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EVALUATION OF sCO₂ POWER CYCLES FOR DIRECT AND WASTE HEAT APPLICATIONS

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ABSTRACT

Supercritical CO₂ (sCO₂) power systems may improve cycle efficiencies of future power plants significantly while reducing size and costs of the related components. This paper provides a thermodynamic evaluation of sCO₂ power cycles compared to conventional power cycles based on water/steam. Different plant layouts for direct and waste heat applications are assessed and the impact of process parameters such as upper and lower process temperatures is evaluated by variational calculations. The results are compared with reference values based on water/steam derived from a state-of-the-art concentrated solar power plant and combined cycle power plants respectively. An optimized cycle layout results in significant efficiency advantages of the sCO₂ power cycles. Regardless of the application, the investigations show that these advantages are strongly dependent on the characteristic values of the components such as TTD, pressure losses and expansion efficiencies.

INTRODUCTION

Power generation based on supercritical carbon dioxide (sCO₂) Brayton cycles has been the object of numerous research studies during the last years. Improved cycle efficiencies and lower component costs compared to conventional Rankine cycles are claimed in various publications [1-4]. Depending on the application type, different cycle layouts have been proposed in literature. A recently published, comprehensive review of investigated cycle configurations is given in [5]. For direct heated applications such as coal fired, concentrated solar or nuclear power plants recompression cycles have been identified as most beneficial [3, 6]. For waste heat applications and combined cycle power plants cascaded cycles are the best choice for high cycle performance [4, 5]. However, theoretically calculated cycle efficiencies are highly dependent on assumed characteristic values such as turbine efficiencies, pressure losses and terminal

temperature differences and may not necessarily reflect economics and practical performance limits. The comparison of published cycle efficiencies with reference values based on water/steam is thus often difficult.

This paper evaluates for both, primary and waste heat applications, the performance of sCO₂ cycles in comparison with state-of-the-art water steam cycles. Basic design estimations such as turbine design are carried out and compared with the equipment used in the reference cycles. The sensitivities of characteristic values on performance and equipment are investigated.

SIMULATION AND MODELING

Thermodynamic cycle simulations were carried out by using the simulation software Cycle-Tempo [7] in combination with REFPROP property tables [8] considering the physical properties of CO₂ and water/steam. The performance values of the reference cycles based on water/steam were calibrated to results derived by SIEMENS in-house simulation software. Initial sCO₂ turbine design calculations were performed with internal software using a simplified fluid property model [8]. In the operating range of the turbines, i.e. the gaseous single phase region, this model, which is actually valid for gaseous fluid mixtures, is in good agreement with the properties derived by REFPROP.

DIRECT HEAT APPLICATIONS

A 150 MW solar tower power plant was chosen as reference for direct heated application. The general cycle layout is shown in figure 1. It consists of three LP preheaters, one deaerator and two HP preheaters. The turbo set consists of high efficient SST700 and SST900 steam turbine modules. Live steam and reheat temperature were varied between 545 °C 605 °C. Live steam and reheat pressures are 140 bar and 27 bar respectively. The HP pressure loss across boiler and main steam piping is approximately 12 % and reheat pressure loss 10 %.

The terminal temperature difference of the condenser was set to 3 K. Cooling water inlet temperatures were varied between 10 °C and 45 °C.

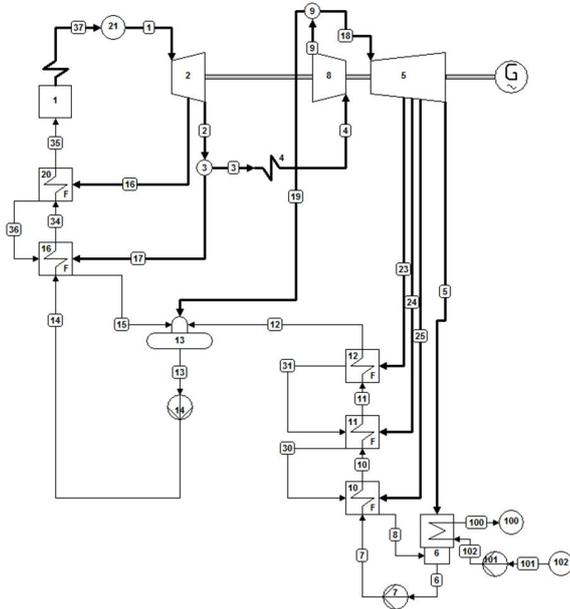


Figure 1: Heat balance diagram of reference cycle, a 150 MW CSP Rankine cycle, modeled with Cycle-Tempo.

The thermodynamic cycle of a simple recuperated, reheated $s\text{CO}_2$ Brayton cycle is illustrated in figure 2. Comparing the mean temperatures of heat input and heat sink for the Brayton and the reference Rankine cycle the advantage of direct heated Brayton cycle becomes obvious.

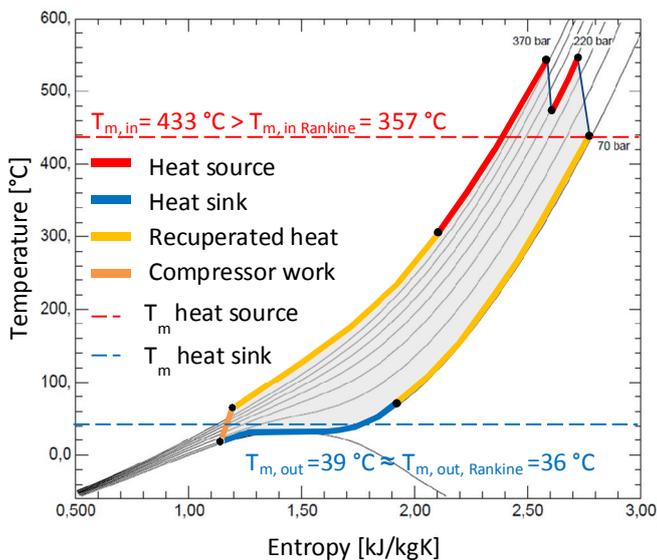


Figure 2: T-s diagram of recuperated $s\text{CO}_2$ cycle for a reheat power cycle with 545 °C as live steam and reheat temperature.

While the mean temperature of the heat sink differs only marginally, the mean temperature of the heat source is significantly higher in the Brayton cycle. Introducing the Carnot efficiency according to equation 1 the idealized benefit of the $s\text{CO}_2$ cycle can be expressed in a $\Delta\eta_{\text{Carnot}} \approx 5\%$ for this case.

$$\eta_{\text{Carnot}} = \left(1 - \frac{T_{\text{Heat Sink}}}{T_{\text{Heat Source}}}\right) \quad (1)$$

However, for comparing the Brayton cycle with a Rankine cycle and thus for assessing the thermodynamic benefits, the exergy losses of the process have to be considered. The characteristic values such as pressure losses, turbine and compressor efficiencies and terminal temperature differences (TTD) of the heat exchangers which were used in initial cycle calculations are summarized in table 1. The values for pressure losses, TTD and compressor efficiency were chosen in line with published data [3, 4, 6]. A sensitivity study of these values as well as first turbine design considerations, from which the turbine efficiency was derived, is presented in detail later in this paper.

Two cycle architectures were considered. A simple recuperated cycle with reheat is shown in figure 3 and a recompressed, reheated cycle with single stage intercooling, which is shown in figure 4. The inlet pressure of the first turbine was set to 370 bar and the lower pressure level at the compressor inlet was set to 70 bar. The initial recompression ratio is 25 % which means that only 75 % of the total mass flow is led to the heat sink.

Table 1: Characteristic values for efficiencies, pressure losses and terminal temperature differences used for the initial thermodynamic cycle calculations.

Parameter	Value	Unit
Pressure loss recup., boiler, reheat, cond.	2	%
Terminal temperature difference heat sink	3	K
Terminal temperature difference recup.	5	K
Efficiency turbines	91	%
Efficiency compressors	85	%
Efficiency generator	98,25	%

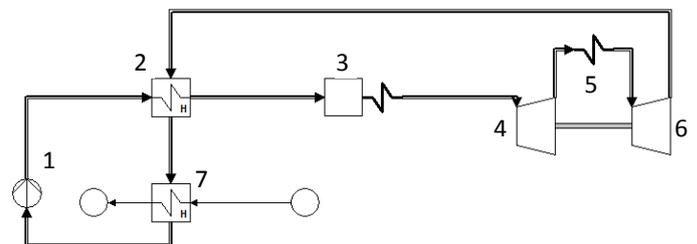


Figure 3: Layout of a simple recuperated, reheated $s\text{CO}_2$ cycle. 1-compressor, 2-recuperator, 3-boiler, 4-high pressure turbine, 5-reheat, 6-reheat turbine, 7-heat sink.

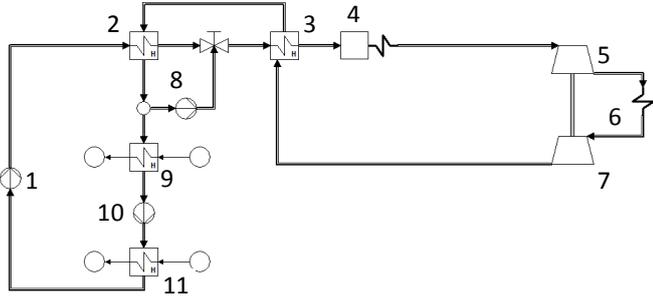


Figure 4: Layout of a recompressed, intercooled and reheated sCO₂ cycle. 1-compressor, 2-low temperature recuperator (LTR), 3-high temperature recuperator (HTR), 4-boiler, 5-high pressure turbine, 6-reheater, 7-reheat turbine, 8-compressor, 9-heat sink, 10-compressor, 11-intercooler.

Since the cycle performance depends significantly on the lower process temperature, cycle calculations were carried out with varying cooling water inlet temperatures for both the water/steam cycle and the sCO₂ cycle. As the cooling water temperature rises, the losses in the condenser and compressor also increase, as these process steps are completely shifted to the right side of the critical point. By increasing the back pressure, this can be corrected and the losses can be reduced. For simplification the calculations were carried out with constant expansion and compression efficiencies. In figure 5 the deviation of the sCO₂ net efficiency values from the reference values based on water/steam are shown.

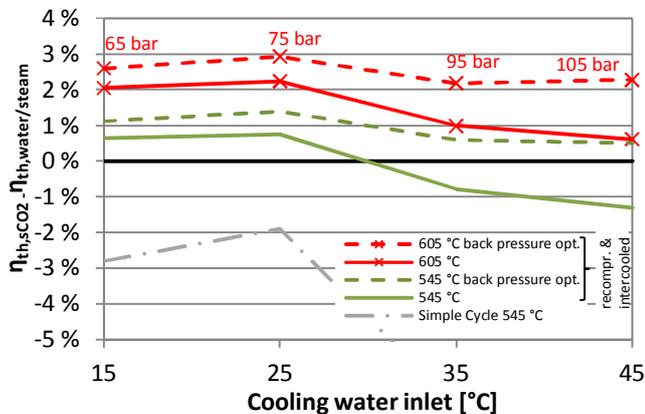


Figure 5: Deviations of net efficiency values compared to reference values based on water/steam. The initial back pressure was 75 bar. The optimized back pressures are given for 605 °C. Further process parameters were kept constant.

The main contributor for the significant lower performance of the simple recuperated Brayton cycle compared to the reference values is the large exergy loss in the recuperator based differences in isobaric specific heat capacity of the hot and cold stream. By introducing the recompression cycle these losses can be reduced significantly which increases the overall process efficiency as shown in figure 6. This improvement is,

however, associated with an increase of the heat transfer surface. A sensitivity study is described later in this paper.

However, with increasing cooling water temperature and thus with increasing compressor inlet temperature the efficiency drops considerably because the thermodynamic condition at the compressor inlet changes from a “as-liquid-state” to an “as-gaseous state”.

Thus, the compressor work and losses as well as the exergy losses in the heat sink are increasing significantly. This effect can be compensated by increasing the lower pressure level of the process, i.e. the compressor inlet pressure. In figure 5 the calculation points for the optimized backpressures (dashed line) are labeled with the chosen pressure values.

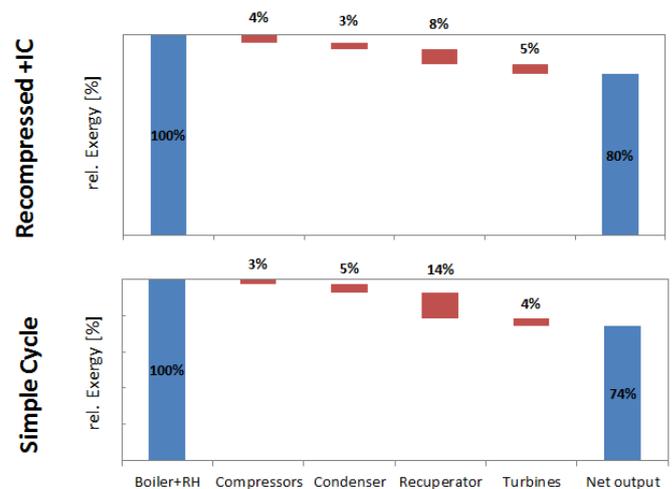


Figure 6: Exergy losses in a simple recuperated cycle and in a recompressed, intercooled cycle. The amount of supplied energy in the boiler and reheater is set to 100 %.

The impact of the upper and lower process pressure on the thermal efficiency is shown in figure 7 exemplarily for a turbine inlet temperature of 605 °C and a cooling water inlet temperature of 25 °C. In this case, the thermodynamic condition of the compressor inlet is close to the liquid phase as shown in figure 2 leading to an optimum backpressure of approximately 75 bars.

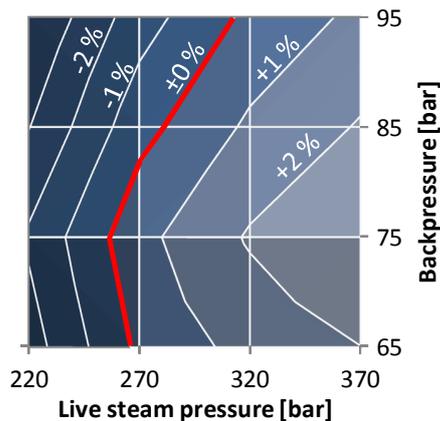


Figure 7: Deviation of net efficiency values compared to reference values based on water/steam depending on upper and lower process pressure for a design point of 605 °C turbine inlet temperature and 25 °C cooling water temperature.

It can be also concluded from figure 7 that the turbine inlet pressure has to be higher than approximately 255 bars to get a performance benefit compared to the reference performance based on water/steam. With decreasing inlet pressure the recuperated heat is increasing and thus the exergy loss rises significantly.

The sCO₂ turbine modules were derived from the standard steam turbine product portfolio. Both, the high pressure turbine and the reheat turbine are high efficient barrel type turbines as shown in figure 8. Due to the small enthalpy drop in the Brayton cycle the number of stages and thus the axial length is significantly lower compared to the reference turbines. Whereas the high pressure inlet volume flow is even higher compared to the water/steam cycle, the exhaust volume flow at the lower pressure side is considerably smaller compared to the exhaust flow of the low pressure steam turbine, which leads to smaller component sizes at the cold end. However, the high pressure level in the sCO₂ turbines leads to large wall thicknesses in casings, valves and piping in comparison to the steam turbines. Improvements concerning performance and economics can be gained from the implementation of advanced sealing technologies such as dry gas seals and the development of a single casing turbine design and must be addressed in future development activities.

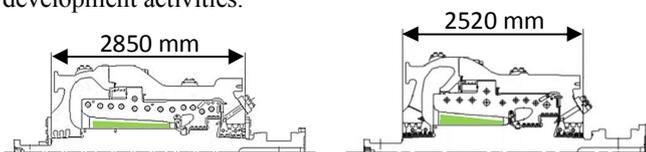


Figure 8: Illustration of high pressure sCO₂ turbine (left) and reheat turbine (right).

Table 2: Comparison of enthalpy differences as well as volume flows at the inlet and the outlet of the turbines in the CO₂- and the reference cycle based on water/steam.

Turbine		Water/Steam	sCO ₂
HP	\dot{V}_{inlet} [m ³ /s]	2,7	4,6
	\dot{V}_{outlet} [m ³ /s]	9,3	8,5
	Δh [kJ/kg]	415	108
Reheat	\dot{V}_{inlet} [m ³ /s]	13,5	10
	\dot{V}_{outlet} [m ³ /s]	1841,5	18
	Δh [kJ/kg]	1210	97

A central component in a supercritical CO₂ Brayton cycle is the recuperator which transfers the heat from the exhaust flow of the turbine to the high pressure flow before it enters the boiler. The thermodynamic effect on the cycle efficiency is comparable to the feedwater preheating in a Rankine cycle. However, the amount of transferred heat is significantly higher in the Brayton cycle. In order to achieve a good overall cycle efficiency the TTD in the recuperator has to be kept small leading to a considerable high transfer capability (kA values) and thus to high heat transfer surfaces. Thus, the necessity of high efficient heat exchangers with high surface densities and low specific costs becomes obvious. In table 3 the characteristic data of the recuperating/ preheating can be compared.

Table 3: Comparison of recuperated heat and transfer capability in the CO₂ and the reference cycle based on water/steam.

	Water/Steam	sCO ₂ Recompressed & Intercooled
Recuperated heat [MW]	79	505
kA _{LP/LTR} [MW/K]	3,8	23
kA _{HP/HTR} [MW/K]	3,4	10,5

Based on these results, the impact of the recompression bypass ratio and TTD on both the cycle efficiency and the heat transfer surface can be studied. The heat transfer surfaces in the recuperators were approximated by applying Nusselt-correlations for counter-flow tube-in-tube-heat exchangers. Lowering the bypass value from the initial value of 25 % by e.g. 5 % leads to an efficiency drop of approximately 1,1 % as shown in figure 9. On the other hand with this reduction the heat transfer surface in the recuperator can be reduced by approximately 30 % as illustrated in figure 10.

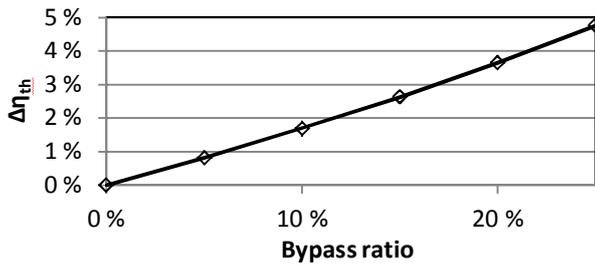


Figure 9: Deviation of the cycle efficiency as a function of the recompression bypass ratio.

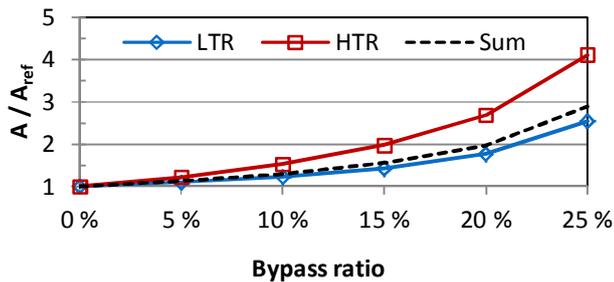


Figure 10: Deviation of the relative heat transfer surface in the low temperature heat exchanger (LTR) and the high temperature heat exchanger (HTR) as a function of the recompression bypass ratio.

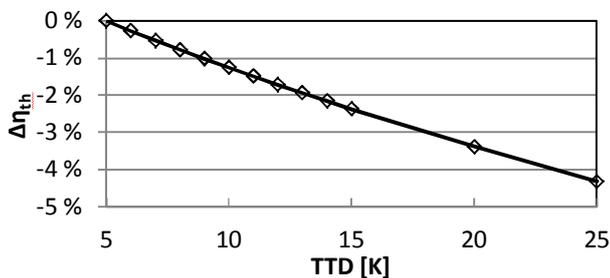


Figure 11: Deviation of the cycle efficiency depending on the terminal temperature difference

The sensitivity of the cycle efficiency and the heat transfer surface depending on the chosen terminal temperature difference is shown in figure 11 and figure 12 respectively. Increasing the terminal temperature difference from the initial value of 5 K to e.g. 10 K leads to an efficiency drop of approximately 1,3 % whereas the heat transfer surface could be reduced by about 40 %.

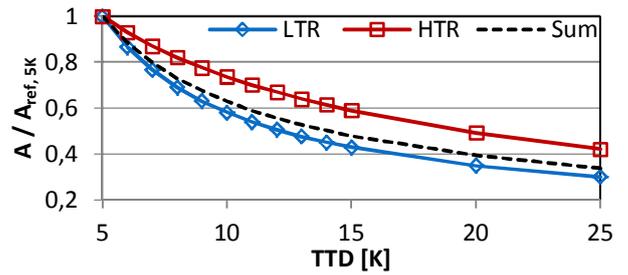


Figure 12: Deviation of the relative heat transfer surface in the low temperature heat exchanger (LTR) and the high temperature heat exchanger (HTR) as a function of the terminal temperature difference.

The initial pressure loss for each recuperator and the boiler was set to 2 % as listed in table 1. Thus, the pressure loss of the high pressure flow amounts to 6 % compared to 12 % in the reference cycle as written before. Assuming the same relative pressure loss of 12 % leads to an efficiency drop of approximately 1,1 % as shown in figure 13. Keeping in mind the fact of the higher pressure level and the larger volume flow at the turbine inlet, the necessity of a compact plant layout with short pipe length becomes obvious.

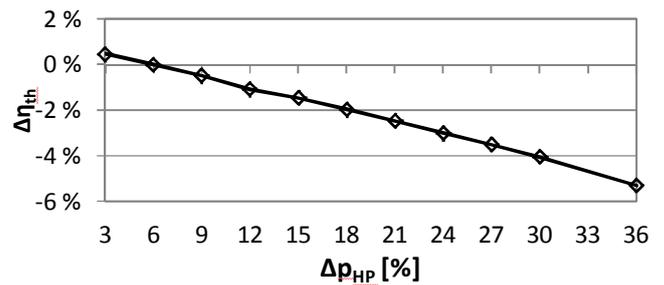


Figure 13: Deviation of the cycle efficiency depending on the assumed pressure loss of the high pressure flow (compressor outlet to HP turbine inlet).

In conclusion it can be stated that a supercritical Brayton cycle may lead to a significant performance benefit for the investigated 150 MW CSP application. Mandatory prerequisite is an optimized cycle architecture (recompressed and intercooled) with small terminal temperature differences, small pressure losses, high efficient turbo machinery, high turbine inlet pressures and optimized backpressures depending on the cooling conditions. Furthermore an increasing benefit with increasing upper process temperature can be noted. However, the shown performance benefit for the sCO₂ cycle cannot be generalized. For applications with high power output e.g. large coal fired power plants where supercritical Rankine cycles are beneficial, the results might be different and should be evaluated separately.

The necessary economic assessment and optimization of a sCO₂ power plant for a solar application needs to be done in a subsequent step considering the overall plant configuration including solar field, receiver etc.

WASTE HEAT APPLICATIONS

Two Combined Cycle Power Plants were chosen for the assessment of sCO₂ bottoming cycles in comparison to reference cycles based on water/steam. In the first reference cycle a dual pressure Rankine cycle recovers the waste heat coming from an aero derivative Trent60 gas turbine. The exhaust gas temperature is 431 °C. The second case is based on an SGT800 gas turbine with an exhaust gas temperature of 567 °C. Table 4 summarizes the design parameter of the two reference cycles.

Table 4: Design parameter of the reference cycles.

Parameter	Trent60	SGT800
Exhaust gas temperature [°C]	431	567
Exhaust gas mass flow [kg/s]	151,4	269,4
TTD HRSG [K]	10	10
Pressure loss HRSG HP part [%]	13,0	11,6
Bottoming cycle net output [MW]	14,6	50,0
Efficiency generator [%]	100	100

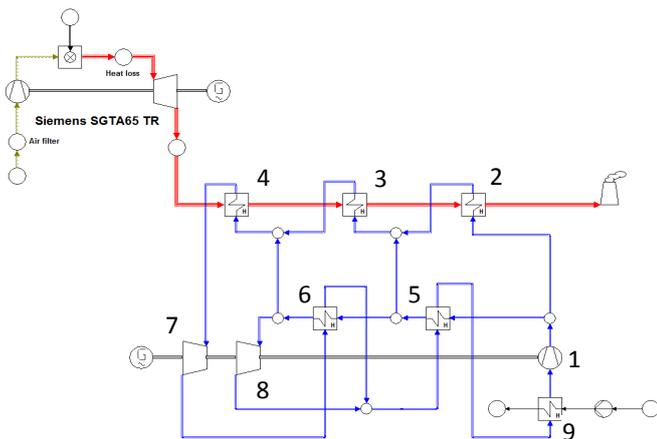


Figure 14: Layout of a dual split cascaded sCO₂ cycle. 1-compressor, 2, 3, 4-waste heat exchanger, 5-low temperature recuperator (LTR), 6-high temperature recuperator (HTR), 7-main turbine, 8-secondary turbine, 9-heat sink.

Cascaded sCO₂ cycles have been identified for high cycle performance in waste heat applications. In comparison to Rankine cycles the exergy losses in the heater can be minimized leading to a performance benefit of the bottoming cycle [4, 5, 9]. In figure 14 the cycle layout is shown exemplarily for the Trent60 bottoming cycle. It consists of one compressor, two recuperators, two turbines, one cooler and three heaters. After the compression, the fluid is split to two separate streams. The main stream passes through the heaters which recover the heat from the exhaust gas. After the expansion in the main turbine, it flows through two

recuperators where the heat of the main turbine exhaust is transferred to the second stream which expands in the secondary turbine.

Since the specific heat of sCO₂ decreases with increasing temperature, the exergy losses in a single heater would be significantly high. By splitting up the heater and adding split flows from the second stream the temperature differences across the heat exchangers and thus the exergy losses can be minimized as shown in figure 15. The corresponding TTD as well as the TTD of the recuperators, the assumed pressure losses, turbine and compressor efficiencies and thermodynamic parameters for both the Trent60 and the SGT800 are listed in table 5.

Table 5: Design parameters of the sCO₂ cycles.

Parameter	Trent60	SGT800
Back pressure [bar]	75	65
Cooling water inlet temperature [°C]	15	15
Pressure loss heaters, recu., cond. [%]	2	2
TTD waste heat exchanger [K]	10	10
TTD HTR [K]	10	10
TTD heat sink [K]	5	5
Turbine inlet pressure [bar]	220	255
Turbine inlet temperature [°C]	387	516
Efficiency main turbine [%]	86,2	89,2
Efficiency secondary turbine [%]	79,5	86,2
Efficiency compressors [%]	80	80
Efficiency generator [%]	100	100

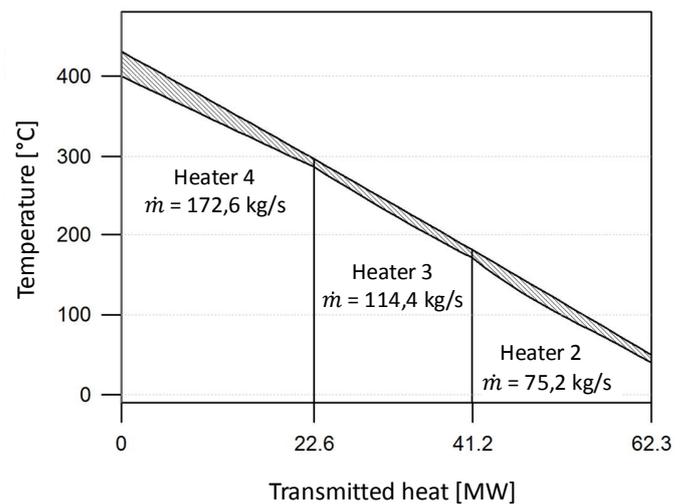


Figure 15: T-Q Diagram of heat recovery in a dual split sCO₂ cycle (example: Trent60). The diagram contains the CO₂ mass flow values in each heater.

The turbine inlet pressures and temperatures were optimized based on variational calculations. The turbine efficiencies were derived from first design calculations which are described subsequently. The influence of varying pressure

losses and TTD were investigated which is also presented in detail later in this paper.

The exergy analysis for both, the cascaded sCO₂ cycle and the reference cycle illustrates the potential of the cascaded sCO₂ cycles, i.e. the significant lower exergy losses in heaters/HRSG and stack as shown in figure 16. The benefit is partly compensated by the losses in the recuperator, however with the assumed parameters for TTD and pressure losses a performance benefit can be noted for the Trent60 case at 25 °C cooling water inlet temperature.

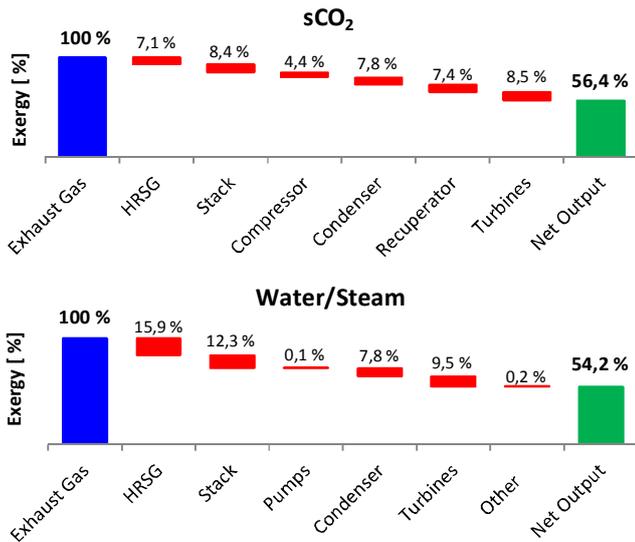


Figure 16: Comparison of relative exergy losses of the cascaded sCO₂ cycle and the dual pressure reference cycle. The calculations are based on the Trent60 CCPP with a cooling water inlet temperature of 25 °C.

The CO₂ turbine modules for both the Trent60 and the SGT800 bottoming cycle were derived by applying scaling laws on high pressure steam turbine modules. For each application dual casing turbines were considered each connected with a gear box. Since the power range and thus the volume flows of both cases are small, the resulting efficiencies are significantly lower compared to the previously described, direct heated application. The dominant losses in these small scaled turbine cylinders are the leakage losses related to both, the bladepath and the shaft sealings. Thus, appropriate measures for improving the performance need to be addressed in further development activities. Potential measures are the application of innovative sealing technologies and new turbine concepts such as single casing turbines in order to reduce the number of shaft sealings whereby the second measure would make a considerable contribution to cost savings.

Cycle calculations were carried out with varying cooling water inlet temperatures for both the water/steam cycles and the sCO₂ cycles. For simplification purpose, the impact of the ambient conditions on the gas turbine output and exhaust gas conditions were neglected. Furthermore the turbine efficiencies

for both the sCO₂- and the reference cycles were kept constant. In figure 17 and 18 the deviation of the sCO₂ power output and the combined cycle efficiency values from the reference values based on water/steam are presented for the Trent60 and the SGT800 case respectively.

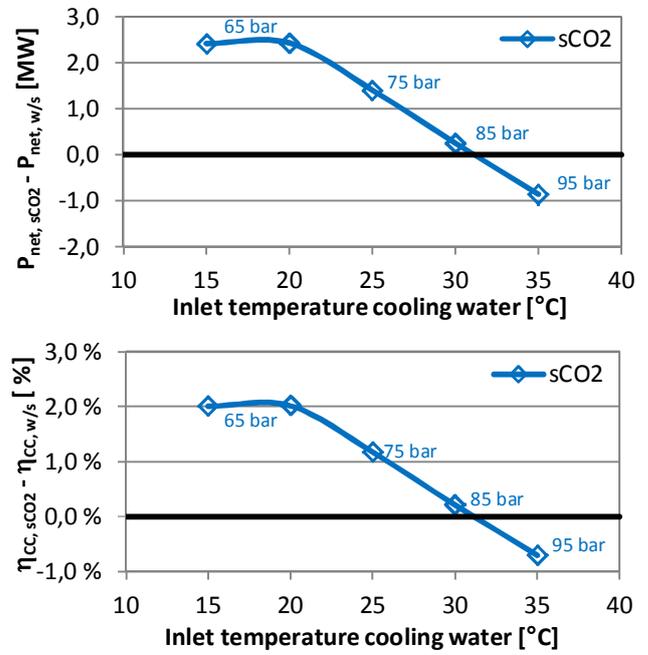


Figure 17: Deviation of the power output and cycle efficiency values compared to reference values based on water/steam (Trent60). Only the back pressure was optimized at each cooling water temperature.

For both applications, a performance benefit at cold cooling water inlet conditions can be determined. With increasing cooling water temperature and thus with increasing compressor inlet temperature the efficiency benefit drops considerably leading to a performance disadvantage at high cooling water inlet temperatures even with optimized backpressures in the sCO₂ cycle. In analogy to the direct heated application this can be explained by the thermodynamic condition at the compressor inlet which changes from a “as-liquid-state” to an “as-gaseous state” when the inlet temperature is exceeding the critical temperature of the fluid.

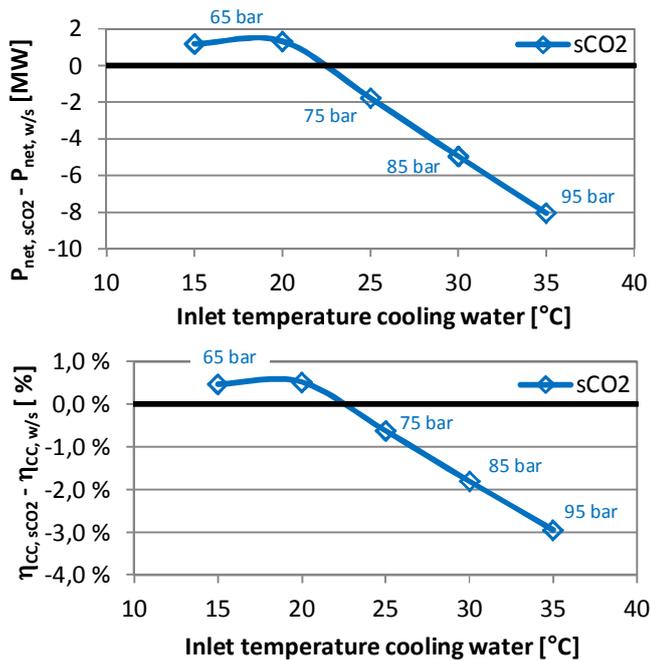


Figure 18: Deviation of the power output and cycle efficiency values compared to reference values based on water/steam (SGT800). Only the back pressure was optimized at each cooling water temperature.

Comparing figure 17 and figure 18 it is obvious that a sCO_2 bottoming cycle is more beneficial for the Trent60 than for the SGT800. Introducing the exergy loss analysis for the SGT800 bottoming cycles as shown in figure 19, it can be concluded that the exergy losses in HRSG and stack for the reference cycle of the SGT800 are clearly lower compared to the Trent60 reference cycle (figure 16). A similar result can be found in [10] i.e. the potential of a sCO_2 bottoming cycle in a combined cycle power plant is decreasing with increasing temperature level of the exhaust gas.

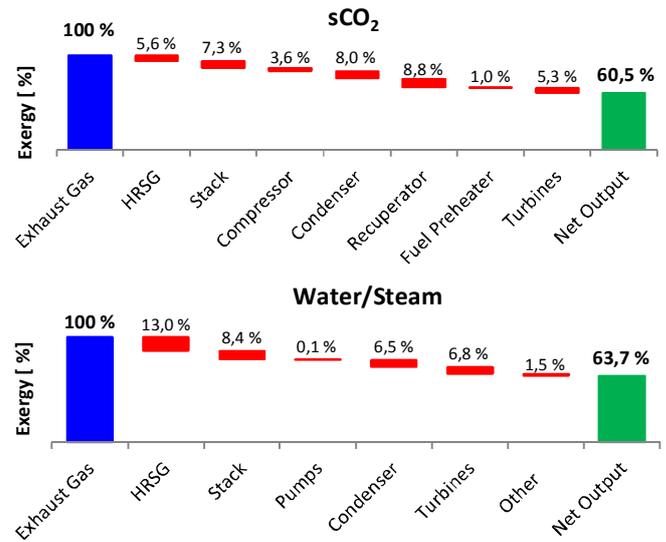


Figure 19: Comparison of relative exergy losses of the cascaded sCO_2 cycle and the dual pressure reference cycle. The calculations are based on the SGT800 CCPP with a cooling water inlet temperature of 25 °C.

The above discussed performance results are based on the assumed parameters listed in table 5. The resulting heat transfer surfaces of the heaters, the recuperators and the condenser were approximated by applying Nusselt-correlations for cross-flow shell-and-tube-heat exchangers. The comparison of the heating surface of the sCO_2 cycle and the reference cycle as well as the impact of varying the TTD of HRSG and recuperator are shown in figures 20 and 21 respectively.

It is obvious that a TTD of 10 K combined with an optimized dual split cascaded heater leads to small temperature differences and thus to heating surface which are approximately three times higher than the heating surfaces in the HRSG of the reference cycle. By doubling the TTD in the heater the heating surface can be nearly halved, however associated with a performance loss of approximately 1 MW. Equivalently, increasing the TTD of the recuperator from 10 K to 20 K the related heating surfaces can be almost halved but a significant performance loss of approximately 1,3 MW has to be taken into account.

Furthermore, a variation of the pressure loss within the heat exchangers was carried out. The result is shown in Figure 22. Low pressure losses, as specified in table 5, lead to low flow velocities and thus to larger heat transfer surfaces compared to water/steam. Therefore, a smaller surface area is required for higher pressure losses. As the analysis of the deviation of the power output values shows, low losses are nevertheless necessary to generate a performance benefit with sCO_2 . As a result, doubling the pressure loss of all heat exchangers on the high and low pressure side of the cycle reduces the advantage by more than 1 MW. In comparison, a corresponding change in the pressure drop only on the high pressure side reduces the performance advantage approximately by half.

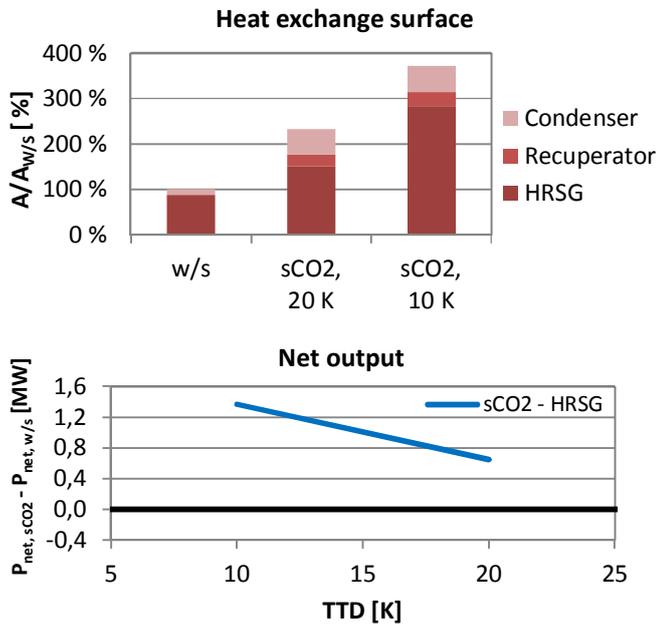


Figure 20: Heat transfer surface and deviation of the power output values depending on the TTD of the HRSG compared to reference values based on water/steam.

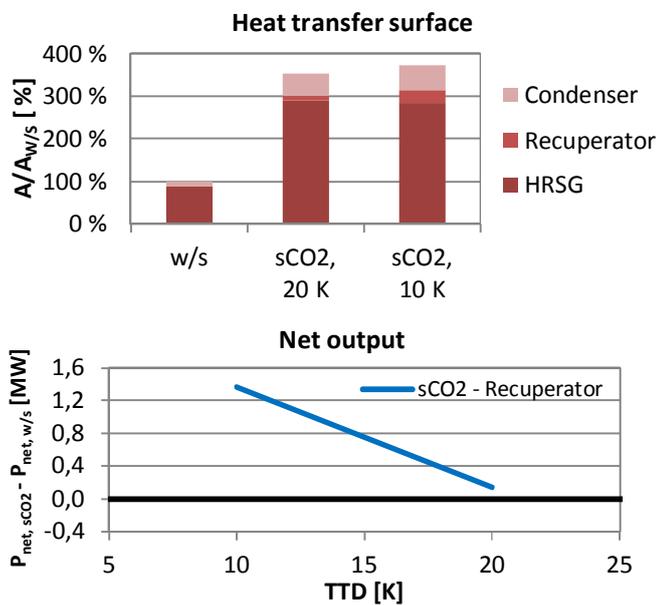


Figure 21: Heat transfer surface and deviation of the power output values depending on the TTD of the recuperator compared to reference values based on water/steam.

In summary, it can be emphasized that supercritical sCO₂ Brayton cycles may improve the performance of waste heat recovery processes, whereby the performance potential is decreasing with increasing temperature of the heat source. Further improvements can be gained by optimized design of the

turbines. A cascaded cycle architecture minimizes the exergy losses in the heater resulting in large heat exchanger surfaces. The optimization of the backpressures may compensate the efficiency drop at high cooling water inlet conditions.

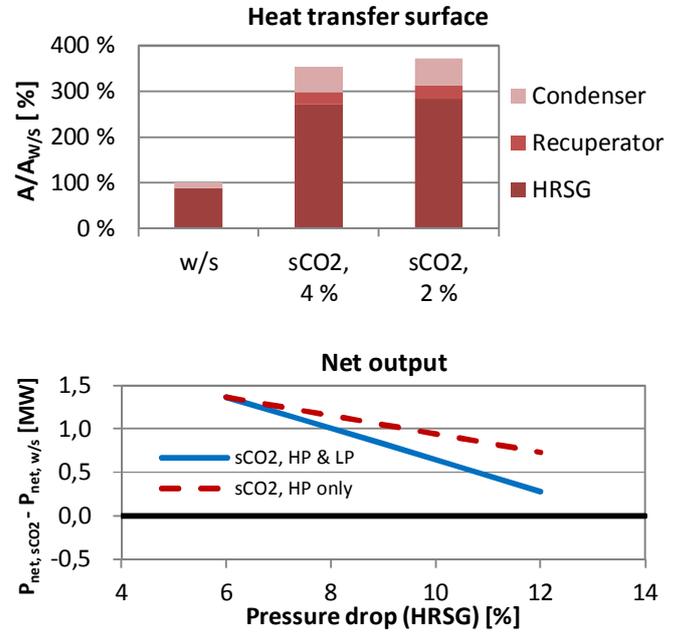


Figure 22: Heat transfer surface and deviation of the power output values depending on the pressure drop of the heat exchangers compared to reference values based on water/steam.

SUMMARY AND CONCLUSION

sCO₂ power cycles have been evaluated for a concentrated solar power plant and two combined cycle power plants and have been compared with reference cycles based on water/steam. A performance benefit can be noted for the CSP application in particular at turbine inlet temperatures above approximately 600 °C. For waste heat applications the potential for sCO₂ power cycles is rising with lower heat source temperatures. In all cases the performance benefits are associated with large heating surfaces, which make the development of high efficient and cost effective heat exchangers necessary. Initial turbine design calculations result in significant lower stage numbers compared to the reference turbines in the Rankine cycles. Much smaller turbine exhaust volume flows reduce the turbine sizes at the cold end. However, increased wall thicknesses in casings, valves and piping on the higher pressure side have to be considered. Furthermore optimized turbine designs and plant layouts must consider the large impact of the pressure losses on the cycle efficiency. In a subsequent step the economic assessment and optimization of the cycles as well as the development of high efficient and cost balanced components needs to be addressed.

OUTLOOK

An industrial-scientific partner consortium has joined forces with regard to the technological development of the sCO₂ cycles. In addition to a pilot plant for method development and basic research, the development of a prototype for a waste heat recovery application is planned in order to contribute to further improvement of the technologies. In a preceding step, the thermo-economic evaluation of the applications under consideration will be carried out.

NOMENCLATURE

<i>A</i>	Heat transfer surface [m ²]
<i>CSP</i>	Concentrated Solar Power
<i>h</i>	Mass-specific Enthalpy [kJ/kg]
<i>HP</i>	High Pressure
<i>HTR</i>	High Temperature Recuperator
<i>HRSG</i>	Heat Recovery Steam Generator
<i>k</i>	Heat transition coefficient
<i>LP</i>	Low Pressure
<i>LTR</i>	Low Temperature Recuperator
<i>p</i>	Pressure [bar]
<i>P</i>	Power output [MW]
<i>ref</i>	Reference
<i>sCO₂</i>	Supercritical Carbon Dioxide
<i>T</i>	Temperature [°C or K]
<i>T_m</i>	Mean Temperature [°C or K]
<i>TTD</i>	Terminal Temperature Difference [K]
<i>\dot{V}</i>	Volume flow [m ³ /s]
<i>w/s</i>	Water/Steam
<i>Δ</i>	Difference
<i>η</i>	Efficiency
<i>η_{th}</i>	Thermal Efficiency

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