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A modeling software linking approach for the analysis of an integrated reforming combined cycle with hot potassium carbonate CO_2 capture

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Abstract

The focus of this study is the analysis of an integrated reforming combined cycle (IRCC) with natural gas as fuel input. This IRCC consisted of a hydrogen-fired gas turbine (GT) with a single-pressure steam bottoming cycle for power production. The reforming process section consisted of a pre-reformer and an air-blown auto thermal reformer (ATR) followed by water-gas shift reactors. The air to the ATR was discharged from the GT compressor and boosted up to system pressure by an air booster compressor. For the CO_2 capture sub-system, a chemical absorption setup was modeled. The design case model was modeled in GT PRO by Thermoflow, and in Aspen Plus. The Aspen Plus simulations consisted of two separate models, one that included the reforming process and the water-gas shift reactors. In this model were also numerous heat exchangers including the whole pre-heating section. Air and CO_2 compression was also incorporated into the model. As a separate flow sheet the chemical absorption process was modeled as a hot potassium carbonate process. The models were linked by Microsoft Excel. For the CO_2 capture system the model was not directly linked to Excel but instead a simple separator model was included in the reforming flow sheet with inputs such as split ratios, temperatures, and pressures from the absorption model. Outputs from the potassium model also included pump work and reboiler duty. A main focal point of the study was off-design simulations. For these steady-state off-design simulations GT MASTER by Thermoflow in conjunction with Aspen Plus were used. Also, inputs such as heat exchanger areas, compressor design point, etc., were linked in from the Aspen Plus reforming *design* model. Results indicate a net plant efficiency of 43.2% with approximately a 2%-point drop for an 80% part load case. Another off-design simulation, at 60% load, was simulated with a net plant efficiency around 39%. The CO_2 capture rate for all cases was about 86%, except for the reference case which had no CO_2 capture.

Keywords: Carbon capture and storage (CCS); CO_2 capture; Pre combustion capture; Integrated reforming combined cycle (IRCC); hot potassium carbonate; modeling; process simulation

1. Introduction

Pre-combustion CO_2 capture is one possible route to fossil fueled power generation with low CO_2 emissions. There exist many possible configurations for a pre-combustion plant, not the least in relation to the fuel feed. Eide and Bailey [1] describe and discuss different pre-combustion decarbonization processes. One such process is the integrated reforming combined cycle (IRCC). An IRCC is fueled by natural gas which is reformed to a synthetic gas, mainly consisting of H_2 and CO. The reformed gas is water-gas-shifted, the CO_2 can be separated out, and the resulting hydrogen-rich fuel used in a gas turbine (GT). For the CO_2 separation many options exist. One alternative is to use a chemical absorption system utilizing a hot potassium carbonate solution. The potassium carbonate solvent is an aqueous alkaline solvent particularly suited for processes with high total pressure and high CO_2 concentration. The process performs on the principle of a pressure swing absorption-desorption cycle

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with absorption taking place at high pressures. CO_2 capture by the use of potassium carbonate is, for example, described by Kohl and Nielsen [2].

The topic of this study is analysis of an IRCC process, with a special focus on off-design simulations. Similar pre-combustion process configurations have been studied by Andersen et al. [3] and Ertesvåg et al. [4]. Consonni and Viganò [5] also analyzes a pre-combustion setup but with co-generation of power and hydrogen. Hoffmann et al. [6] investigates a pre-combustion cycle using partial oxidation reforming. The cited studies focus on design case analysis. There is limited amount of literature in terms of off-design analysis of CO_2 capture cycles. Part load analyses of natural gas post-combustion systems are performed by Möller et al. [7]. Haag et al. [8] and Naqvi et al. [9] analyze the part load behavior of some of the proposed oxy-fuel cycles.

The remainder of the paper is divided into the following sections: Section 2 describes the process. Section 3 describes the details of the methodologies used in the paper. The results are shown and analyzed in Section 4, and concluding remarks are given in Section 5.

2. Process description

The process reforms natural gas to a syngas as shown in Figure 1. The syngas is water-gas shifted converting CO to CO_2 and the CO_2 separated out before the hydrogen-rich fuel is used for the gas turbine. As the auto-thermal reformer (ATR) is air-blown there will be a significant portion of nitrogen in the gas. This nitrogen is used as fuel diluent for NO_x abatement in the GT combustor. The air needed for the ATR is bled from the GT compressor discharge plenum and boosted up to system pressure with a booster compressor. There are a number of heat exchangers in the system. The pre-heating of the reforming streams is handled in various zones in the heat recovery steam generator (HRSG). The syngas cooler, located after the ATR, acts as an evaporator for the high-pressure (HP) steam cycle. The other heat exchangers for the process streams either generate low-pressure (LP) steam for the reboiler in the capture sub-system or pre-heat the fuel for the GT. The selected gas turbine is a GE 9FB set up for an integrated gasification combined cycle (IGCC). The requirements for an IRCC GT and an IGCC GT are very similar. The bottoming steam cycle, including the HRSG and a steam turbine (ST), is a one-pressure system at approximately 85 bar. The CO_2 capture sub-system consists of a hot potassium carbonate process. The CO_2 is compressed to 150 bar in the CO_2 compression and pump train.

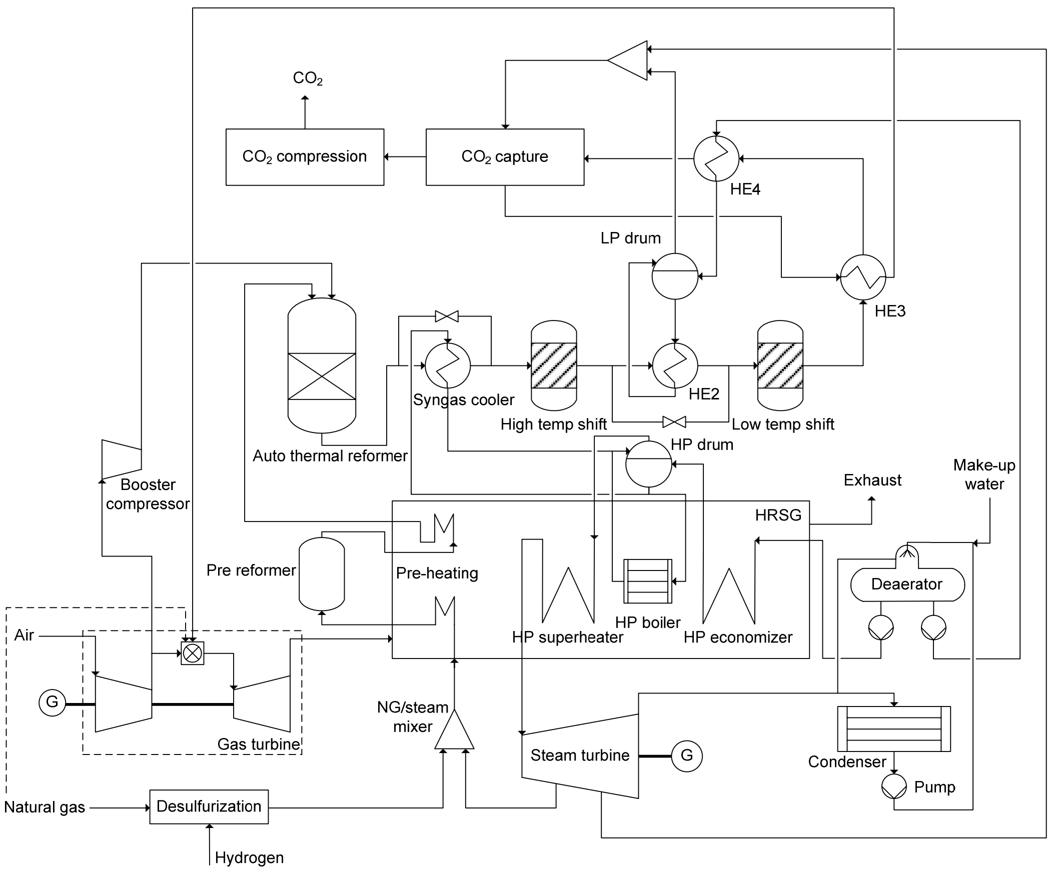

Figure 1. Process flow sheet of integrated reforming combined cycle

3. Methodology

The design case model was modeled in GT PRO by Thermoflow, and in Aspen Plus. The Aspen Plus simulations consisted of two separate models, one that included the reforming process and the water-gas shift reactors. In this model, numerous heat exchangers were included, among those the whole pre-heating section. Air and CO_2 compression was also incorporated into the model. The chemical absorption process was modeled as a hot potassium carbonate model in a separate flow sheet. The models were linked by Microsoft Excel utilizing Aspen Simulation Workbook and the Thermoflow E-LINK. For the CO_2 capture system the model was not directly linked to Excel, instead a simple separator model was included in the reforming flow sheet with inputs such as split ratios, temperatures, and pressures from the absorption model. Outputs from the capture model also included pump work and reboiler duty.

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. For these steady-state off-design simulations, GT MASTER by Thermoflow in conjunction with Aspen Plus were used. Also, inputs such as heat exchanger areas, compressor design point, etc., were linked in from the Aspen Plus reforming *design* model. The overall simulation overview with the linking is displayed in Figure 2.

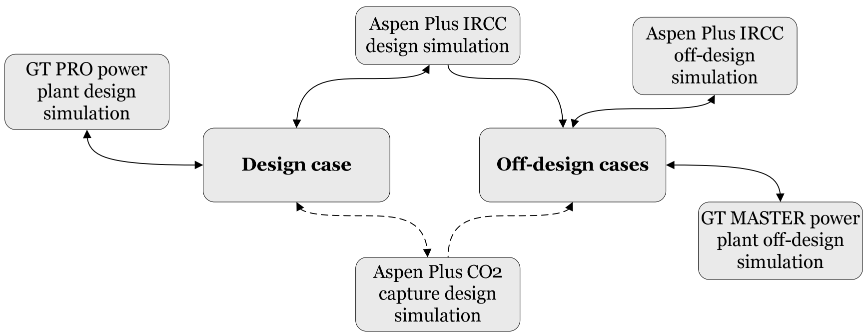

Figure 2. Overall simulation overview with software linking

3.1. Design case assumptions

The process was designed with a requirement of at least 85% CO_2 capture rate. The capture rate is here defined as the fraction of formed CO_2 that is captured. To achieve an overall capture rate of 85% the chemical absorption system was modeled for a 90% capture rate. It can be argued that the 90% capture rate is too low and that a higher design capture rate for the absorption sub-system would have been preferable and instead operate the system at a lower steam-to-carbon ratio (S/C) to achieve the overall capture rate of 85%. A lower S/C would lead to a higher methane slip from the ATR and a lower CO conversion in the water-gas-shift reactors but the lower amount of steam used would increase the net plant efficiency. The selected S/C for the simulations is 1.5. During the simulation work it was noted that the low-pressure and intermediate-pressure sections in the HRSG became quite small because of the significant pre-heating requirements (which could have been compensated with duct firing in the HRSG). Because of this and to simplify the process it was decided to have a one-pressure level in the HRSG. The pressure level was set at approximately 85 bar for the design case. Other assumptions include a condenser pressure of 0.04 bar, a 3% pressure drop (of inlet pressure) in the heat exchangers and reactors, and ISO ambient conditions.

3.2. Off-design analysis

One of the main focal points for the process analysis was related to off-design conditions. Two part load points were analyzed; 80% and 60% of the design case gas turbine load respectively. The off-design analyses were steady-state based. It should also be mentioned that off-design considerations can affect the design of the process. For example, in this process, steam extracted for the reboiler in the absorption system had to be extracted at a higher pressure than necessary to achieve a sufficient pressure level also at part load. In addition, off-design considerations for the booster compressor may lead to a selection of a less than optimum design point to achieve the required pressure ratio at reduced mass flow rates. In the following sub-sections the theory and methodology used for the part load scenarios will be presented.

3.2.1. Heat exchanger analysis

In the off-design scenarios the overall heat transfer coefficient U in the heat exchangers will vary. With inclusion of surface fouling and fin effects (extended surface) U can be expressed as:

$$\frac{1}{UA} = \frac{1}{(\eta_o hA)_c} + \frac{R''_{f,c}}{(\eta_o A)_c} + R_w + \frac{R''_{f,h}}{(\eta_o A)_h} + \frac{1}{(\eta_o hA)_h} \tag{1}$$

In the off-design analysis presented in this paper an as-new plant is assumed with no aging or fouling. The fouling factors $R''_{f,c}$ and $R''_{f,h}$ are therefore set to 0. The wall conduction term R_w is also neglected. η_o is the overall surface efficiency of a finned surface and A is the heat transfer area. Subscripts c and h refer to the cold and hot side of the heat exchanger, respectively. In the pre-heating heat exchangers in the HRSG the cold side has a high steam content. In the syngas cooler, as well as in HE2 and HE4 (refer to Figure 1) the cold side has water and steam only. Compared to the hot side, which contains gas with a lower level of steam, the cold side heat transfer coefficient is assumed much larger, that is $h_c \gg h_h$. Equation (1) can then be simplified to:

$$\frac{1}{UA} = \frac{1}{(\eta_o hA)_h} \tag{2}$$

The area A and fin efficiency η_o are constant when comparing design to off-design conditions. Using the Nusselt number and an empirical correlation including the Reynolds and Prandtl numbers (Incropera and DeWitt [10]):

$$Nu_D \equiv \frac{hD}{k} = C\, Re_D^m\, Pr^n \tag{3}$$

The constants C, m, and n, are assumed independent of the nature of the fluid. The Prandtl number Pr and thermal conductivity k are assumed constant from design to off-design conditions. The diameter D is constant. For the simulations, it is of interest relating the off-design UA to the design $(UA)_d$. Equation (2) can then be written as:

$$\frac{UA}{(UA)_d} = \frac{Re_D^m}{Re_{D,d}^m} \tag{4}$$

By using $Re_D = \frac{\dot{m}D}{A\mu}$, where the dynamic viscosity μ is assumed constant, a simple expression for correction of the UA-value when going from design to off-design simulations can then be derived:

$$\frac{UA}{(UA)_d} = \left(\frac{\dot{m}_h}{\dot{m}_{h,d}}\right)^m \tag{5}$$

The m-constant is dependent on the geometry of the shell and tube heat exchanger. $\dot{m}$ is the fluid mass flow. Subscript d refers to design conditions.

3.2.2. Pressure drop analysis

Assuming fully developed turbulent flow, meaning the pressure gradient dp/dx is a constant, the pressure drop from axial position x_1 to x_2 can be expressed as:

$$\Delta p = -\int_{p_1}^{p_2} dp = f\,\frac{\rho u_m^2}{2D}\int_{x_1}^{x_2} dx = f\,\frac{\rho u_m^2}{2D}(x_2 - x_1) \tag{6}$$

where u_m is the mean fluid velocity and ρ the density of the fluid. The Darcy friction factor f is defined as:

$$f \equiv \frac{-(dp/dx)D}{\rho u_m^2/2} \tag{7}$$

By using $\dot{m} = \rho u_m A$, and comparing to design conditions the following expression can be derived for off-design considerations:

$$\frac{\Delta p}{\Delta p_d} = \left(\frac{\dot{m}}{\dot{m}_d}\right)^2 \frac{\rho_d}{\rho} \tag{8}$$

3.2.3. Compressor map

For the air booster compressor in Figure 1, a compressor map has been used for calculating the outlet pressure and isentropic efficiency in off-design operating points. The map has been adopted from the original, presented in a map collection by Kurzke [11], to fit the process in the analysis. The following non-dimensional ratios have been used:

The pressure ratio $\Pi = \frac{p_{02}}{p_{01}}$, the corrected mass flow $\dot{m}_{corr} = \frac{\dot{m}\sqrt{T_{01}}}{p_{01}}$, and the corrected rotational speed $N_{corr} = \frac{N}{\sqrt{T_{01}}}$. p_{01} is the stagnation pressure at compressor inlet and p_{02} at compressor discharge. T_{01} is the stagnation temperature at compressor inlet.

In Figure 3, $\dot{m}_{corr}$ is plotted versus Π and the isentropic efficiency η_{is} for different N_{corr}. The corrected mass flow and the corrected speed are relative to design value. The surge line is also visible in the figure. The chosen design point for the booster air compressor is indicated in the graph. For off-design operating conditions it is assumed that the compressor can be speed controlled. For the GT compressor, GT MASTER used maps built-in to the program.

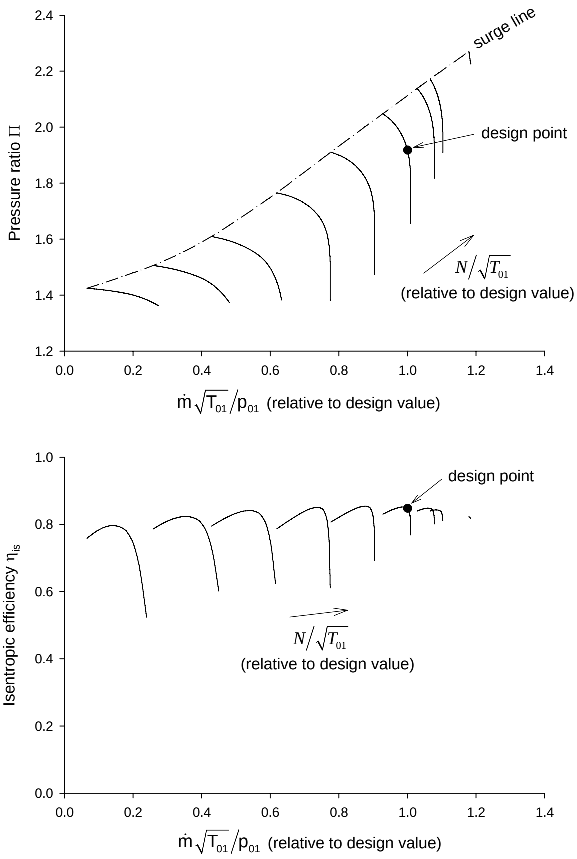

Figure 3. Compressor map for air booster compressor

When choosing the design and off-design operating points it is important to consider the surge margin to ensure stable compressor operation and avoid surge conditions. The surge margin is here defined as:

$$SM = \frac{\dot{m}_{corr,rel} - \dot{m}_{corr,rel,surge}}{\dot{m}_{corr,rel}} \tag{9}$$

Here $\dot{m}_{corr,rel}$ is the corrected mass flow relative to the design corrected mass flow. $\dot{m}_{corr,rel,surge}$ is the corrected mass flow at surge conditions on the operating line (constant speed line) relative to the design corrected mass flow.

4. Results and discussion

The main results for the design and off-design cases are presented in Table 1. The net plant efficiency is 43.2% for the design case with about two percentage-points drop to each subsequent part load case. The net plant efficiency is defined as:

$$\eta_{net,plant} = \frac{(W_t - W_c)\eta_m \eta_{gen} + W_s \eta_m \eta_{gen} - (W_{comp} + W_p)/(\eta_m \eta_{drive}) - W_{aux}}{\dot{m}_{NG} LHV_{NG}} \qquad (10)$$

Here W_t is the GT turbine power, W_c the GT compressor power, W_s the ST power, W_{comp} the total power consumption by the air and CO_2 compression. W_p is the pump power for feed water pumps, pumps in the absorption sub-system, etc. W_{aux} is the auxiliary power requirement. η_m is the mechanical efficiency and η_{gen} is the generator efficiency. η_{drive} is the efficiency of the drives for the different compressors and pumps. $\dot{m}_{NG}$ is the natural gas mass flow entering the system and LHV_{NG} the lower heating value of the natural gas. Note that all the power terms are defined as their absolute values meaning all power terms are considered positive and the sign is handled in the equation itself.

The capture rate is just above 86% for all cases, except for the reference case which has no CO_2 capture. The reference case is based on a natural gas combined cycle plant with a GE 9FB gas turbine and a triple-pressure steam cycle.

Table 1. Result summary for design case (100%), off-design cases (80% and 60%) and reference case (100% ref.)

Gas turbine relative load [%]	100	100 (ref.)	80	60
Natural gas LHV input [MW]	805.4	754.1	690.9	573.9
Gross power output GT [MW]	245.6	285.1	196.5	147.4
Gross power output ST [MW]	139.3	144.6	119.4	102.1
Gross power output [MW]	384.8	429.6	316.0	249.4
Gross power output [% of LHV input]	47.8	57.0	45.7	43.5
Air compression [MW]	14.0	-	10.7	7.9
Air compression [% of LHV input]	1.7	-	1.6	1.4
CO_2 compression [MW]	15.5	-	12.9	10.8
CO_2 compression [% of LHV input]	1.9	-	1.9	1.9
CO_2 capture pumps [MW]	1.5	-	1.1	0.9
CO_2 capture pumps [% of LHV input]	0.2	-	0.2	0.2
BFW pumps in pre-comb process [MW]	0.0	-	0.0	0.0
BFW pumps in pre-comb process [% of LHV input]	0.0	-	0.0	0.0
Auxiliaries [MW]	5.8	5.2	5.5	5.3
Auxiliaries [% of LHV input]	0.7	0.7	0.8	0.9
Net power output [MW]	**348.1**	**424.4**	**285.6**	**224.5**
Net plant efficiency [% of LHV input]	**43.2**	**56.3**	**41.3**	**39.1**
Efficiency capture penalty [%-point loss to ref. case]	13.1	-	-	-
CO_2 emissions [g CO_2 / net kWh el.]	68.3	377.6	71.8	73.7
CO_2 capture rate [%]	**86.1**	**0**	**86.0**	**86.4**

The turbine inlet temperature for the gas turbine set is a critical parameter for the overall plant performance. For the base case of this analysis, a conservative temperature of 1297°C has been selected. The reason for this assumption is two-fold. For one, the IGCC setup of GE's 9FB GT includes replacing the hot gas path of the FB with FA parts. The 9FA design turbine inlet temperature is 1327°C. Secondly, because of the hydrogen fuel which leads to an increase in steam content in the turbine compared to when firing natural gas, the heat transfer rate to the turbine blades increase, leading to a higher blade metal temperature. Because of this, another 30°C decrease in firing rate has been implemented in the model leading to the 1297°C turbine inlet temperature. If the GT could be fired at the full 9FB firing rate of 1427°C the net plant efficiency would increase from 43.2% to 44.7%. Chiesa et al. [12], and Todd and Battista [13] addresses issues related to firing hydrogen in gas turbines.

The CO_2 capture reboiler duty is another parameter affecting the plant efficiency. With the setup as shown in Figure 1, a part of the steam for the reboiler in the capture sub-system is extracted from the ST. If the reboiler duty would decrease from the current 1980 kJ/kg CO_2 captured down to approximately 1250 kJ/kg CO_2 captured, no steam extraction from the ST to the capture sub-system would be necessary and the ST output would increase. This would increase the net plant efficiency from the base case efficiency of 43.2% to 43.9%.

Figure 4 shows a T-Q diagram of the HRSG. Notable is that the boiler (HPB1 in Figure 4) is small compared to a more standard HRSG design. The reason for this is the large amount of steam generated in the syngas cooler. This means the economizer and the superheater in the HRSG are rather large but with a smaller boiler. The vertical jumps of the gas temperature in the figure are due to the pre-heating of the process streams. It should be noted that the majority of the pre-heating is upstream of the superheater meaning a significant portion of the available heat of the gas stream is removed before any steam is generated. The unconventional design is because of the integration with the reforming process. One can argue that a triple-pressure steam cycle would have a higher efficiency compared to the single-pressure system applied here. However, because of all the pre-heat streams and the syngas cooler acting as an evaporator, the low-pressure and intermediate-pressure sub-systems would have been very small adding very limited value at an increased complexity. Duct firing could have changed this picture, however for this work it was decided not to utilize supplementary firing.

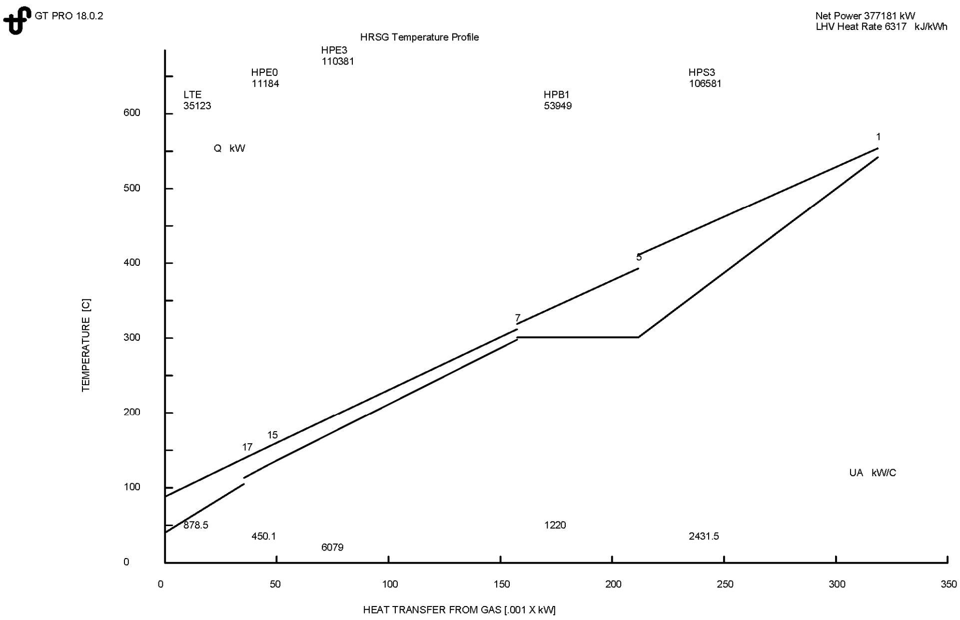

Figure 4. GT PRO T-Q diagram for single-pressure heat recovery steam generator

5. Conclusions

By combining simulation tools for chemical engineering and power plant engineering analyses respectively, a helpful representation of the overall system can be accomplished for an IRCC process. The IRCC process may involve a significant integration between the power cycle and the reforming process as is the case for the cycle studied. It can therefore be advantageous to combine the tools as is shown in this paper.

The results indicate a rather low net plant efficiency compared to the reference case. This is for one, strongly influenced by the conservative selection of turbine inlet temperature. An increase of the temperature would significantly reduce the capture penalty. Secondly, the reboiler duty could be lowered, again leading to a higher plant efficiency. Other items to consider is supplementary firing in the HRSG to allow for a more standard design of the steam generator and the use of low-temperature process heat instead of low-pressure steam for the reboiler in the chemical absorption system. Also, a higher design CO_2 capture rate for the absorption system could be advantageous. However, it should be mentioned that the IRCC process is complex and many options and configuration possibilities are present. In the end, a line had to be drawn how far to extend the analysis work.

The off-design simulation results show the possibility to run a plant like this at part load conditions down to approximately 60% gas turbine load. Reducing the load further down may not be practical for several reasons. For example, the CO emissions from the GT would increase and potentially also NO_x. Further, the efficiency drop would at some point be too large to justify. The air booster compressor pressure ratio would continue to decrease (if using a one-compressor train and not multiple compressors) meaning the overall system pressure would keep decreasing until the level is too low for realistic plant operation. The possibility would then be to switch to natural gas fuel for the gas turbine. Indeed, the plant is designed for having natural gas as back-up fuel for the GT. In fact, to start up a plant like the one in the study, natural gas is required. At a load around 30% or above the switch-over to the hydrogen-rich fuel would take place during a start-up.

6. Acknowledgments

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