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Modelling of a supercharged semi-closed oxyfuel combined cycle with CO_2 capture and analysis of the part-load behavior

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Abstract

Based on different sources a gas turbine model including a cooling model has been set up using the simulation software EBSILON®*Professional*. A gas turbine (GT) in simple cycle and a natural gas combined cycle (NGCC) including a triple pressure HRSG have been investigated and used as reference cycles. The NGCC has been modified to a semi-closed oxyfuel combined cycle (SCOC) with CO_2 capture. Results from the ENCAP project [1] have been used to evaluate the simulation results of this work. As a novel configuration a supercharged SCOC process (S-SCOC) has been presented. Both design-point and part-load simulations have been performed. In design-point the NGCC achieves a net efficiency of 56.6 %. For the SCOC process the net efficiency is lowered by 8.3 %-points due to the capture of CO_2. At part-load a sliding pressure (variable supercharge) has been introduced for the S-SCOC process. The sliding pressure for part-load leads to a constant volumetric flow rate through the GT over the whole load range. Therefore, the supercharged process yields a constant pressure ratio and a constant exhaust temperature at part load. Sliding pressure operation allows for a superior performance of the GT for the whole load range. The influence of a higher pressure of the exhaust gas stream on the heat transfer in the HRSG has been investigated. Fuel mass flow and flow conditions have been kept constant for the different processes. It has been found that the flow conditions for both processes (SCOC: constant density, volumetric flow rate changes; S-SCOC: vice versa) lead approximately to the same results for the Reynolds number on the flue gas side at part load. Although the conditions differ for the atmospheric SCOC process and the S-SCOC process with sliding pressure, the impact of part-load operation on the heat transfer is very similar. Since the efficiency of the supercharged GT hardly changes at part load, the net efficiency of the S-SCOC at part load remains very high. At 50% load the S-SCOC achieves an efficiency of 96 % of the design point efficiency compared to 91 % for the NGCC.

Keywords: CO_2 capture, oxyfuel, semi-closed gas turbine, supercharged, part-load

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1. Introduction

Several CO_2 capture technologies have been proposed over the last two decades; corresponding reviews were published, see for example [2] and [3]. Today, the process development related work focuses on novel concepts, which reduce the drop of the net efficiency caused by the CO_2 capture process. Thereby not only design-point but also part-load operation has to be considered. The semi-closed oxyfuel combined cycle investigated in the present work belongs to the group of processes with integrated CO_2 capture. As a new aspect, the well known semi-closed process has been supercharged. The inlet pressure of the compressor has been varied from 1 to 3 bar and even up to 10 bar for some design-point calculations. Before the supercharged process is described in detail, the layout of the reference cycle is explained below.

2. Model of the reference cycle

A conventional natural gas combined cycle (NGCC) without CO_2 capture has been used as reference process for the concepts investigated in this work. The NGCC process consists of the main components: gas turbine (GT) and heat recovery steam generator (HRSG) including HP/IP/LP steam turbines. The NGCC has been modelled using parameters and models agreed upon within the ENCAP project [1]. The main features are summarized in the following section.

The model of the HRSG assumes a triple pressure configuration, where each pressure level (125/30/4.5 bar) possesses an economizer, evaporator and superheater. The live steam is superheated and re-heated (after the HP steam turbine) to 560 °C. The condenser pressure is defined to be 48 mbar. The GT has been modelled as a generic cooled gas turbine. The model assumes that all cooling air is extracted after the compressor exit and is completely mixed with the hot gas coming from the combustion chamber before the expansion in the turbine takes place. The

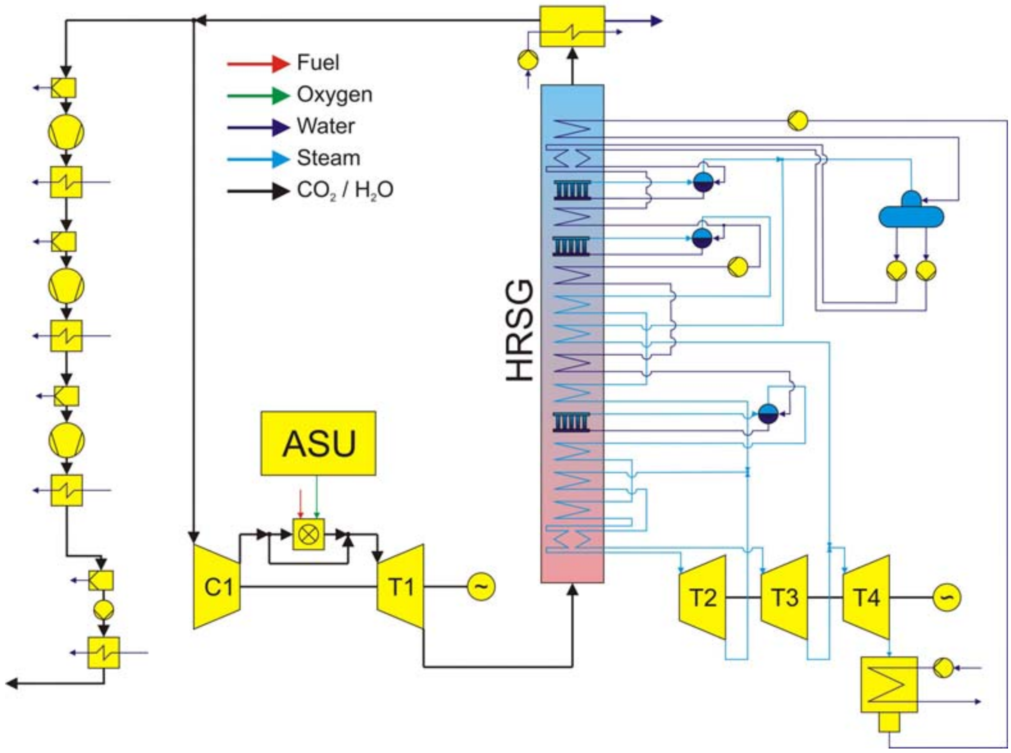

Figure 1 Schematic layout of the semi-closed oxyfuel combined cycle (SCOC) with CO_2 capture.

method corresponds to ISO 2314 [4]. The cooling model has also been used within the ENCAP project [1] and is described in detail by *Jonsson et al.* [5]. For the design point the hot gas temperature is 1425 °C and a pressure ratio of 17 has been assumed for the compressor of the gas turbine.

3. Model of the semi-closed oxyfuel combined cycle

As far as GT and HRSG including HP/IP/LP steam turbines are concerned, the semi-closed oxyfuel combined cycle (SCOC) uses the same process layout as the NGCC. In the SCOC process the fuel is combusted in presence of technically pure oxygen (95 mol-%) provided by a cryogenic air separation unit (ASU). The ASU is described using models and parameters that have been agreed upon within the ENCAP project [1]. Water and carbon dioxide are the main components in the flue gas of the SCOC process. Due to the lack of nitrogen and excess air, another fluid is necessary to control the temperature in the combustor. This is done by recirculated flue gas, which consists mainly of carbon dioxide (89 mol-%) after most of the water resulting from the combustion of natural gas has been condensed out. In the discussed SCOC process approximately 90 % of the CO_2 in the expanded flue gas is recirculated to the GT compressor inlet. The schematic layout of the SCOC process is presented in Figure 1.

The remaining 10 % of the water-depleted flue gas are cooled down further to 19 °C to condense more water out. The cooled CO_2-rich stream is compressed to 110 bar to enable transport and storage. The four staged CO_2 compression is also part of the model. The required compression work is taken into account when the net power output of the overall cycle is calculated.

As a consequence of the changed working fluid, the pressure ratio in the SCOC ($\Pi_{SCOC} = 40$) was chosen significantly higher than in the NGCC ($\Pi_{NGCC} = 17$) to adjust the outlet temperature of the GT. The compressor mass flow is set by the turbine inlet temperature *TIT* and the demand of cooling gas, respectively. In the NGCC this determines the amount of excess air (1.14), while in the SCOC the oxygen flow rate is independent of the cooling gas flow (recirculated CO_2). The excess oxygen is set to 0.01 in the SCOC process.

4. Model of the supercharged SCOC process

The supercharged semi-closed oxyfuel combined cycle (S-SCOC) with CO_2 capture is basically the same process as the (atmospheric) SCOC process. The hardware configuration remains unchanged. Supercharging means the variation of the compressor inlet pressure and, consequently, the outlet pressure of the gas turbine. For design-point calculations the compressor inlet pressure has been increased up to 10 bar. In general, supercharging the SCOC process has the following consequences:

(i) The inlet pressure of the CO_2 compression train increases when supercharging the GT process. Since the compression end pressure is defined as 110 bar, the amount of energy required for CO_2 compression is reduced when the compressor inlet pressure is increased.

(ii) The ASU has to deliver oxygen at an increased combustor pressure – e.g. at 120 bar for a compressor inlet pressure of 3 bar and a pressure ratio of $\Pi = 40$. This higher pressure leads to a drastic increase of the specific work (kJ/kg O_2) required by the ASU. The temperature after turbine (*TAT*) is kept constant.

(iii) When constant geometries (identical flow conditions), speeds and efficiencies are assumed, the power output of a GT depends only on the pressure ratio, on the turbine inlet temperature and on the mass flow. Thus, a simplified assumption is that supercharging a GT does not affect its efficiency and that its power output will be proportional to the mass flow through the turbine, which is determined by the compressor inlet pressure. Of course, the components of the GT (blades and vanes, casing, combustor) need to withstand the increased operating pressure and larger dynamic stresses. The impact of pressure losses may become smaller at supercharged operation. An assessment of secondary losses, such as tip flows in the compressor requires detailed modelling. The same is true for an analysis of heat-transfer mechanisms and cooling air flows at increased operating pressure. The allowable turbine-inlet temperature may be affected by supercharged operation. However, in this work these effects are neglected. This work focuses on the interaction between a supercharged gas turbine, the equipment required for oxyfuel operation (ASU and CO_2 compression train) and the heat recovery steam generator, assuming that a parametric description validated for conventional gas turbines describes supercharged machines with sufficient accuracy as well.

(iv) The overall heat transfer coefficient in the heat exchangers of the HRSG depends on thermophysical properties of the flue gas, in this case of a mixture consisting mostly of CO_2 (see [6-9]) and on the flow conditions, which change differently for both investigated part-load methods. However, the impact of part-load flow conditions on heat transfer is comparable for both processes, see below.

5. Design-point calculations for the NGCC and SCOC processes

Design data and simulation results for the design point of the NGCC and the SCOC processes are shown in Table 1. The CO_2-rich working fluid of the SCOC process results in a higher temperature after the turbine, *TAT*, though the pressure ratio of the gas turbine has been increased from 17 to 40. The *TAT* for the SCOC process is about 50 K higher than that of the NGCC process. However, the heat flow to the HRSG is only slightly higher for the SCOC process because the exhaust mass flow rate and the heat capacity of the flow is smaller than for the NGCC process. A higher live steam temperature could utilise the higher *TAT,* but this option has not been considered. For better comparability, the HRSG arrangement and the live steam conditions remain unchanged for both processes, resulting in different outlet temperatures of the HRSG.

Table 1 Design data and results of design-point calculations for the conventional combined cycle without CO_2 capture (NGCC) and for the semi-closed oxyfuel combined cycle (SCOC) with CO_2 capture.

	NGCC	SCOC
Plant net power P_{net}	386.3 MW	329.7 MW
Plant net efficiency η_{net}	56.6 %	48.3 %
Air separation unit P_{ASU}	–	58.0 MW
CO_2 compression train P_{CO2}	–	17.2 MW
Compressor mass flow $\dot{m}_{C1}$	646.8 kg/s	531.0 kg/s
Pressure ratio gas turbine Π	17	40
Turbine inlet temperature *TIT*	1229.4 °C	1231.8 °C
Temperature after turbine *TAT*	584.5 °C	637.8 °C
Temperature after HRSG t_{HRSG}	99.7 °C	91.7 °C
Exhaust mass flow $\dot{m}_{Exhaust}$	661.6 kg/s	604.6 kg/s
Heat flow to HRSG $\dot{Q}_{HRSG}$	356.9 MW	365.2 MW

6. Impact of supercharging on design-point calculations – calculation for the S-SCOC

Supercharging a gas turbine means that the compressor inlet pressure is increased by a supercharge factor $x_p = p_{compr.\ inlet} / p_a$. The net plant efficiency of the S-SCOC process at different supercharge factors is shown in Figure 2. With the restrictions and the effects discussed above, supercharging results in an almost linear correlation between supercharge factor x_p and efficiency loss.

In design-point calculations the S-SCOC process achieves a slightly lower net efficiency (0.1 %-points for $x_p = 3$; 0.5 %-points for $x_p = 10$) compared to the atmospheric SCOC process. In this study this effect is solely attributed to the impact of supercharging on auxiliary systems. The increase of the total energy consumption of the ASU is larger than the reduction of the total compression work of the CO_2 compression unit, mostly because the O_2 mass flow is larger than the mass flow of CO_2. Thus, the energy demand of the auxiliary units (ASU and CO_2 compression) increases with the supercharge factor. The energy demand for fuel compression is neglected.

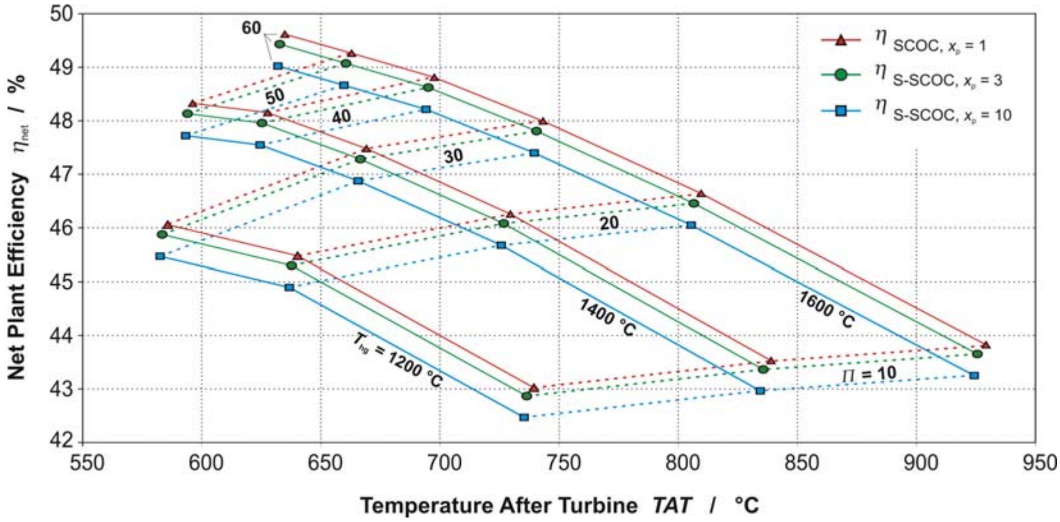

Figure 2 Net plant efficiency of the supercharged semi-closed oxyfuel combined cycle (S-SCOC) with CO_2 capture over temperature after turbine for hot gas temperatures between 1200 and 1600 °C and pressure ratios from 10 to 60 for different supercharge-pressures.

7. Variable-supercharge gas turbine at part load

In general, if an atmospheric gas turbine process is operated at part-load, the compressor mass flow rate is reduced by closing the variable inlet guide vanes (VIGV) of the compressor. The reduced mass flow rate is combined with a smaller pressure ratio of the compressor to keep the changes of the volumetric flow rate as small as possible. If the GT operates in a combined cycle the hot gas temperature is lowered to achieve a constant GT exhaust gas temperature. Both the reduced pressure ratio and the decreased hot gas temperature lead to a significantly reduced efficiency. This part-load behavior has initiated the idea of a sliding supercharge of the GT process.

The supercharged cycle is operated with sliding compressor inlet pressure (variable supercharge) to keep the volumetric flow rate constant for different mass flow rates. Hence, the supercharged process exhibits a constant pressure ratio and a constant exhaust temperature at the same time. With the assumptions discussed above, the efficiency of the GT process remains nearly constant. The influence of sliding pressure operation on heat transfer in the HRSG has been investigated in some detail. In a conventional HRSG operated at atmospheric pressure, a lower flue-gas mass flow rate at part load leads to a reduced flow velocity as a consequence of the decreased volumetric flow rate (density stays constant). For the S-SCOC process the flow velocity of the exhaust stream remains constant due to sliding pressure operation, but at the same time the density changes because of the sliding pressure. Thus, flow conditions for both processes (SCOC: constant density, volumetric flow rate changes; S-SCOC: vice versa) lead approximately to same results of the Reynolds number

$$\mathrm{Re} = \frac{c \ \rho \ l}{\eta} \ . \qquad \text{(Eq. 1)}$$

Though the conditions for the atmospheric SCOC process and for the S-SCOC process with sliding pressure are different at part-load, their impact on the heat transfer in the HRSG is very similar.

In our calculations a steam temperature of 600 °C has been used for the part-load model of the oxyfuel process. The part-load model of the HRSG has been simplified for better convergence. Heat exchangers are modelled separately and pressure losses in the water/steam cycle have been neglected. A dual pressure HRSG has been considered instead of the triple pressure HRSG described above. All part-load investigations are related to this simplified model, which achieves an efficiency reduced by 1.3 %-points at design conditions compared to the S-SCOC process described above.

8. Discussion of simulation results

ENCAP results for the NGCC process without CO_2 capture and for the SCOC process with CO_2 capture have been used as reference processes for the present study. The results for design-point calculation of the NGCC process agree well with the results from the ENCAP project, see Figure 5. Minor differences are related to the different thermodynamic property sets used within the simulation software. The net efficiency of the SCOC process from this work is slightly higher (0.6 %-points) than the result from ENCAP. In ENCAP, a double pressure HRSG has been considered instead of a triple pressure HRSG. At full load the calculated net efficiency is reduced by 8.3 %-points due to the capture of CO_2.

Although the pressure ratio has been increased from 17 to 40 for the SCOC process, the exhaust gas temperature raises by more than 50 K due to the changed working fluid. The steam process becomes more important in the SCOC process. In our model the power output of the steam cycle increases slightly due to more favourable conditions in the HRSG. A further increase would be possible if live steam conditions were adapted to the higher turbine outlet temperature.

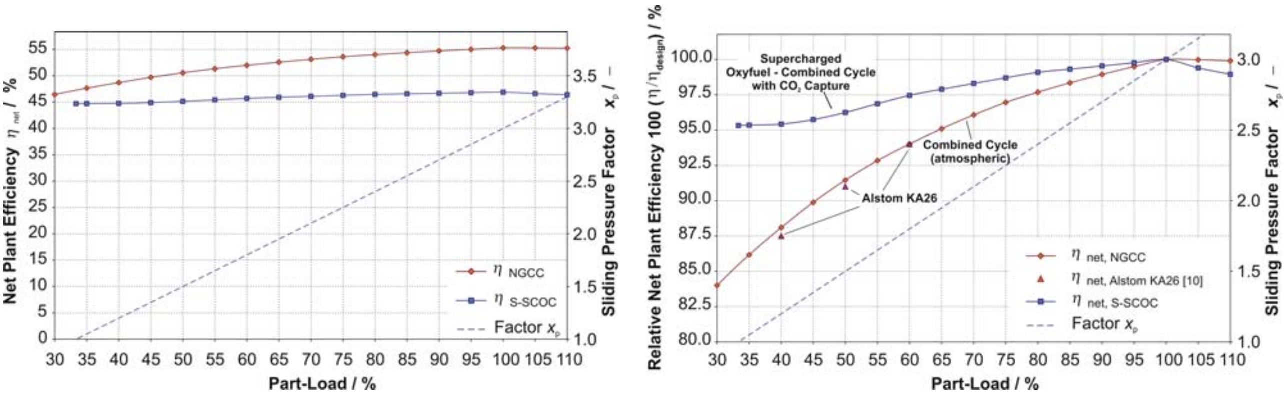

Figure 3 Net plant efficiency absolute (left hand side) and relative to full load (right hand side) vs. part-load for the NGCC, the Alstom KA26 and the S-SCOC process.

For part-load investigations, x_p = 3 has been assumed for the design point to allow for sliding pressure operation down to 33 % load without running gas paths at pressures below atmospheric pressure. The simplified NGCC model achieves a net plant efficiency of 55.4 %. The S-SCOC process results in a net plant efficiency of 46.9 % using the part-load model. Figure 3 shows results for part load both as absolute efficiency and relative to full load. Literature values [10] for the relative net plant efficiency of the Alstom KA26 combined cycle (right hand side of Figure 3) agree well with the obtained results.

At 50 % part-load the S-SCOC efficiency is only lowered to 45.1 % (corresponding to 96.2 % of the design-point efficiency). The reduction is almost exclusively caused by the steam cycle because the GT is hardly affected over the whole load range. Together with the assumptions discussed above Figure 4 explains this result for the GT. The left hand side of Figure 4 shows that the turbine inlet temperature *TIT* stays constant over the full load range. The *TIT* of the conventional CC decreases by approx. 200 K from full load to 50 % part-load. The reduced mass flow rate in the NGCC results in a smaller pressure ratio in the compressor, which subsequently requires a lower *TIT* to keep the exhaust temperature unaffected. The combination of reduced turbine inlet temperature and smaller pressure ratio leads to the reduction of GT efficiency.

Part-load operation with sliding pressure leads to a constant volumetric flow rate. On the right hand side Figure 4 shows the volumetric flow rate versus the load. The volume flow is approx. 9 times lower for the S-SCOC mainly due to the higher pressure at the outlet of the combustion chamber (with x_p = 3 and Π = 40 the pressure at the outlet of the combustion chamber becomes 118 bar while the corresponding pressure for the NGCC process with Π = 17 is 16.5 bar) and stays constant in contrast to the NGCC process where the pressure ratio is reduced in part-load.

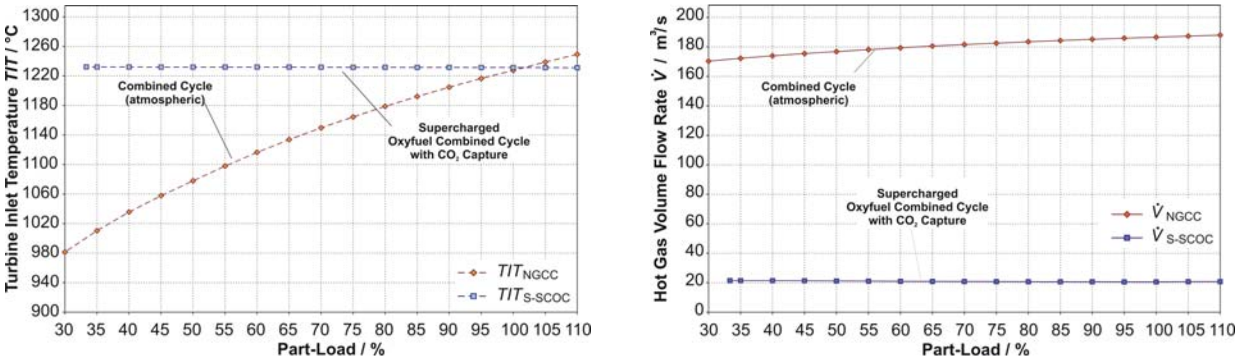

Figure 4 Turbine inlet temperature (left hand side) and volumetric flow rate after compression (right hand side) versus load for the NGCC and the S-SCOC processes.

9. Conclusions

A model for a cooled gas turbine was set up in this work. A triple pressure HRSG was modelled and combined with the GT model to describe a NGCC process. Based on the NGCC process a model for a SCOC process with CO_2 capture was developed. Results from ENCAP were used to evaluate the simulation results of this work. As a novel configuration a supercharged SCOC process (S-SCOC) has been presented. Design-point and part-load simulations have been carried out. For design-point calculations the compressor inlet pressure has been increased up to 10 bar; supercharging factors up to $x_p = 3$ are considered realistic and were investigated in more detail. If a S-SCOC is designed for $x_p = 3$ at 100 % load it can be operated down to 33 % load without lowering gas-path pressures below atmospheric pressure. Sliding pressure operation allows for a superior performance at part-load. Figure 5 summarizes results for both design-point and part-load simulations. The graph on the left hand side shows the verification for the gas turbine in simple cycle, the NGCC and the SCOC process. The SCOC process achieves a net efficiency that is 0.6 %-points higher than calculated for the corresponding process in ENCAP because the HRSG has been modelled as triple pressure HRSG in this study while ENCAP assumed a dual pressure HRSG.

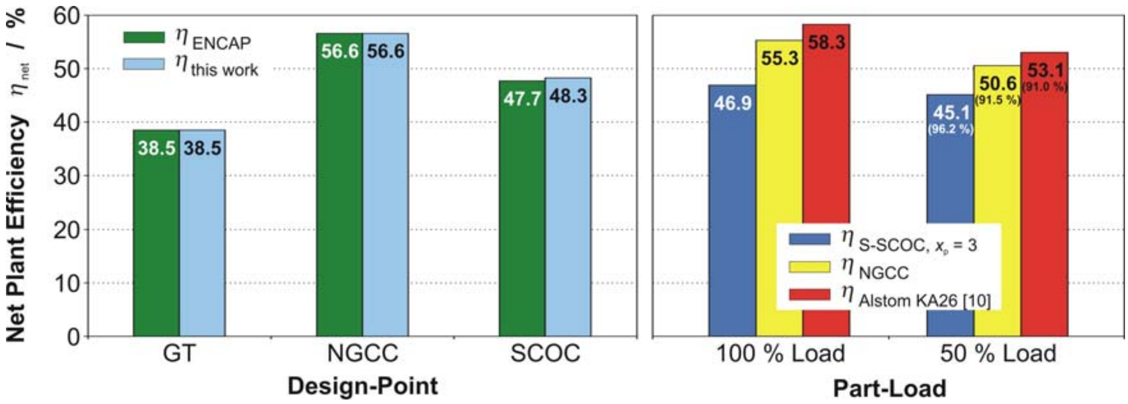

Figure 5 Net plant efficiency in design-point for the detached GT, the NGCC and the SCOC with CO_2 capture in comparison with ENCAP results (left) and net plant efficiency in full load and at 50% part-load of the supercharged S-SCOC, the NGCC and manufacturer values (right).

Part-load simulations underline the positive effect of sliding pressure operation for the S-SCOC process. Relative part-load efficiencies are substantially higher than for the NGCC and the SCOC process. The part load behavior of the steam cycle is similar for all three processes. Since the heat transfer on the flue gas side varies in a similar way for both processes it is assumed that established control strategies for the HRSG can be adapted to supercharged operation. It should be pointed out that this study did not address the adaptation of the actual gas-turbine design to supercharged operation and the resulting efficiency changes. Important aspects like changing relative pressure losses, tip flows or allowable turbine-inlet temperatures need to be addressed in further studies.

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