Available online at www.sciencedirect.com

ScienceDirect

Energy Procedia 63 (2014) 453 – 462

Energy Procedia

Optimisation potentials for the heat recovery in a semi-closed oxyfuel-combustion combined cycle with a reheat gas turbine

Jan Mletzko[a]*, Alfons Kather[a]

[a]Institute of Energy Systems, Hamburg University of Technology, Denickestrasse 15, D-21073 Hamburg, Germany

Abstract

Semi-closed oxyfuel-combustion combined cycle plants are one possibility to realise CCS technologies with natural gas-fired power plants. Due to the recirculation of exhaust gas the working fluid in the gas turbine consists mainly of CO_2. This leads to the necessity for an increased pressure ratio to achieve the efficiency and the exhaust gas temperatures of conventional gas turbines. In this work a conventional natural gas-fired combined cycle plant with a reheat gas turbine modified for oxyfuel operation will be presented. As it is assumed that the necessary pressure ratio is not realisable with axial compressors two different scenarios are considered. On the one hand the pressure ratio is the same as in the conventional process and on the other hand the pressure ratio is set to an assumed maximum of 60. The exhaust gas temperature lies significantly above the values of the conventional plant. Generally the higher pressure ratio results in a higher net efficiency of the plant. An unchanged heat recovery process results in a loss of efficiency of at least 9.3 to 10.8 %-pts. depending on the pressure ratio. Increasing the steam parameters can achieve a reduction of this loss ranging from 0.5 %-pts for the high pressure ratio to 1.0 %-pts for the low pressure ratio. An alternative approach for the heat recovery is the application of an internal recuperator in the gas turbine. When the achievable pressure ratio is far from the optimum value, as in the low pressure ratio case, this approach can lead to significantly lower losses of 9.1 %-pts. With a higher pressure ratio the effects of the optimisation options are similar.

Peer-review under responsibility of the Organizing Committee of GHGT-12

Keywords: power plant; gas turbine; CO_2-capture; oxyfuel; heat recovery; recuperator

* Corresponding author. Tel.: +49-40-42878-4551; fax: +49-40-42878-2841.
E-mail address: jan.mletzko@tu-hamburg.de

Peer-review under responsibility of the Organizing Committee of GHGT-12
doi:10.1016/j.egypro.2014.11.048

1. Introduction

Natural gas-fired combined cycle power plants (CCPP) are characterised by low CO_2 emissions due to the achievable efficiency of approx. 60 % and the low carbon content of the fuel. A further reduction of the CO_2 emissions can be achieved if carbon capture and storage (CCS) technologies are applied. One pathway of CCS technologies is the oxyfuel process. The main target of the oxyfuel process is to increase the partial pressure of CO_2 in the flue gas by substituting combustion air with a mixture of oxygen and recycled flue gas. This facilitates the further processing of the CO_2 towards geological storage specifications. In coal fired power stations the recycle of flue gas is necessary to keep the furnace exit temperatures within material limitations. In gas turbine applications it substitutes the amount of excess air that is used to control the turbine inlet temperature. Additional power demands within the oxyfuel process lead to an efficiency penalty for the overall process. The main energy consumers are the air separation unit (ASU) to supply the oxygen and the gas processing unit (GPU) including the compression to a pressure level suitable for transport and storage.

The semi-closed oxyfuel-combustion combined cycle (SCOC-CC) is one possibility to realise the oxyfuel process for natural gas fired combined cycle plants [1]. Besides the additional energy consumers, in this kind of process the gas turbine is strongly affected by the high CO_2-content in the working fluid. Due to the lower heat capacity ratio the compressor discharge temperature decreases and the turbine exhaust temperature increases at a given pressure ratio of the gas turbine [2]. The high exhaust gas temperature leads to a reduced efficiency of the gas turbine. For this reason the pressure ratio of oxyfuel gas turbines is increased from the range of 20 to around 40 to 60 to achieve similar efficiencies and temperatures as in the air-blown case [1, 3].

Where sequential combustion is applied a significantly higher pressure ratio is required in the air-blown case already, e.g., the ALSTOM® GT26 has a pressure ratio of above 30. If oxyfuel technology is applied to a reheat gas turbine the necessary increase in pressure ratio has to be much higher. The conventional design of a gas turbine with an axial compressor has limited high pressure ratios, because the compressor efficiency drops due to tip clearance losses. As mentioned before, a limited pressure ratio would result in a drop in the gas turbine efficiency and, if the heat recovery process remains unchanged, also in the overall efficiency of the oxyfuel process.

As an alternative solution to increasing the pressure ratio the heat recovery can be modified, so that the highly exergetic exhaust gas flow can be used in a more efficient way. The most obvious possibility to improve heat use is to increase the steam quality to reduce exergy losses in the heat recovery steam generator (HRSG). Then, however, alternative approaches for the use of high temperature heat from the exhaust gas should be considered. Internal recuperation could lower the exhaust gas temperature by preheating the oxidator instead. Still, the technical feasibility of heat exchange at high temperatures and pressures could be a barrier for the latter process.

2. Reference case

A conventional NGCC is modelled in EBSILON®*Professional* as the reference case schematically shown in Fig. 1. It basically consists of two GT26 gas turbines and the bottoming steam cycle. The most important link between the two parts are the HRSGs (one for each GT) to transfer the heat from the exhaust gas side to the water/steam cycle. The steam process is a triple pressure reheat process with a once-through boiler in the HP section. However, both processes are additionally linked with a fuel preheating system fed with preheated IP feed water and by a once-through cooling system (OTC) for the blade cooling air flows that generates HP steam. The temperature levels in the HRSG are determined by the boiling condition of the pressure stages. The mass flow through the different stages is set by the temperature difference at the pinch point ΔT_{PP}, i.e. the temperature difference between boiling temperature at the pressure level and the flue gas temperature. To avoid evaporation in the economiser sections the latter are operated at an elevated pressure of 10 and 8 bar above the pressure in the HP and IP boiling sections, respectively. The condensate in the feed water tank is heated to 60 °C in a pre-heating loop to avoid dew point corrosion. The gas turbine is considered as black box with given power, efficiency, exhaust gas flow, and temperature. Furthermore, the fuel gas pre-heating temperature and the additional steam generation due to the OTC are given. The most important design assumptions for the reference process are given in Table 1. Based on the LHV of 46.2 MJ/kg for the natural gas used as fuel, the reference process reaches an efficiency of 59.8 % gross and 59.2 % net.

Table 1. Design data for the reference process without CCS.

Gas turbine		Water steam cycle	
Ambient pressure in bar	1.013	Pressure levels at turbine inlet (HP/IP/LP) in bar	160/40/4.2
Ambient temperature in °C	10	Steam temperatures (SH/RH) in °C	585/585
Ambient relative humidity in %	83.2	ΔT_{PP} (HP/IP/LP) in K	8/8/14
Power in MW	278	Condenser pressure in mbar	45
El. efficiency in %	37.6	Steam turbine efficiency (HP/IP/LP)	92/93.5/88.5
Exhaust gas temperature in °C	619	Generator efficiency in %	99
Exhaust gas flow in kg/s	644	Pump efficiency in %	80
Pressure at turbine outlet in bar	1.041	Engine efficiency in %	99
Fuel temperature in °C	220	Pressure loss HRSG (steam/water) as percentage of inlet pressure	1/2.5
Cooling Duty OTC in MW	26.9	Pressure loss between HRSG and Turbine (HP/IP/LP) as percentage of inlet pressure	2.5/2.5/5

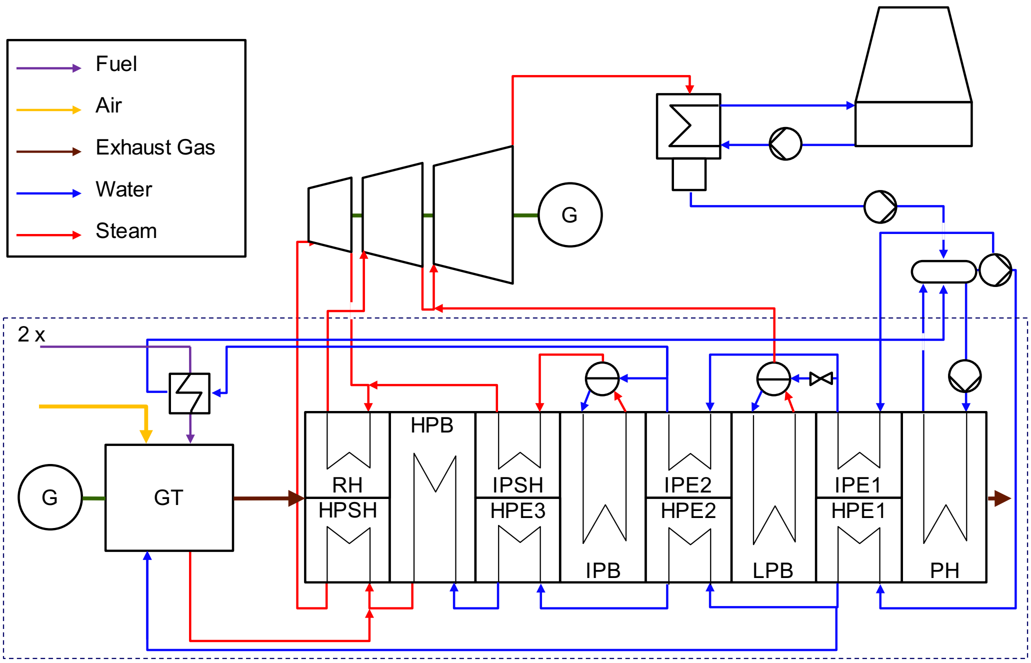

Fig. 1. Schematic layout of the reference process.

3. Semi-closed oxyfuel-combustion combined cycle

To model the combined cycle with oxyfuel technology as a first step the effects on the gas turbine have to be evaluated. Therefore the gas turbine is modelled in a detailed way as shown in Fig 2. The parameters of the model, i.e. ISO turbine inlet temperatures (ISO-TIT), polytropic efficiencies of the compressor and the turbine, mechanical efficiency, pressure losses etc., are then fitted to match the design data given in Table 1. The compressor pressure ratio of 32.8 and more detailed information like compressor outlet temperatures and cooling flows are taken from Boksteen et al [4]. As described by Güthe [5] the fuel mass flow to the EV and SEV combustors are divided approx. 1:1. A cooling model as described in Jonsson [6] and Dae [7] is used to determine the cooling mass flows in the oxyfuel case. Therefore, each of the 3 cooled stages of the LPT is considered as one cooled turbine with cooled nozzle and rotor blades. The cooling mass flows are calculated with Eq. (1).

$$\frac{\dot{m}_c c_{p,c}}{\dot{m}_g c_{p,g}} = b \frac{T_g - T_b}{T_b - T_c} \tag{1}$$

The parameter b is determined for each nozzle and each blade from the reference gas turbine using the cooling mass flows from [4] and a maximum blade temperature of 850 °C. The OTCs are set to cool the first two cooling flows (see Fig. 2) to 345 °C with an efficiency of 97 %, to reach the cooling duty given in Table 1. Depending on the cooling mass flow an additional pressure drop in the turbine is accounted for, as described in Jonsson [6] and Dae [7].

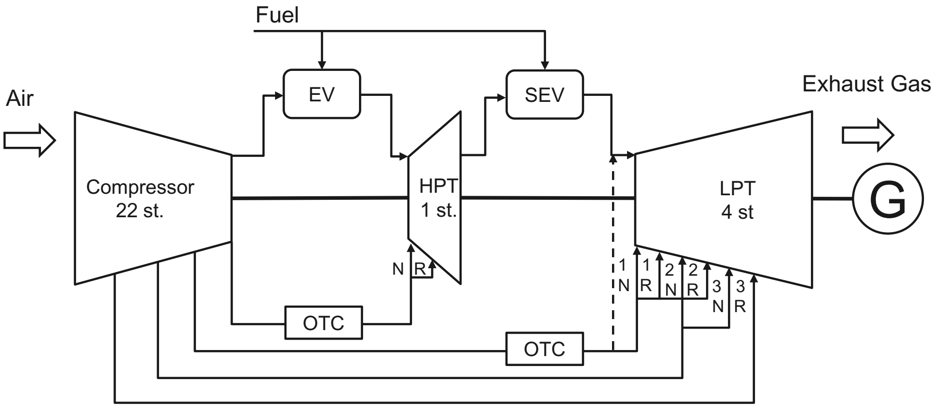

Fig. 2. Schematic layout of the detailed GT26 model [7].

To transform the conventional gas turbine into an oxyfuel gas turbine the fitted parameters are fixed and the NGCC process is modified on the gas side as shown in Fig. 3, the flow sheet of the water steam process remains as in the reference case. The oxygen is supplied upstream of the EV combustor so that the cooling flows are not diluted with oxygen. To keep the combustion temperature within the material limits the exhaust gas is recycled to the compressor substituting the air from the conventional process. The mass flow of the flue gas to be recycled is determined by keeping the ISO-TIT as in the reference case. To reduce the necessary compressor work the exhaust gas is cooled to 40 °C and water is condensed. The excess exhaust gas is compressed to an assumed pipeline pressure of 110 bar for transport and storage. The compressor is modelled as a three stage compressor with intercooling and condensation. No further refining of the CO_2 is considered in this work. The oxygen is provided by a cryogenic ASU. The specific energy demand of the ASU is assumed to be 202 kWh/t_{O_2}, producing oxygen with a purity of 95 vol.-%. As the ASU produces oxygen at atmospheric pressure an intercooled three stage compression is used to achieve the EV-combustor pressure. The global excess oxygen ratio is set to 1.018, considering only the oxygen coming from the ASU. This results in an excess oxygen ratio of 1.3 in the SEV combustor.

As mentioned before, the pressure ratio of the SCOC-CC process has to be increased to achieve similar exhaust gas temperatures and efficiencies like in a conventional gas turbine. In this case the reference pressure ratio is already 32.8, which is high for axial compressors. The necessary pressure ratio for this SCOC-CC turbine would be above 100, which is not a reasonable value to achieve with conventional axial compressors. Therefore, the upper limit of the pressure ratio in this work is assumed to be 60.

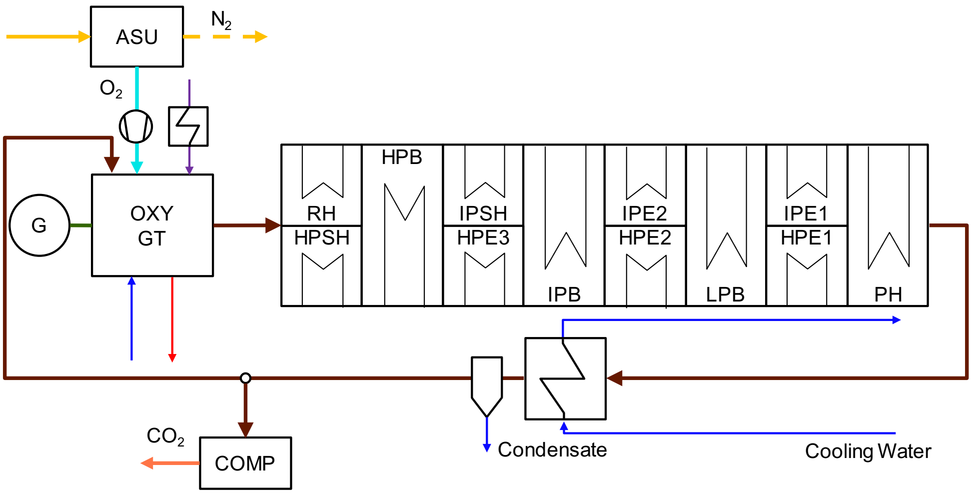

Fig. 3. Schematic layout of the SCOC-CC process.

Based on this assumption two cases are identified for the further study of the optimisation potentials for the heat recovery. On the one hand a SCOC-CC process with the same pressure ratio as in the reference case is examined, whereas on the other hand a modified process with a pressure ratio of 60 is assumed. The results of different cases for the SCOC-CC gas turbine in comparison to the air-blown case are shown in Table 2. It should be noted that the exact mass flow through the turbine is determined by the design of turbo machinery. As this is not included in this work, power and heat duties are given as specific values in relation to the exhaust gas mass flow (index EG), i.e. no scaling effects are taken into account. For the calculation of the electric efficiency only the power of the gas turbine is accounted. This efficiency can be higher than that of the reference gas turbine (as in the PR 60 case) because the compression work for the oxygen is shifted to the energy demand for oxygen supply. Another point to mention is that in the PR 32.8 case the cooling in the OTC is disabled because the temperature level of the cooling flows is not sufficient to generate HP steam.

Table 2. Results of the SCOC-CC gas turbine in comparison with the reference case.

	PR 32.8	PR 60	Air (PR 32.8)
Specific work in kJ/kg_{EG}	438.8	475.1	431.7
El. efficiency of the gas turbine in %	34.4	38.5	37.6
Exhaust gas temperature in °C	791	697	619
Specific cooling duty OTC in kJ/kg_{EG}	--	38.3	41.8
Exhaust gas composition in kg/kg:			
x_{CO2}	0.811	0.813	0.066
x_{H2O}	0.084	0.082	0.057
x_{O2}	0.020	0.021	0.133
x_{N2}	0.012	0.013	0.732
x_{Ar}	0.072	0.072	0.013

It can be seen that the exhaust gas temperature rises from 619 °C to 697 °C for a pressure ratio of 60 and to 791 °C for 32.8, respectively. Kail [8] shows that with rising exhaust gas temperatures the improvement due to the application of multiple pressure level HRSGs in relation to the single pressure process decreases. This can be explained by the fact that more energy is available above the pinch point temperature of the HP level while the amount of usable energy downstream the HP boiler stays the same except for small changes in the specific heat capacity. Consequently, more HP steam can be generated and so more energy is needed to pre-heat the HP feed water. The amount of energy to generate IP and LP steam is more and more reduced. In the case that no IP and LP steam is generated the process virtually becomes a one pressure cycle. According to Kail [8] this is the case above 750 °C for an air-blown gas turbine and a limited steam temperature of 600 °C. If the exhaust gas temperature is higher, the temperature difference at the pinch-point has to be increased to keep HRSG in energy balance.

In consideration of these effects the following procedure was chosen to achieve comparable results for investigating the potentials of the heat recovery:

- As the condensate pre-heat loop needs a sufficient temperature level to operate, the exhaust gas temperature downstream of the first economiser (HPE1, IPE1 in Fig. 1) must not fall below 100 °C. This value is taken in accordance to the results of the reference process.
- As a first step the IP and LP section of the HRSG are disabled and $\Delta T_{PP,HP}$ remains at 8 K, like in the reference process.
- If the exhaust gas temperature does not exceed 100 °C downstream of the first economiser, $\Delta T_{PP,HP}$ is increased until the condition is met.
- If the exhaust gas temperature exceeds 100 °C, the lower pressure levels are activated and the steps are repeated with $\Delta T_{PP,IP}$ and $\Delta T_{PP,LP}$ in the same way.

Because of the high turbine outlet temperature in the PR 32.8 case $\Delta T_{PP,HP}$ has to be increased to 37 K to achieve the temperature of 100 °C. In this case the process can be operated with a single pressure HRSG. For this reason the fuel pre-heating flow is switched to HP feed water. The gross efficiency of the overall process is 61.2 %. This shows that, despite the reduced efficiency of the gas turbine, the gross efficiency lies 1.4 %-pts. higher than the reference process. This is due to the fact that the specific electric energy of the steam turbine is 343 kJ/kg_{EG}, i.e. approx. 33 % more than in the reference case (258 kJ/kg_{EG}). Nevertheless, the additional energy consumers of the oxyfuel process have to be subtracted. The supply of oxygen consists of the oxygen compressor and the ASU which reduce the efficiency by 3.3 %-pts. and 5.9 %-pts., respectively. The compression of the flue gas to 110 bar for geological sequestration will also reduce the efficiency by 2.7 %-pts. Due to the higher cooling duty of the exhaust gas recycle and the intercooled O_2 and CO_2 compression the auxiliary duty of the SCOC-CC plant is higher than in the reference case and reduces the efficiency by 0.9 %-pts. The resulting net efficiency of 48.4 % means a total loss of 10.8 %-pts. in comparison to the reference case.

In the PR 60 the exhaust gas temperature is considerably reduced but still clearly above the value of the reference case. If the IP and LP sections of the HRSG are disabled and $\Delta T_{PP,HP}$ is set to 8 K, the flue gas temperature downstream of the first economiser is 115 °C and the gross efficiency is 63.0 %. The losses of the oxyfuel process remain the same except for the oxygen compressor that now needs to adapt to the higher pressure in the gas turbine. This results in a loss of 3.9 %-pts. Thus the overall net efficiency is 49.3 % which means a total loss of 9.7 %-pts.

To further reduce the exhaust gas temperature the IP pressure level is activated. If $\Delta T_{PP,IP}$ is set to 10 K the flue gas outlet temperature of 100 °C is matched, which means that for this case a two-pressure process can be realised. This results in a gross efficiency of 63.4 % which leads to a net efficiency of 49.9 % or a total loss of 9.3 %-pts.

4. Advanced steam parameters

To reduce the loss of efficiency of the SCOC-CC process without further increasing the pressure ratio the influence of increased steam parameters is investigated. In the reference process a live steam pressure of 160 bar and 585 °C at the turbine inlet is used. The same outlet temperature is used for the reheat. While the pressure level is determined by optimising the heat use, the temperature is chosen to allow a necessary temperature difference between the hot flue gas and the live steam. With the higher exhaust gas temperatures in the SCOC-CC the pressure levels and steam temperatures can be increased.

To evaluate the effects of the pressure at the turbine inlet, this is varied up to 200 bar which still remains subcritical within the HRSG. Three different cases with different live steam temperatures are considered. These are 585 °C (the reference value), 600 °C (the live steam temperature of modern coal fired power plants) and 620 °C for accounting for advancements in material technology. Additionally, another case is considered with supercritical live steam parameters of a modern coal fired power plant. In this case the live steam parameters are 285 bar and 600 °C and 59 bar and 620 °C in the reheat [9]. In the case of supercritical steam parameters for the calculation of $\Delta T_{PP,HP}$ the water side temperature cannot be regarded as the boiling temperature. In this case the HP boiler is removed from the process and the economiser is set to heat the water to the temperature at the critical point (374 °C). $\Delta T_{PP,HP}$ is then calculated at the cold end of the HP super heater.

The simulation results for the subcritical steam parameters are shown in Fig. 4. Generally it is shown that raising the steam pressure can achieve an improvement in the overall net efficiency of the SCOC-CC plant. For the same pressure ratio, the gas turbine efficiency is constant. The efficiency of the HRSG also does not change significantly, because the exhaust gas temperature at the outlet of the economiser is kept at constant temperature. So the increasing efficiency can be traced back to an increased efficiency of the steam process due to the higher pressure. For a constant steam temperature there is an increase of 0.3 to 0.4 %-pts. due to the rising steam pressure. Raising the steam temperature leads to a stronger increase of efficiency in the PR 32.8 case than in the PR 60 case. This can be explained as the rising steam temperature increases the necessary amount of energy that is transferred above the pinch point temperature. In the PR 32.8 case this means that $\Delta T_{PP,HP}$ can be chosen to be lower than in the reference case and this reduces consequently the exergy losses. In the PR 60 case $\Delta T_{PP,HP}$ is already at its minimal value. The consequence is that less HP steam is generated to keep the HRSG in energy balance, and this heat is transferred at lower pressure levels.

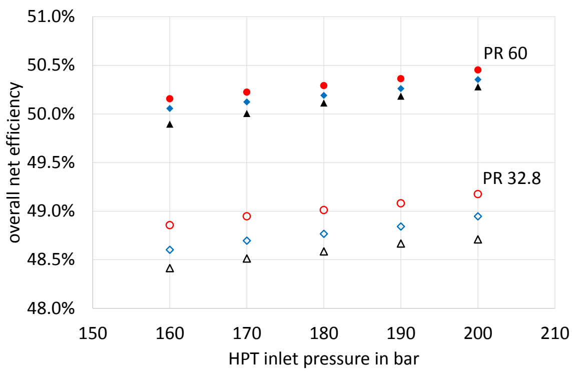

Fig. 4. Overall net efficiency for different live steam pressures and temperatures of 585 (black), 600 (blue) and 620 (red).

When supercritical live steam parameters (285 bar/600 °C/620 °C) are applied, the overall net efficiency is 49.4 % in the PR 32.8 case and 50.4 % in the PR 60 case. This corresponds to increases of 1.0 %-pts. and 0.5 %-pts., respectively, in relation to the unchanged heat recovery. The q-T-diagrams in Fig. 5 show that there is no reduction in the heat transferred to the HP level in the PR 32.8 case, whereas in the PR 60 case there is a clear reduction. It also shows that in the PR 60 case the steam process is much more restricted by the exhaust gas temperatures.

The realisation of HRSGs for the SCOC-CC process as described here means some technical challenges. On the one hand the advanced steam parameters need advanced materials compared to the reference case but, as conventional coal fired plants with these parameters already exist, this is not seen as a problem. On the other hand the internal insulation of the HRSG has to withstand the higher exhaust gas temperatures. A possible solution could be water cooled walls in the hot zone that can be used as heating surfaces for the steam process similar to coal fired boilers.

5. Recuperated oxyfuel gas turbine

An alternative approach for using the waste heat of a gas turbine is to utilise an internal recuperator. The hot exhaust gas is used in the gas turbine to pre-heat the air flow, thus saving fuel in the combustor. The principle of recuperated gas turbine is mainly investigated in aircraft applications where no further use of the waste heat can be realised [10].

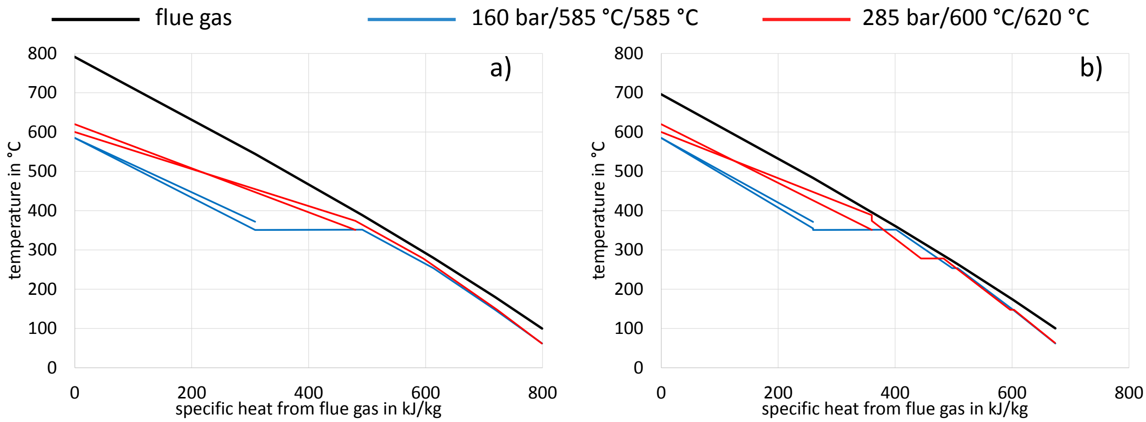

Fig. 5. Simplified q-T-diagram of (a) the PR 32.8 case and (b) the PR 60 case for sub- and supercritical live steam parameters.

In modern conventional combined cycle applications there are two main reasons for which the use of recuperation is not useful. The first reason is that the compressor outlet temperature of 550 °C is nearly as high as the turbine outlet temperature in case of the GT26 [4]. The second reason is that the exhaust gas can be efficiently used in a steam process. Furthermore, a recuperator means additional equipment, more complex construction and additional pressure losses in the system. For the oxyfuel gas turbine the conditions are different. The lower heat capacity ratio of the working fluid does not only result in a higher exhaust gas temperature at the same pressure ratio as seen before. The compressor outlet temperature is also decreased. This leads to a temperature difference between the exhaust gas and the oxidising fluid of approx. 410 °C in the PR 32.8 case and 240 °C in the PR 60 case, respectively.

To investigate the potential of a recuperated SCOC-CC process the gas turbine described before is modified as shown in Fig. 6. Downstream to the mixing of the compressed recycled exhaust gas and the oxygen a heat exchanger is placed, that heats the gas with the exhaust gas coming from the turbine. The heat exchanger is assumed to cool down the exhaust gas to 620 °C so as to have similar conditions at the HRSG inlet as in the reference case. As no detailed design of the recuperator is available, in a first step idealised conditions will be assumed, i.e. no pressure losses are accounted for. The effect of hot and cold side pressure drops is determined afterwards. The HRSG pressure levels and temperatures remain as defined in the reference process. To achieve comparable conditions the flue gas temperature downstream of the first economiser is again set to 100 °C.

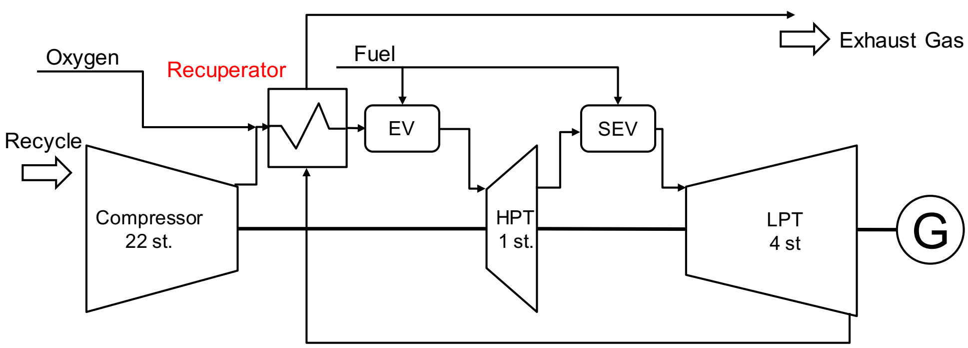

Fig. 6. Schematic layout of the recuperated gas turbine, cooling flows are not shown.

In Fig. 7 the resulting overall net efficiency for the recuperated oxyfuel gas turbine is shown for different pressure ratios. It can be seen that the optimum net efficiency for this process lies at the pressure ratio of 60 which was assumed as the maximum to realise in this work. The overall net efficiency is 50.1 % for PR 32.8 and 50.5 % for PR 60 respectively. For the PR 32.8 case the efficiency is significantly higher than what is achieved by advanced steam parameters. In contrast to this the improvement for the PR 60 case is roughly the same as what is achieved by advanced steam parameters.

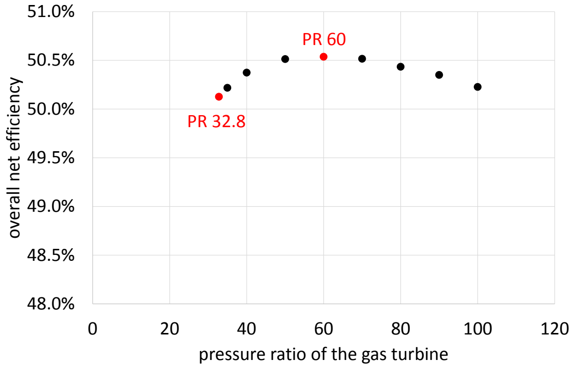

Fig. 7. Overall net efficiency for the idealised recuperated oxyfuel gas turbine depending on different pressure ratios.

For a pressure ratio of 32.8 pressure losses on the cold gas side (high pressure) will reduce the overall efficiency by approx. 0.2 %-pts. per bar of pressure drop. For a pressure ratio of 60 this value is about half of that, as the relative effect of the pressure drop is lower. On the hot gas side the effects are much higher as the pressure is low and a small increase in turbine outlet pressure reduces the pressure ratio significantly. The reduction in the overall efficiency is approx. 0.06 %-pts. per mbar on the hot gas side for both pressure ratios considered.

This shows the necessity for designing the process in a way that the pressure losses will not annihilate the gains in efficiency. Another challenge will be the availability of heat resistant recuperator materials. It was shown in the previous sections that the exhaust gas temperatures can rise up to 800 °C. McDonald [10] assumes that the application range of nickel based alloys was up to 790° C of hot gas temperature. Alternatives would be ceramic or carbon composite materials. In addition, the temperature of the gas and the pressure difference between the hot and cold side have to be considered.

6. Conclusion and Outlook

This work shows that optimising the heat recovery offers the potential for reducing the efficiency losses associated with SCOC-CC cycles, especially when these cannot be operated at their optimum pressure ratio. If the pressure ratio of the reference gas turbine remains unchanged, the net efficiency loss is 10.8 %-pts. in relation to the conventional NGCC. Increasing the steam parameters can reduce this loss to 9.8%-pts. A recuperated gas turbine cycle could achieve a further reduction potential down to a loss of 9.1 %-pts. However, the pressure losses of the recuperated system are crucial for the process. If the pressure ratio is increased to 60, the efficiency with unchanged heat recovery is 9.3 %-pts. In this case the loss in efficiency can be reduced to 8.7 %-pts. In contrary to a lower pressure ratio there is no big difference between the discussed optimisation options.

This work is outlined to show the potentials of an optimised heat recovery. As heat recovery with multiple pressures offers many variables, further work should focus especially on the optimisation of the IP and LP sections of the HRSG.

As this work focusses primarily on the thermodynamic behaviour, another issue is to define in more detail the technical options for realising the processes. Technical and economic issues need to be considered to evaluate the feasibility of the various possible concepts.

Acknowledgements

The GUD-POXY project is executed by a collaboration of the Institute of Energy Systems (Hamburg University of Technology, TUHH) and the Laboratory for Turbomachinery (Helmut Schmidt University Hamburg, HSU). The research project is funded by the Federal Ministry for Economic Affairs and Energy (project number 03ET7013A) and supported by the industry partners ALSTOM Carbon Capture GmbH, EnBW AG, E.ON Technologies GmbH and RWE Power AG. We would like to thank all members of the GUD-POXY project for their valuable discussions and contributions to this work.

Supported by:

Federal Ministry for Economic Affairs and Energy

on the basis of a decision by the German Bundestag

References

[1] Kvamsdal HM, Jordal K, Bolland O. A quantitative comparison of gas turbine cycles with CO_2 capture. Energy 2007;32:10-24.
[2] Yang HJ, Kang DW, Ahn JH, Kim TS. Evaluation of Design Performance of the Semi-Closed Oxy-Fuel Combustion Combined Cycle. J Eng Gas Turbines Power 2012;136.
[3] Riethmann T, Sander F, Span R. Modelling of a supercharged semi-closed oxyfuel combined cycle with CO_2 capture and analysis of the part-load behavior. Energy Procedia 2009;1:415-422.
[4] Boksteen SZ, van der Vecht DJ, Pecnik R, van Buijtenen, JP. Performance Modeling of a Modern Gas Turbine for Dispatch Optimization. Proceedings of ASME Turbo Expo 2012; Copenhagen, Denmark:Paper-no. GT2012-68137.
[5] Guethe F, Hellat J, Flohr P. The Reheat Concept: The Proven Pathway to Ultralow Emissions and High Efficiency and Flexibility. J Eng Gas Turbines Power 2009;131.
[6] Jonsson M, Bolland O, Buecker D, Rost M. Gas Turbine Cooling Model for Evaluation of Novel Cycles. Proceedings of ECOS 2005; Trondheim, Norway:641-650.
[7] Dae HJ, Soo HY, Jong JL, Tong SK. Evaluation of component characteristics of a reheat cycle gas turbine using measured performance data. J Mech Sci Technol 2008;22:350-360.
[8] Kail C. Analyse von Kraftwerksprozessen mit Gasturbinen unter energetischen, exergetischen und ökonomischen Aspekten (in German). PhD-Thesis 1998; TU Munich.
[9] Ebert K, Kuebel M, Lorey H, Michele W. 912 MW Supercritical Boiler for the next Coal Fired Plant Generation. Power-Gen Europe 2013; Vienna, Austria.
[10] McDonald CF, Massardo AF, Rodgers C, Stone A. Recuperated gas turbine aeroengines. Part III: engine concepts for reduced emissions, lower fuel consumption, and noise abatement. Aircraft Engineering and Aerospace Technology 2008;80/4:408-426.