurations have been studied by Andersen et al. (2000); Lozza and Chiesa (2002a,b); Corradetti and Desideri (2005); Ertesvåg et al. (2005). Their results from heat and mass balance analyses show lower heating value (LHV) net plant efficiencies ranging from approximately 46% to 49%. Another possibility for this type of plant is to utilize it for co-production of hydrogen and electricity (Consonni and Viganò, 2005); however, the focus of this paper is on power production only. Kvamsdal et al. (2007) performs comparative heat and mass balance simulations for a number of CO_2 capture cycles including pre-combustion cases. The cited studies focus on design case analysis. Little is found in the literature in terms of off-design analysis of CO_2 capture cycles. Part load analyses of post-combustion systems are performed for coal cycles by Chalmers and Gibbins (2007) and for natural gas cycles by Möller et al. (2007). Haag et al. (2007) and Naqvi et al. (2007) analyze the part load behavior of some of the proposed oxy-fuel cycles. For NGCC pre-combustion plants no off-design publications have been found by the author.

The remainder of the paper is divided into the following sections: Section 2 describes the process where the details of the cycle are explained. Section 3 describes the methodology and lists the assumptions used in the study. The results are shown and analyzed in Section 4 and concluding remarks are given

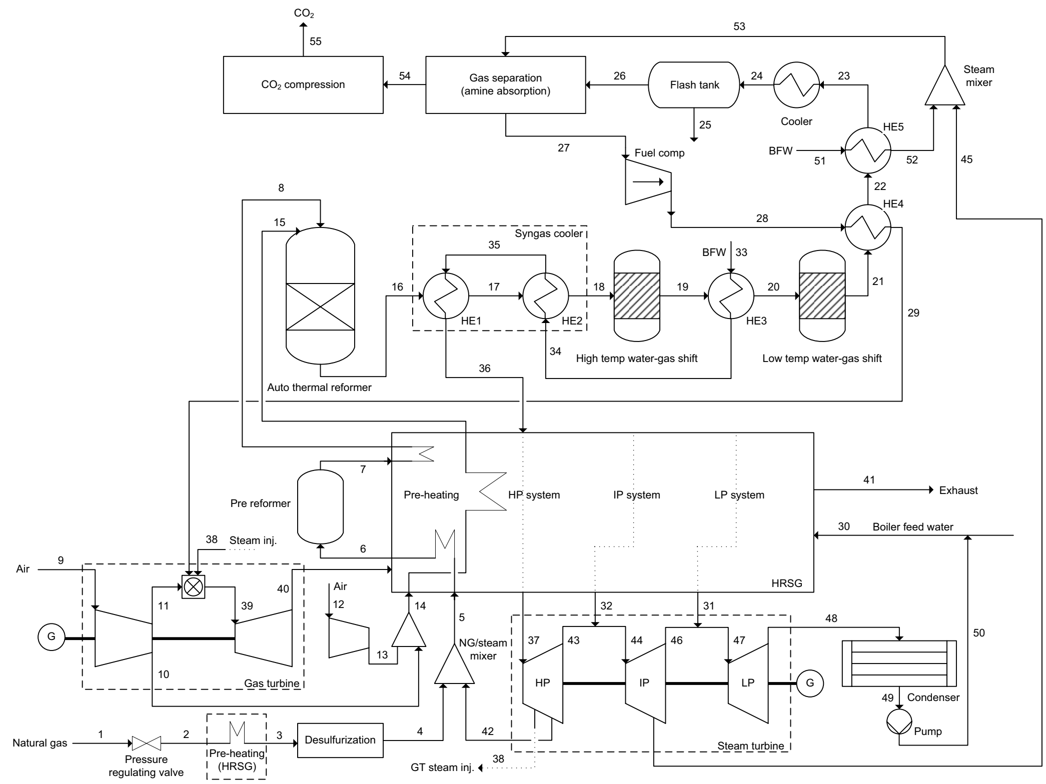

Fig. 1. Pre-combustion process flow sheet.

in Section 5.

2 Process description

The selected process for the work is a pre-combustion CO_2 capture cycle in a natural gas combined cycle power plant as shown in Fig. 1. The power cycle consists of a General Electric (GE) 9FA H_2–fired gas turbine (GT) and a triple pressure steam cycle. The heat recovery steam generator (HRSG) includes pre-heating for the various process streams. The pre-combustion cycle consists of a pre-reformer, an air-blown auto thermal reformer, two water-gas shift reactors, a gas separation stage in form of amine absorption to separate out the CO_2, as well as a CO_2 compression block.

As mentioned, the fuel input to the process is natural gas (stream 1 in Fig. 1). After the natural gas has been regulated down to system pressure (stream 2), pre-heated to 400°C (3), and desulfurized (4), it is mixed with steam (5) before another pre-heating section (500°C) and introduced to the pre-reformer (6). The steam to carbon ratio (S/C) is set at 1.5. In the pre-reforming reactor the hydrocarbons higher than methane are converted to protect against coking in the primary reformer according to reactions (1) and (2).

$$C_xH_y + xH_2O_{(g)} \rightleftharpoons xCO + (x + \frac{y}{2})H_2 \qquad -\Delta H^0_{298} < 0\, kJ/mol \tag{1}$$

$$CO + 3H_2 \rightleftharpoons CH_4 + H_2O_{(g)} \qquad -\Delta H^0_{298} = 206\,\mathrm{kJ/mol} \tag{2}$$

Also, the exothermic water-gas shift reaction (3) converting the CO into CO_2 occurs to some degree in the pre-reforming reactor.

$$CO + H_2O_{(g)} \rightleftharpoons CO_2 + H_2 \qquad -\Delta H^0_{298} = 41\, kJ/mol \tag{3}$$

Before entering the ATR the stream from the pre-reformer (7) is again pre-heated to 500°C (8). Also, air extracted from the compressor discharge stream of the gas turbine (10) combined with an additional compressor air stream (13) is pre-heated and supplied to the ATR (15). The external compressor is introduced in order to better utilize the operation of the gas turbine. If too much air is removed prior to the combustion chamber in the gas turbine the effect on performance and temperature profile can be negative. With the additional compressor another degree of freedom is attained and the gas turbine can be utilized in a more efficient manner. In the ATR the exothermic reaction (4) provide heat to the endothermic reaction (5).

$$CH_4 + \frac{1}{2}O_2 \rightarrow CO + 2H_2 \qquad -\Delta H^0_{298} = 36\, kJ/mol \tag{4}$$

$$CH_4 + H_2O_{(g)} \rightleftharpoons CO + 3H_2 \qquad -\Delta H^0_{298} = -206\, kJ/mol \tag{5}$$

As in the pre-reformer the water-gas shift reaction (3) converts some of the CO into CO_2. Further on, the syngas is cooled in the syngas cooler before entering the water-gas shift reactors where most of the remaining CO is converted into CO_2 according to reaction (3). The reasons behind dividing the water-gas shift reaction into a high temperature reactor and a low temperature one are due to conversion rate and catalysts. To get a higher degree of conversion of the CO to CO_2, two reactors are favorable compared to a one-reactor setup. Also, there is a need for a more active catalyst at the lower region of the temperature range (Moulijn et al., 2007). It can therefore make sense to use a standard catalyst at the higher temperature range and then have a separate reactor with a more active catalyst for the low end temperature. Heat exchanger 3 (HE3) and the syngas cooler are utilized for producing high-pressure saturated steam to be added to the high-pressure superheater in the HRSG. The reason for not superheating the steam in the heat exhanger is because of the risk of metal dusting (Grabke and Spiegel, 2003). Heat exchanger 4 (HE4) is used to preheat the fuel to the gas turbine to 200°C (29). In this model the pre-combustion capture (Gas separation) is using the chemical absorbent activated MDEA (Zhang et al., 2003; van Loo et al., 2007) and is modeled as a 'black box'. Assumptions for the capture section include a CO_2 capture rate of 95% and the heat required for the stripper reboiler at 1.5 MJ/kg CO_2. Heat exchanger 5 (HE5) is producing some of the steam necessary for the reboiler in the amine absorption process. The CO_2 (54) is passed on to the compression section where the gas is compressed in the four compressor/intercooler stages and excess water is removed. To achieve the exit pressure of 110 bar a pump is utilized at the end of the compression train.

From the gas separation stage the fuel mix (27) is passed on to the gas turbine via a fuel compressor. In principle, the fuel consists of an H_2/N_2 mixture. The N_2 diluent is used to be able to operate with available IGCC-type combustors in the gas turbine. For further NO_x control, steam is injected into the flame. In addition to running the GT on a hydrogen based fuel, the idea is to be able to operate on natural gas if the pre-combustion process is shut-down and during plant start-up. This requires fuel flexibility for the combustor system (Tomczak et al., 2002; Shilling and Jones, 2003; Moliere, 2005). The gas turbine exhaust stream (40) passes through the HRSG for pre-heating of process streams and steam generation before emitted to the atmosphere through the stack (41).

The steam cycle is designed for pressure levels of approximately 83/10/3 bars for the high, intermediate, and low pressure (HP/IP/LP) systems respectively. The pre-heating makes the HRSG design more complex and a lot of heat is removed from the gas stream at the hot part of the HRSG due to the high temperature requirements of some of the process streams. Note that the pre-heating is not entirely in the hot end of the HRSG but instead inter-mixed with the low, intermediate, and high-pressure sections. The steam turbine (ST) has extractions for the GT steam injection (38), the reforming process steam (42), and for the reboiler in the amine absorption system (45). After exiting the last low pressure turbine stage (48) the steam is condensed in the direct seawater cooled condenser (49). The condenser pressure is assumed at 0.04 bar.

There are certainly many configuration options for a plant like this. For example, one could operate the system at a higher pressure by boosting the air pressure from the gas turbine compressor discharge with an additional compressor. In this way a fuel compressor would not be necessary. The impact of this option was investigated by Andersen et al. (2000) where it was concluded that operating at a lower system pressure and having a fuel compressor improves the overall efficiency for the cycle in their study. This effect was due to the need for extra process stream pre-heating in the elevated pressure case since the air was cooled before the compression to minimize compressor work. Other configuration options include utilizing an oxygen-blown ATR with an air separation unit (ASU) for the oxygen supply. Or using a steam reformer instead of the ATR. Configurations with less integration between the power cycle and syngas process could also be attractive. For the power cycle one could employ a more recent gas turbine model as for example the GE 9FB type with a higher turbine inlet temperature (TIT) and cycle efficiency. The steam cycle could include a reheat cycle leading to a higher efficiency but also more complexity. For the capture section one could use other absorbants, such as, hot potassium carbonate.

A reference case which excludes the pre-combustion cycle and only consist of the power cycle without CO_2 capture was analyzed at both design and off-design conditions for comparison. The reference case consists of the same type GE 9FA gas turbine but is instead of the IGCC combustor using a regular pre-mix natural gas combustor without steam injection. The steam cycle is again triple pressure without reheat.

3 Methodology

This section provides details into the process models simulated in the study. Assumptions for the design case analysis are described in Section 3.1. Included in the thermodynamic analysis are steady-state off-design calculations, that is, analysis when the plant is operating at part load. In a scenario where

CO_2 capture plants become common-place, part load operation will be an important part of the operation scheme. For a plant such as the one modeled in this work the goal is certainly to run it at base load operation for the majority of the time but as part of an overall grid strategy part load operation will come into play. Assumptions for the part load cases are described in Section 3.2.

The pre-combustion cycle, including the pre-heating section, was modeled with Aspen HYSYS. The property package was modeled with the Kabadi-Danner equation of state. The Kabadi-Danner is a modification of the Soave-Redlich-Kwong equation of state to take into account hydrocarbon solubility in the water phase. The power cycle was modeled with GT PRO for the design case and GT MASTER for the off-design cases. For the steam properties in GT PRO/GT MASTER the IAPSW-IF97 formulation was used (Wagner et al., 2000).

3.1 Design model assumptions

The selected gas turbine is a GE 9FA from the model library of GT PRO version 17. Steam is injected into the flame for NO_x control at a rate of 20% of the fuel mass flow. The GT turbine inlet temperature has been reduced because of the high steam content in the turbine. The hydrogen fuel together with the injected steam lead to an H_2O content entering the turbine of about 18.2 vol%. This leads to a higher heat transfer rate to the blades compared to a natural gas fired turbine. As a result, the metal temperature of the turbine blades is higher for the same turbine inlet temperature as in a conventional gas turbine. To obtain similar life of the turbine parts, the turbine inlet temperature reduction is necessary. Chiesa et al. (2005) report TIT decreases of 10-34

K for hydrogen combustion with nitrogen or steam diluent (VGV operation cases). As a model assumption, a TIT reduction of 30 K has been assumed for this work. The inlet filter pressure drop is set to 10 mbar and the total exhaust losses (GT exhaust and HRSG) to 25 mbar. The maximum allowable GT power output is increased from 260 to 286 MW (IGCC setup). Air from the compressor discharge is re-directed to the reforming section at a rate of 75 kg/s. This is approximately 12% of the GT inlet air flow. Addtional air required for the reforming is supplied by an external (to the GT) compressor with a polytropic efficiency of 85%. A polytropic efficiency of 85% is also assumed for the fuel compressor for the hydrogen-rich fuel.

The high-pressure steam is set to 83 bar at 568°C before the stop valve to the steam turbine. The intermediate-pressure level is 10.3 bar and the LP drum pressure is 2.8 bar. The pinch point temperature difference is assumed to be 10 K for all three pressure levels. The subcooling approach temperature difference at the exit of the economizers is assumed at 5 K.

The natural gas composition (stream 1) is listed in Table 2 with the exception of the H_2S content which is set to be 5 ppmvd. The sulfur is removed in the desulfurizer unit, which is modeled as a separator. The air composition (9) is also listed in Table 2. The ambient pressure is assumed to be 1.013 bar with a temperature of 15°C and a relative humidity of 60%.

The pressure drops in the pre-reformer and ATR are set at 5% of the inlet pressure. The tube side pressure drop in the heat exchangers modeled in HYSYS is set to be 0.5 bar (approximately 3% of inlet pressure) with the exception of HE3-HE5 which each has an assumed pressure drop of 0.85 bar due to two shell passes compared to one shell pass for the other heat exchangers.

The shift reactors are modeled with a 0.5 bar pressure drop. The pre-reformer and the water-gas shift reactors are modeled as equilibrium reactors. A Gibbs reactor model is used for the ATR.

A splitter is used for the amine absorption section model. The reboiler duty is set to 1.5 MJ/kg CO_2 and the total pump work is assumed to be 0.16 MJ/kg CO_2. The reboiler temperature is set to 120°C. A 95% capture rate is assumed for the absorption system.

Polytropic efficiencies for the CO_2 compression train are assumed at 85%, 80%, 80%, and 75% for the four compressor stages respectively (listed in flow direction). The pump that pressurizes the CO_2 stream to the end pressure of 110 bar is assumed to have an adiabatic efficiency of 75%.

3.2 Off-design model assumptions

The selected part load points are 60% and 80% of the design case gas turbine load. The reason for selecting the relative part load points as a function of gas turbine load is because the GT dictates the overall plant load. By changing the GT load, the steam cycle, as well as the pre-combustion process, will follow. Gas turbine part load operation commonly employs variable inlet guide vanes (VIGV). This is the case for the GE 9FA which has one row of variable guide vanes where the flow angle entering the first stage of the compressor can be varied. The VIGV operation allows reduction of the air flow and the turbine exhaust temperature can remain high at part load operation. The high exhaust temperature means the part load combined cycle efficiency can be maintained at a high level. However, at the lower part load range the cycle efficiency drops off quicker. The steam cycle part load operating concept involves sliding pressure operation with fully open steam valves down to approximately 50% steam turbine load (Kehlhofer et al., 1999). At lower loads the operating concept is based on fixed steam pressure operation by closing of the steam valves. This leads to throttling losses in the ST inlet valves. These factors combined may suggest that it does not make sense to operate a plant, such as the one in the study, at a much lower GT load than 60%. Certainly, the plant still has to be able to operate at lower part load points, not the least during transients such as start-ups and shut-downs; however, transient analysis is not covered in this study.

All the hardware in the off-design cases are identical to the design case. This also means that the extractions of the steam turbine are set. Since the part load operation is with sliding pressure operation of the steam cycle the steam pressures at the extraction points will decrease. In the case of the steam for the reboiler in the amine absorption system the design case was actually "over-designed" to allow for a sufficient steam pressure (and hence a sufficiently high condensation temperature) for the part load cases.

The turbine inlet temperature reduction was removed for the off-design simulations since the temperature was decreased anyway for part load operation at the 80% and 60% relative load levels. The air extraction from the compressor discharge was decreased to 60 kg/s (approximately 11% of GT inlet air flow) for the 80% case and 45 kg/s (approximately 10% of GT inlet air flow) for the 60% case. The fuel compressor exit pressure is assumed constant from the design case.

In the design case the inlet temperatures to the desulfurization unit, the reforming reactors, and the water-gas shift reactors were fixed. For the off-design calculations these constraints were removed. Instead, for each part load case a check was performed to see if the inlet temperatures were within the operational window of each reactor. Based on the resulting inlet temperatures it was not necessary to use by-pass valves for the various heat exchangers at the steady-state part load cases simulated (although likely needed during lower part load and start-up and shut-down).

For the analysis of the various heat exchangers a correction of the heat transfer coefficient was done based on the gas massflow. The correction is based on course literature from Bolland (2006) as displayed in Equation (6).

$$\frac{U}{U_{design}} = \left(\frac{\dot{m}_{gas}}{\dot{m}_{gas,design}}\right)^{m} \tag{6}$$

U is here the heat transfer coefficient, $\dot{m}_{gas}$ the gas massflow, and m a constant. For a staggered tubes configuration with assumed tube pitches of 2.5 (Incropera and DeWitt, 1990):

$$\left.\begin{aligned} \frac{S_T}{D} &= 2.5 \\ \frac{S_L}{D} &= 2.5 \end{aligned}\right\} \Rightarrow m \simeq 0.57$$

S_T is the transverse pitch, that is, the distance 90° off from the flow direction between the centers of two adjacent tubes. S_L is the longitudinal pitch, that is, the distance in flow direction between the centers of two adjacent tubes. D is the tube diameter in the heat exchanger. In HYSYS there is the option to lock in the UA specification for a heat exchanger. Since the area A is constant one could re-write Equation (6) as:

$$UA = U_{design} A \left(\frac{\dot{m}_{gas}}{\dot{m}_{gas,design}}\right)^{0.57} \tag{7}$$

A similar expression, the exception being the m-factor which was set at 0.6, was used by Haag et al. (2007).

4 Results

The main results are summarized in Table 1. Included in the table is the power consumption for the air compressor (external to GT), the fuel compressor, the CO_2 compression, the pump work in the amine absorption system (gas separation pumps), as well as the additional boiler feed water pumps in the pre-combustion system, and the remaining plant auxiliaries. The auxiliaries post in Table 1 includes, among other items, the regular boiler feed water pumps and the cooling water pumps.

The design case LHV based cycle efficiency is 41.9% with a net power output of approximately 362 MW. The net power output is here defined as the gross power output at the generator terminals minus the power needed for air compression, fuel compression, CO_2 compression, pump work, and auxiliaries, as displayed in Table 1. The cycle efficiency is the net power output divided by the natural gas lower heating value input. The design case results should be compared to the reference case net power output of approximately 385 MW and efficiency of 55.9% leading to a capture efficiency penalty of approximately 14%-points. The calculated design case cycle efficiency is low and the capture efficiency penalty high compared to the literature (Andersen et al., 2000; Lozza and Chiesa, 2002a,b; Corradetti and Desideri, 2005; Ertesvåg et al., 2005). This can be explained to a large degree by the practical considerations included in this work. For one, steam is injected into the gas turbine for NO_x control which lowers the overall efficiency. Also, the turbine

Table 1

Summary of results for design case (100%), off-design cases (80% and 60%), and reference cases (100% ref., 80% ref., and 60% ref).

	100%	100% ref.	80%	80% ref.	60%	60% ref.
Natural gas LHV input [MW]	865.2	689.1	729.8	599.0	588.2	501.0
Gross power output GT [MW]	277.0	253.5	221.6	204.0	166.2	153.8
Gross power output ST [MW]	137.6	137.2	122.5	127.6	103.8	113.7
Gross power output [MW]	414.6	390.7	344.1	331.6	270.0	267.5
Gross power output [% of LHV input]	47.9	56.7	47.1	55.4	45.9	53.4
Air compression [MW]	8.2	-	7.9	-	6.9	-
Air compression [% of LHV input]	0.9	-	1.1	-	1.2	-
Fuel compression [MW]	13.6	-	14.7	-	17.0	-
Fuel compression [% of LHV input]	1.6	-	2.0	-	2.9	-
CO_2 compression [MW]	17.7	-	15.0	-	12.2	-
CO_2 compression [% of LHV input]	2.0	-	2.1	-	2.1	-
Gas separation pumps [MW]	7.6	-	6.4	-	5.2	-
Gas separation pumps [% of LHV input]	0.9	-	0.9	-	0.9	-
BFW pumps in pre-comb process [MW]	1.0	-	0.8	-	0.5	-
BFW pumps in pre-comb process [% of LHV input]	0.1	-	0.1	-	0.1	-
Auxiliaries [MW]	4.5	5.4	4.4	5.3	4.3	5.2
Auxiliaries [% of LHV input]	0.5	1.4	0.6	1.6	0.7	1.9
Net power output [MW]	**362.2**	**385.3**	**294.9**	**326.3**	**223.8**	**262.3**
Net plant efficiency [% of LHV input]	**41.9**	**55.9**	**40.4**	**54.5**	**38.0**	**52.4**
Efficiency capture penalty [%-point loss to ref. case]	14.0	-	14.1	-	14.3	-
CO_2 emissions [g CO_2/net kWh el.]	33.2	380.1	30.7	390.1	29.3	405.9
CO_2 capture rate [%]	**93.4**	**0**	**94.1**	**0**	**94.7**	**0**

inlet temperature is decreased by 30 K which further will bring the efficiency down. In addition, for the design case, considerations were taken of the part load scenarios. For example, a steam turbine extraction had to be taken at a higher than necessary pressure during design case analysis to have sufficient pressure also at the off-design cases. This also has a negative effect on the design case plant efficiency.

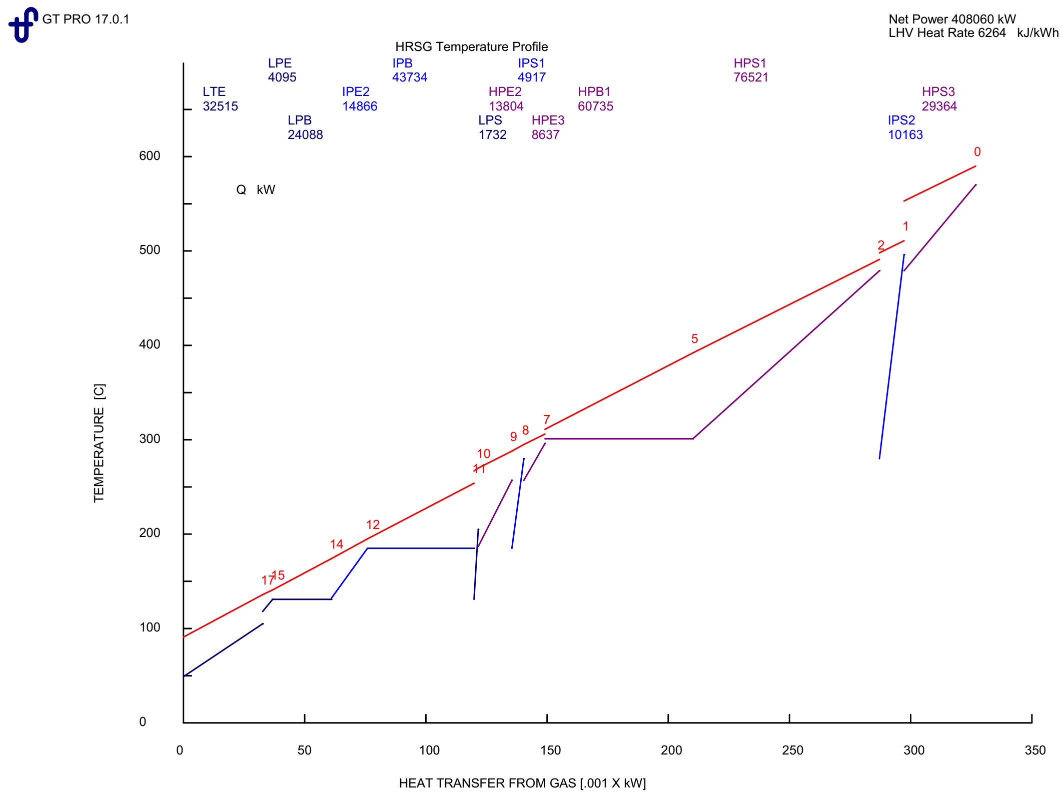

Fig. 2. GT PRO T-Q diagram for heat recovery steam generator.

The HRSG has a different design than would be present in a typical NGCC plant. A large portion of the heat in the GT exhaust gases are utilized in the pre-heating and in the HP superheaters, as displayed in Fig. 2. Because of the saturated steam introduced from the syngas cooler the massflow to the high-pressure superheaters are more than three times as high as the massflow in the HP boiler. The vertical gas temperature jumps in the T-Q diagram represent the pre-heating sections in the HRSG.

The off-design calculations resulted in net plant efficiencies of 40.4% and 38.0% for the 80% and 60% load cases respectively. The capture penalties for the part load cases are very similar to the design case, that is, around 14%-points.

The CO_2 capture rate varies between 93% and 95% for the different cases, with CO_2 emissions of 29-33 g/net kWh electricity. The CO_2 capture rate is defined as the fraction of formed CO_2 that is captured.

Stream data for the design case is displayed in Table 2, for the 80% load case in Table 3, and for the 60% load case in Table 4.

5 Conclusions

The pre-combustion NGCC cycle is a system well worth studying. Advantages include the reduced size of the capture system and the increased CO_2 partial pressure compared to post-combustion capture. A post-combustion capture system would have to deal with separating out CO_2 from flue gases with very large flow rates at a low pressure. Disadvantages compared to post-combustion capture include conversion losses in the natural gas reforming process. Another advantage for a post-combustion capture system is that natural gas fired gas turbines are a more mature product than hydrogen fired ones. Pre-mix combustion with low NO_x emissions is one of the advantages of a standard GT fired with natural gas. In the case of the hydrogen diffusion combustion system, diluents such as steam and/or nitrogen are necessary. In this study, nitrogen was used as diluent and steam was injected directly into the flame in the combustor.

A high degree of process integration is present in the cycle studied. This can be advantageous from an efficiency stand-point but the complexity of the plant increases. This is exemplified in the HRSG where several of the process streams are pre-heated and high-pressure steam are introduced from the syngas cooler to the HP superheaters. The heat from the syngas is used for the economizing and boiling of the high-pressure water. This heat integration increases the cycle

Table 2

Stream results for the design case.

No.	T	p	$\dot{m}$	MW	CH_4	C_2+	H_2	CO	CO_2	H_2O	O_2	N_2	Ar
	(°C)	(bar)	(kg/s)	(kg/kmol)	(vol%)	(vol%)	(vol%)	(vol%)	(vol%)	(vol%)	(vol%)	(vol%)	(vol%)
1	16.0	31.00	19.0	20.73	79.84	16.72	-	-	2.92	-	-	0.51	-
3	400.0	17.68	19.0	20.73	79.84	16.72	-	-	2.92	-	-	0.51	-
5	371.3	17.68	49.5	18.97	28.01	5.86	-	-	1.03	64.91	-	0.18	-
6	500.0	17.18	49.5	18.97	28.01	5.86	-	-	1.03	64.91	-	0.18	-
7	451.4	16.32	49.5	17.30	35.20	0.00	8.77	0.12	5.21	50.53	-	0.16	-
8	500.0	15.82	49.5	17.30	35.20	0.00	8.77	0.12	5.21	50.53	-	0.16	-
9	15.0	1.01	629.3	28.86	-	-	-	-	0.03	1.01	20.74	77.29	0.93
10	394.0	16.35	75.0	28.85	-	-	-	-	0.03	1.02	20.73	77.29	0.92
11	394.0	16.35	483.9	28.86	-	-	-	-	0.03	1.01	20.74	77.29	0.93
13	436.1	16.35	18.5	28.85	-	-	-	-	0.03	1.02	20.73	77.29	0.92
15	500.0	15.85	93.5	28.85	-	-	-	-	0.03	1.02	20.73	77.29	0.92
16	950.0	15.03	143.0	19.24	0.08	0.00	28.87	10.38	5.16	21.36	0.00	33.74	0.40
18	350.0	14.03	143.0	19.24	0.08	0.00	28.87	10.38	5.16	21.36	0.00	33.74	0.40
19	433.7	13.53	143.0	19.24	0.08	0.00	35.94	3.31	12.23	14.29	0.00	33.74	0.40
20	205.8	12.68	143.0	19.24	0.08	0.00	35.94	3.31	12.23	14.29	0.00	33.74	0.40
21	241.4	12.18	143.0	19.24	0.08	0.00	38.79	0.46	15.08	11.44	0.00	33.74	0.40
26	25.0	9.98	128.1	19.39	0.09	0.00	43.65	0.52	16.96	0.35	0.00	37.97	0.45
29	200.0	20.00	79.9	14.55	0.11	0.00	52.32	0.63	0.64	0.19	0.00	45.56	0.54
31	203.3	2.47	11.1	18.02	-	-	-	-	-	100.00	-	-	-
32	494.8	10.30	21.9	18.02	-	-	-	-	-	100.00	-	-	-
36	301.4	87.62	86.7	18.02	-	-	-	-	-	100.00	-	-	-
37	568.0	83.00	129.4	18.02	-	-	-	-	-	100.00	-	-	-
38	377.0	22.00	16.0	18.02	-	-	-	-	-	100.00	-	-	-
39	1295.0	15.70	579.8	26.73	-	-	-	-	0.37	18.21	9.28	71.28	0.86
40	591.0	1.04	650.1	26.94	-	-	-	-	0.34	16.48	10.43	71.89	0.87
41	90.6	1.01	650.1	26.94	-	-	-	-	0.34	16.48	10.43	71.89	0.87
42	346.0	17.68	30.5	18.02	-	-	-	-	-	100.00	-	-	-
45	227.0	4.00	24.0	18.02	-	-	-	-	-	100.00	-	-	-
53	209.6	4.00	30.5	18.02	-	-	-	-	-	100.00	-	-	-
55	40.9	110.00	47.4	43.89	0.00	0.00	0.12	0.00	99.58	0.25	0.00	0.05	0.00

Table 3

Stream results for the 80% load case.

No.	T (°C)	p (bar)	$\dot{m}$ (kg/s)	MW (kg/kmol)	CH_4 (vol%)	C_2+ (vol%)	H_2 (vol%)	CO (vol%)	CO_2 (vol%)	H_2O (vol%)	O_2 (vol%)	N_2 (vol%)	Ar (vol%)
1	16.0	31.00	16.0	20.73	79.84	16.72	-	-	2.92	-	-	0.51	-
3	397.5	17.68	16.0	20.73	79.84	16.72	-	-	2.92	-	-	0.51	-
6	494.9	15.56	41.8	18.97	28.01	5.86	-	-	1.03	64.91	-	0.18	-
8	494.2	14.28	41.8	17.30	35.21	0.00	8.76	0.11	5.22	50.53	-	0.16	-
9	15.0	1.01	535.6	28.86	-	-	-	-	0.03	1.01	20.74	77.29	0.93
10	377.0	13.91	60.0	28.85	-	-	-	-	0.03	1.02	20.73	77.29	0.92
15	489.3	13.41	79.5	28.85	-	-	-	-	0.03	1.02	20.73	77.29	0.92
18	334.9	11.70	121.3	19.27	0.06	0.00	28.74	10.34	5.18	21.38	0.00	33.90	0.40
20	195.4	10.35	121.3	19.27	0.06	0.00	36.03	3.06	12.46	14.10	0.00	33.90	0.40
21	229.2	9.85	121.3	19.27	0.06	0.00	38.71	0.37	15.15	11.41	0.00	33.90	0.40
29	200.4	20.00	67.8	14.57	0.08	0.00	52.24	0.50	0.65	0.20	0.00	45.78	0.55
31	207.5	2.21	8.0	18.02	-	-	-	-	-	100.00	-	-	-
32	486.2	9.26	18.9	18.02	-	-	-	-	-	100.00	-	-	-
36	297.5	82.88	74.1	18.02	-	-	-	-	-	100.00	-	-	-
37	568.0	73.69	114.4	18.02	-	-	-	-	-	100.00	-	-	-
39	1270.0	13.35	497.0	26.75	-	-	-	-	0.33	17.98	9.45	71.38	0.86
40	602.0	1.03	556.8	26.96	-	-	-	-	0.30	16.28	10.58	71.98	0.87
45	228.0	3.68	19.9	18.02	-	-	-	-	-	100.00	-	-	-
53	208.3	3.68	25.8	18.02	-	-	-	-	-	100.00	-	-	-
55	40.9	110.00	40.3	43.89	0.00	0.00	0.12	0.00	99.58	0.25	0.00	0.05	0.00

efficiency but the price is paid in the resulting increased plant complexity.

Part load calculations are one way of investigating how flexible the plant is to off-design conditions. In the analysis performed in the study, part load behavior is rather good with efficiency reductions from baseload operation comparable to the reference combined cycle plant. Based on the analysis performed in the paper, it is possible to operate a complex plant like this one at part loads down to 60% GT load and possibly lower. Not included in the part load study are compressor mapping for off-design calculations for the air

Table 4

Stream results for the 60% load case.

No.	T ($^\circ$C)	p (bar)	$\dot{m}$ (kg/s)	MW (kg/kmol)	CH_4 (vol%)	C_2+ (vol%)	H_2 (vol%)	CO (vol%)	CO_2 (vol%)	H_2O (vol%)	O_2 (vol%)	N_2 (vol%)	Ar (vol%)
1	16.0	31.00	12.9	20.73	79.84	16.72	-	-	2.92	-	-	0.51	-
3	400.5	13.22	12.9	20.73	79.84	16.72	-	-	2.92	-	-	0.51	-
6	501.6	12.72	33.7	18.97	28.02	5.86	-	-	1.03	64.90	-	0.18	-
8	497.9	11.58	33.7	17.24	34.89	0.00	9.50	0.12	5.38	49.95	-	0.16	-
9	15.0	1.01	465.3	28.86	-	-	-	-	0.03	1.01	20.74	77.29	0.93
10	355.0	11.84	45.0	28.85	-	-	-	-	0.03	1.02	20.73	77.29	0.92
15	489.6	11.34	63.8	28.85	-	-	-	-	0.03	1.02	20.73	77.29	0.92
18	314.5	9.76	97.4	19.24	0.04	0.00	28.89	10.39	5.16	21.32	0.00	33.79	0.40
20	182.4	8.41	97.4	19.24	0.04	0.00	36.49	2.79	12.76	13.72	0.00	33.79	0.40
21	214.2	7.91	97.4	19.24	0.04	0.00	39.00	0.29	15.27	11.22	0.00	33.79	0.40
29	205.1	20.00	54.3	14.49	0.06	0.00	52.56	0.39	0.65	0.21	0.00	45.58	0.55
31	204.0	1.92	6.4	18.02	-	-	-	-	-	100.00	-	-	-
32	489.6	7.90	15.6	18.02	-	-	-	-	-	100.00	-	-	-
36	286.0	69.98	60.2	18.02	-	-	-	-	-	100.00	-	-	-
37	568.1	62.25	95.5	18.02	-	-	-	-	-	100.00	-	-	-
39	1200.0	11.36	433.5	26.89	-	-	-	-	0.28	16.82	10.25	71.79	0.86
40	590.0	1.03	485.5	27.09	-	-	-	-	0.26	15.23	11.30	72.35	0.87
45	233.0	3.27	14.8	18.02	-	-	-	-	-	100.00	-	-	-
53	207.0	3.27	20.8	18.02	-	-	-	-	-	100.00	-	-	-
55	40.9	110.00	32.7	43.89	0.00	0.00	0.12	0.00	99.58	0.25	0.00	0.05	0.00

compressor, fuel compressor, and CO_2 compression train. Energy requirement changes per kg of CO_2 for the reboiler in the amine absorption system at off-design points are not considered either. Including these details in the model could show a different part load behavior.

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