SUPERCRITICAL CO_2-BASED WASTE HEAT RECOVERY SYSTEMS FOR COMBINED CYCLE POWER PLANTS

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ABSTRACT

The development of sCO_2 is pursued for various power applications because early and more recent studies have documented possible advantages in terms of efficiency and turbine compactness. This study on medium and large combined cycle configurations is focused on the preliminary assessment several solutions based on specific configurations of bottoming units adopting sCO_2 as working fluid. The results demonstrate that the optimal bottoming unit layout is that of the so-called *dual rail cycle* configuration, where the pinch point problem in the low-temperature recuperator is overcome by splitting the CO_2 flow leaving the compressor into two streams. For the heavy-duty gas turbine case, the dual rail power cycle allows for a net conversion efficiency similar to that of a state-of-the-art CCGT power plant with a three pressure-level and reheat steam Rankine bottoming unit. For medium power capacity gas turbines, the dual rail power cycle, instead, overperforms the conventional steam bottoming unit, which generally consists of a two-pressure level steam cycle. This result suggests that high-efficiency distributed power generation may be the target application for a first deployment of sCO_2 power technology.

1. INTRODUCTION

State-of-the-art gas turbine combined cycle (GTCC) power plants achieve net conversion efficiencies as high as ~63% and the mid-term goal of major equipment manufacturers is to bring the efficiency of large-capacity GTCC power plants beyond 65%. One of the technological challenges which should be overcome is the limited potential for improvement of the bottoming unit (BU) performance: the matching between the temperature profiles of steam and exhaust gases in the heat recovery steam generator (HRSG) can be improved only with costly solutions due to the so called-pinch point problem. Examples of these solutions entail increasing of the number of pressure levels of the HRSG or making the high-pressure section of the HRSG supercritical. In recent times, the ongoing shift from centralized to distributed power generation in many parts of the world caused an increase in R&D related to CCGT power plants featuring high operational flexibility and lower capital expenditure.

The adoption of a different thermodynamic cycle for the bottoming unit is a radical but possibly rewarding approach to pursuing the targeted conversion efficiency of large power plants and also to addressing the new requirements of the energy market. This study investigates the performance of heat recovery units based on innovative thermodynamic cycle configurations employing supercritical CO_2 (sCO_2) as the working fluid. In particular, the examined cases are related to the heat recovery from heavy duty gas turbines (SGT6-8000H, 298 MW), industrial gas turbines (2 x SGT-800, 108 MW) and aeroderivative gas turbines (SGT-A65, 52 MW). Among the possible configurations, the sCO_2 BU is also combined with an organic Rankine cycle (ORC) turbogenerator in an attempt to maximize the maximum thermal energy recovery from the GT exhaust gases. The performance of selected plant configurations is assessed and optimized at the nominal operating point of the GT by means of a steady-state thermodynamic and exergy analysis software tool.

2. THE BENCHMARK PLANT

The bottoming plant assumed as benchmark for the solutions evaluated in this work is a traditional steam Rankine cycle power unit. For the case of heat recovery from the heavy-duty gas turbine SGT6-8000H (case study #1), the steam plant features three pressure levels and reheat. It represents the state of the art for modern GTCC power stations for an overall plant semi-net efficiency $\eta_{\text{semi-net}}$[1] of 61.8%. For the case studies with the industrial gas turbines (2 x SGT-800 – case study #2) or the aeroderivative SGT-A65 (case study #3) as topping unit, the benchmark consists of a two-pressure level steam cycle enabling a combined plant $\eta_{\text{semi-net}}$ at nominal conditions of 58.3% and 55%, respectively. The lower overall conversion efficiency with respect to that of case study #1 is due to the lower gas turbines efficiency, the reduced steam maximum temperature, and, in the case of the aeroderivative GTCC plant, the use of an air-cooled condenser in place of a water-cooled condenser. The main characteristics of the three combined cycle power plants taken as benchmarks are summarized in Table 1.

Table 1: Main characteristics of the benchmark combined cycle power plants

Topping Unit	$\eta_{\text{semi-net}}$	$\eta_{\text{GT,gross}}$	Steam T_{max}	Steam p_{max}	Condenser Type	p_{cond}
SGT6-8000H	61.8%	40.4%	600 ºC	170 bar	Water-cooled	0.05 bar
2 x SGT-800	58.3%	39.9%	550 ºC	80 bar	Water-cooled	0.045 bar
SGT-A65	55%	42.4%	400 ºC	40 bar	Air-cooled	0.074 bar

[1] Semi-net conversion efficiency is defined as the ratio of the net power of the plant, excluding auxiliaries power consumption and generator losses, over the fuel chemical energy input per unit time, Q_{fuel}. For the application at hand, it reads: $\frac{W_{\text{GT,gross}}+W_{\text{BU turbs}}-W_{\text{BU pumps}}-W_{\text{cool. sys.}}}{Q_{\text{fuel}}}$, where the terms $W_{BU\ turbs}$, $W_{BU\ pumps}$ are shaft power.

The thermodynamic efficiency of the steam Rankine cycle of state-of-the-art CCGT plants is remarkably high and difficult to improve: consider the efficiency of an ideal prime mover converting the thermal energy of a gas turbine exhaust into work, namely a Lorentz cycle machine whose process is qualitatively represented in Figure 1. For an exhaust gas temperature of 633ºC, i.e., the discharge temperature of SGT6-8000H, and a sink temperature of 15 ºC, the conversion efficiency of a GTCC plant with such a bottoming unit reaches 68.6 %, see Table 2. However, if we account in the calculation for the performance of the cycle components and the fact that the maximum cycle temperature is bounded to 600ºC due to the current limitations of high-temperature corrosion-resistant materials, the GTCC efficiency reduces to 63.3 %, only 1.5 % percentage points higher than that achieved with the three pressure-level steam cycle considered as benchmark in case study 1.

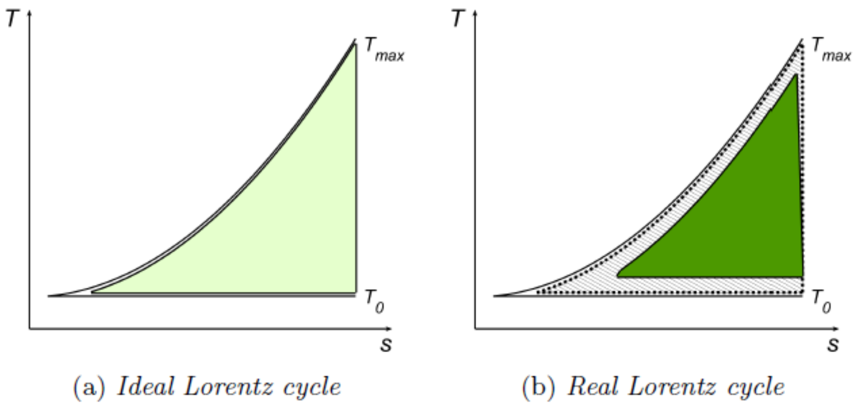

Figure 1: Qualitative temperature-entropy diagram of a) ideal Lorentz cycle, and b) Lorentz cycle if the equipment performance is accounted.

Table 2: Theoretical efficiency of a GTCC plant with a Lorentz-cycle machine as bottoming unit.

$T_{\text{exh.gas}}$	$T_{\text{max cycle}}$	T_{stack}	T_{cond}	η_{turb}	η_{GT}	η_{Lorentz}	η_{GTCC}
633ºC	633ºC	15ºC	15ºC	100 %	40.4%	46.6%	**68.6%**
633ºC	600ºC	80ºC	35ºC	90 %	40.4%	42.2%	**63.3%**
633ºC	600ºC	45ºC	35ºC	90 %	40.4%	40.0%	**63.5%**
675ºC	600ºC	45ºC	35ºC	90 %	40.4%	40.0%	**65.1%**
675ºC	650ºC	45ºC	35ºC	90 %	40.4%	41.6%	**66.1%**

3. CYCLE CONFIGURATIONS

As shown in Table 2, a significant improvement in GTCC performance can be pursued only with next-generation GT technology which may exhibit a higher turbine discharge temperature, possibly followed by a rise in the maximum temperature of the bottoming cycle. A further increase in the steam live temperature is, however, hindered by issues related to material corrosion, which severely augments with steam temperature, as well as the cost and the current technical limitations of superalloys (e.g., low thermal conductivity). Although high-temperature materials suitable for CO_2 are still in an early development phase, sCO_2 power cycles can become a valid alternative to steam technology for future CCGT power stations, given their increasingly higher thermodynamic performance when the temperature level of the thermal source is raised (Angelino, 1968) and the arguably lower corrosiveness of

CO_2 compared to water. Moreover, once it has become a mature technology, sCO_2 power cycle gen-sets may offer a better trade-off between performance and CAPEX and, possibly, greater operating flexibility, thanks to the high power density of the thermodynamic cycle.

Nevertheless, so far only few studies to assess sCO_2 technology for combined cycle power plants have been published (see, e.g., Wright et al. 2016; Sharma et al. 2017, Kim et al., 2017). While it is well known that the recompression cycle first proposed by Angelino in 1968 is the best sCO_2 cycle configuration for the conversion of primary energy sources, there is, instead, limited consensus on which cycle layout is the most suited for heat recovery from gas turbines. Therefore, five sCO_2-based BU configurations, whose arrangement is schematically represented in Figure 2, are here proposed and have been assessed.

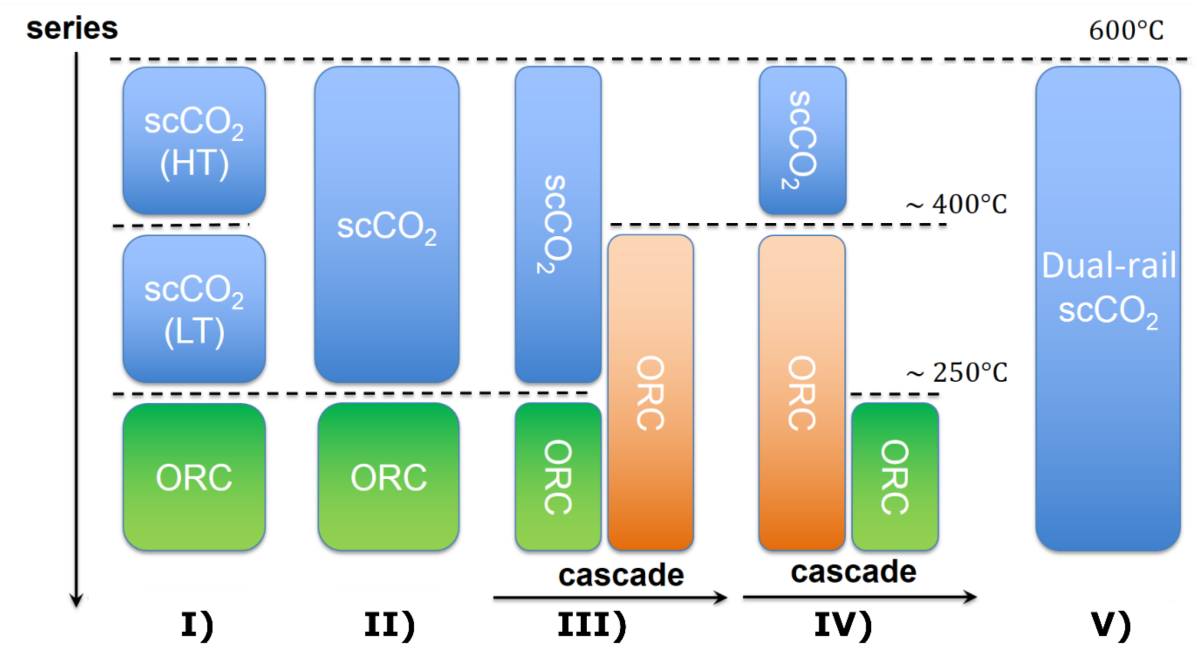

Figure 2: Schematic representation of the 5 $scCO_2$ -based cycle layouts assessed in the study.

In configuration **I** two sCO_2 recompression cycles are combined with an ORC in series. The idea is to maximize the BU efficiency and the thermal energy harvesting from the exhaust gases by exploiting the high efficiency of the sCO_2 recompression cycle and by compensating for its low heat recovery factor using different units in series which progressively cool down the exhaust gases. When the exhaust gas temperature reaches about 250 ºC, heat recovery is accomplished by means of a simple-cycle ORC unit, since the high degree of internal recuperation of the sCO_2 recompression cycle would limit the thermal energy recovery from the thermal source.

The BU configuration **II** is similar to the first one. The main difference is that the two sCO_2 recompression cycle units are now integrated into one single power block, given that the low-temperature loop of the two separate units operate at very similar temperature levels. The benefit is a simplification of the plant layout, see Figure 3a, at the cost of a reduction in the number of degrees of freedom of the system which can be tuned to maximize the BU thermodynamic performance.

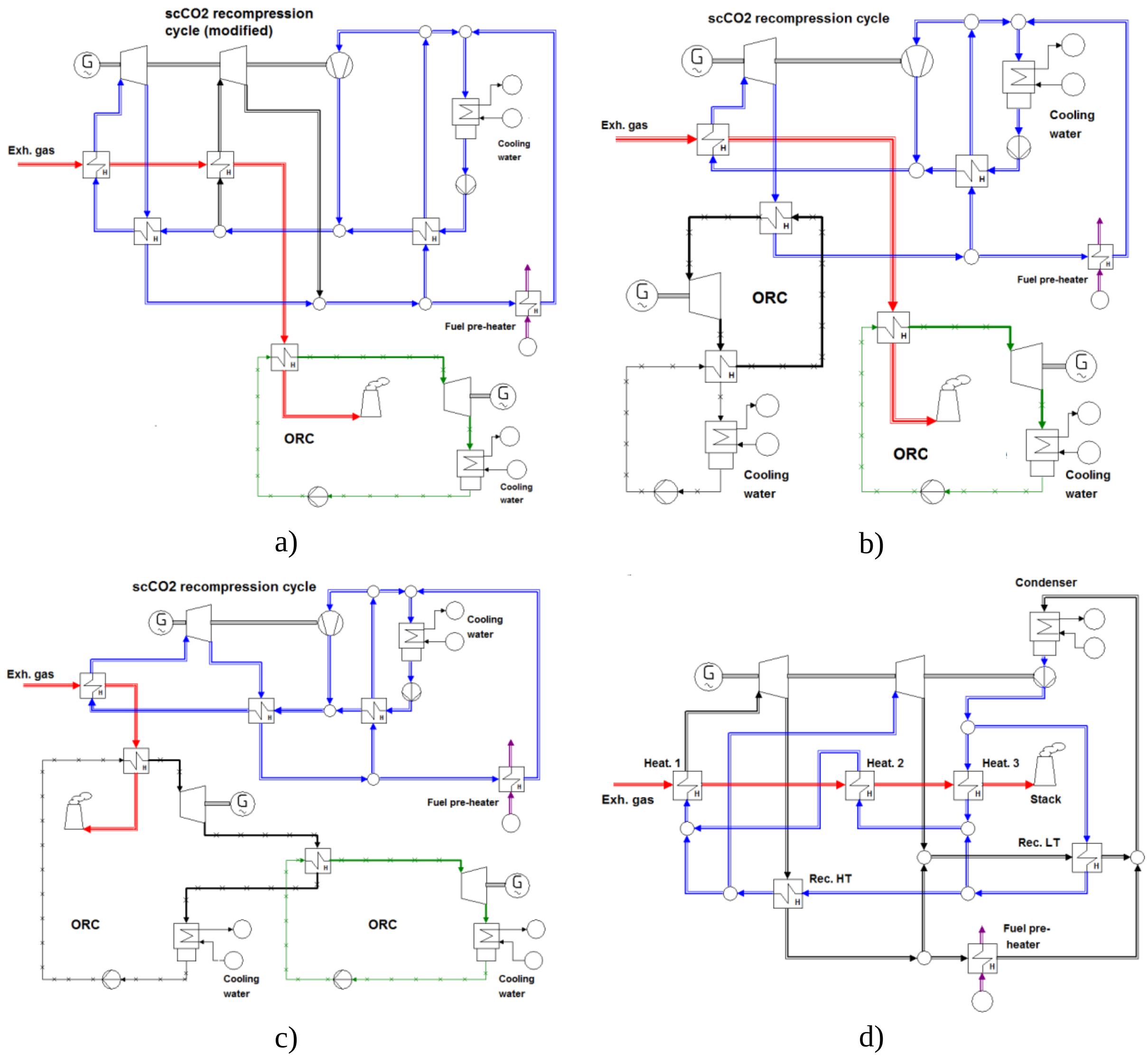

Figure 3: From top left clockwise, process flow diagram of the bottoming unit configurations analyzed in the study and labeled in Figure 2 from *II* to *V*.

Configuration **III** and **IV** are based on one single sCO_2 recompression cycle unit which is used for heat recovery at high temperature ($T_{\text{exh gases}} = 400$ °C), while two ORC turbogenerators are adopted to convert the remaining thermal energy of the gas turbine exhausts. More in detail, in configuration **III**, the high-temperature recuperator of the sCO_2 recompression cycle circuit is replaced by the primary heat exchanger of a supercritical ORC unit, while the other ORC power block is fed with the exhaust gases leaving the sCO_2 unit at 250 ºC (Figure 3b), as in the previous plant configurations. The rationale behind this choice is to pursue a further simplification of the sCO_2 cycle layout and to reduce internal recuperation in the sCO_2 cycle and the size of the associated heat transfer equipment. Internal recuperation is, instead, performed at a lower temperature level in the supercritical ORC. The same purpose is pursued with configuration **IV**, where the two ORC turbogenerators are arranged in cascade, see Figure 3c. A high temperature ORC loop is fed with exhaust gases at 400 ºC, previously cooled in a the sCO_2 recompression cycle unit. The thermal energy of the superheated vapor discharged by the turbine of this ORC loop is, then, used as thermal input for a low-temperature ORC genset.

A fifth bottoming cycle configuration, the so-called dual rail cycle (Wright, S., 2016), can be considered a variation of plant configuration **II**. The main difference lies in the solution adopted to reduce the pinch-point limitation in the low-temperature recuperator. In a recompression cycle, the CO_2 stream leaving the turbine is split into two streams at the low-temperature recuperator outlet: one stream is sent to the cooler, the other one is recompressed and sent to the high-temperature recuperator of the cycle. The optimum flow split fraction is that whereby the heat capacity rate of the two streams of the low-temperature recuperator becomes similar. The same effect is obtained in the dual-rail cycle by reducing the mass flow rate of the cold stream of the same recuperator. A portion of the flow leaving the main compressor is sent to a preheater (indicated as Heater 3 in Figure 3d), where it is heated up by the GT exhaust gases and it is then mixed with the CO_2 stream coming from the low-temperature recuperator before Heater 2, see Figure 3d. The advantage is that the GT exhaust gases can be cooled down to low temperature without the need of an additional ORC unit, as in configuration **I** and **II**.

4. MODELING AND SIMULATION

The models of the power plant configurations considered in this study have been implemented in a flowsheeting program for energy systems analysis and optimization initially developed at TU Delft by van der Stelt, Woudstra, and Colonna (2013). The graphical user interface of the software provides, for example, the process flow diagrams shown in Section 3. Fluid thermodynamics properties are computed with an in-house library which allows to estimate all primary and secondary properties with a variety of models for pure fluids and mixtures, see Colonna, van der Stelt, and Guardone (2019). Properties of carbon dioxide are calculated according to the multiparameter equation of state model of Span and Wagner (1996). Constrained optimization algorithms of a multipurpose numerical computing environment (Mathworks, 2016) have been coupled with the flowsheeting program in order to calculate the maximum performance and corresponding operating parameters for a given cycle configuration. The coupling is via input/output text files. As for the optimization, first the variables' region of the global optimal solution is identified by means of a procedure based on a genetic algorithm (*ga* function). The optimal solution is then calculated with a gradient-based optimizer (*fmincon* function).

Given the power capacity and the relative high temperature of the application, suitable working fluids candidates for the ORC turbogenerators are limited to alkanes, cycloalkanes, and a few refrigerants which exhibit excellent thermal stability. The organic compounds considered in this study are the alkane Pentane, the cycloalkanes Cyclobutane, Cyclopentane, and Cyclohexane, and the refrigerants R245fa and R125. The working fluid selection is here only driven by thermodynamic considerations and is carried out by repeating the thermodynamic cycle analysis and optimization of the BU configuration under consideration for each of the abovementioned organic compounds.

Finally, several assumptions regarding the performance of the plant components have to be considered in order to perform the thermodynamic analysis of the system. Cycle parameters are listed in Table 3, and include: the pressure drop and the minimum temperature difference in the heat exchangers, the turbomachines efficiencies, and the cooling water temperature in the sCO_2 cycle cooler or the ORC condenser(s).

Table 3: Model assumptions for the thermodynamic cycle simulations

$T_{\text{cool.w,in}}$	ºC	19	$T_{\text{max BU}}$	ºC	600
$T_{\text{cool.w,out}}$	ºC	27	$\eta_{\text{turb sCO}_2}$	%	92
$\Delta T_{\text{pp HEX}}$	ºC	8	$\eta_{\text{compr sCO}_2}$	%	85
$\Delta T_{\text{pp cooler/cond.}}$	ºC	4	$\eta_{\text{turb ORC}}$	%	90
$\Delta p/p_{\text{in HEX}}$	%	2	$\eta_{\text{pump ORC}}$	%	80

5. RESULTS

All the bottoming cycle configurations described in Section 3 have been assessed for case study #1 (heavy-duty gas turbine, SGT6-8000H). Only the cycle configuration yielding the best performance according to the simulations have been considered for the smaller GT's, i.e., case study #2 (2 x SGT-800) and #3 (aeroderivative SGT-A65).

The key-performance parameters estimated for the candidate BU configurations in the case of heat recovery from the heavy-duty gas turbine SGT6-8000H are reported in Table 4. The conversion efficiency differs, at most, by about 1 percentage point between the proposed power plant concepts. Its value remains lower than that of the selected benchmark plant, except for the case of the dual-rail sCO_2 cycle, which allows for a net power output almost identical to that of the triple-pressure steam Rankine cycle taken as reference. This occurs in spite of the higher fraction of thermal energy recovered from the exhaust gases. The value of the recovery factor χ, which is defined as the ratio between the actual thermal energy recovered in the bottoming unit and the maximum amount recoverable when the exhaust gases are cooled down to the ambient temperature, is similar for all the alternative bottoming cycle configurations. It ranges between about 93% and 94%, against 89.4% estimated for the benchmark plant.

Table 4. Results of the simulation and optimization study for the case of heat recovery from SGT6-8000H

		Benchmark	Conf. **I**	Conf. **II**	Conf. **III**	Conf. **IV**	Conf. **V**
η_{GTCC}	%	61.8	60.9	60.6	60.5	60.8	61.8
W_{BC}	MW	157.3	151.1	148.5	148.1	150.4	157.3
Q_{rec}	MW	396.2	416.6	410.6	416.3	412	400.4
χ_{rec}	-	89.4	94	92.7	94	93	90.4
$p_{\text{max,cycle}}$	bar	169.6	330.7 / 214.5	359	343.1	351.7	335
$p_{\text{max,ORC}}$	bar	-	60.8	59.8	245.8 / 65.4	202.4 / 39.4	-
ORC fluid(s)		-	Pentane	Pentane	R125 / Pentane	R125 / Pentane	-

To gain more insight regarding the results of Table 4, an exergy analysis was carried out. This reveals that the performance of the BU configurations adopting a sCO_2 cycle are, in general, penalized by higher turbomachine losses with respect to the benchmark steam cycle, see Table 5. The efficiency penalty associated with the CO_2 compressors is, indeed, an order of magnitude larger than that corresponding to the pumps of the steam Rankine cycle, although CO_2 compression is performed in a thermodynamic region close to the critical point. Among the proposed BU configurations, the dual-rail sCO_2 cycle features the lower turbomachinery losses, since the recompression of a portion of the CO_2 stream leaving the low temperature recuperator is avoided by using a preheater.

Table 5. Results of the exergy analysis

	Benchmark	Conf. **I**	Conf. **II**	Conf. **III**	Conf. **IV**	Conf. **V**
$\eta_{II,GTCC}$ %	72.3	69.2	68.1	67.9	68.9	72.3
$\Delta\eta$ Heaters/HRSG	7.5	3.2	4.4	3.7	3.4	3.8
$\Delta\eta$ Recuperators	-	5.5	3.9	6.3	5.1	3.5
$\Delta\eta$ Heat transfer	7.5	8.7	8.3	10.0	8.5	7.3
$\Delta\eta$ Condenser	7.1	8.7	9.8	9.6	9.8	8.6
$\Delta\eta$ Stack	5.0	3.6	3.9	3.6	3.8	3.9
$\Delta\eta$ Heat rejection	12.1	12.3	13.7	13.2	13.6	12.5
$\Delta\eta$ Turbines	5.0	4.9	5.0	4.5	4.8	3.7
$\Delta\eta$ Compr./Pumps	0.2	3.7	3.7	3.4	3.3	2.6
$\Delta\eta$ Turbomachines	5.2	8.6	8.7	7.9	8.1	6.3
Others	2.9	1.2	1.2	1.0	1.0	1.6
Total	100	100	100	100	100	100

In general, the adoption of a supercritical fluid allows for a better thermal coupling between the exhaust gases and the bottoming unit, as shown, for instance, in Figure 4. It results that the overall exergy losses in the heaters of the sCO_2 cycles are, approximately, half of those occurring in the triple-pressure HRSG. However, this benefit is more than compensated by the losses associated to heat transfer in the sCO_2 cycle recuperators.

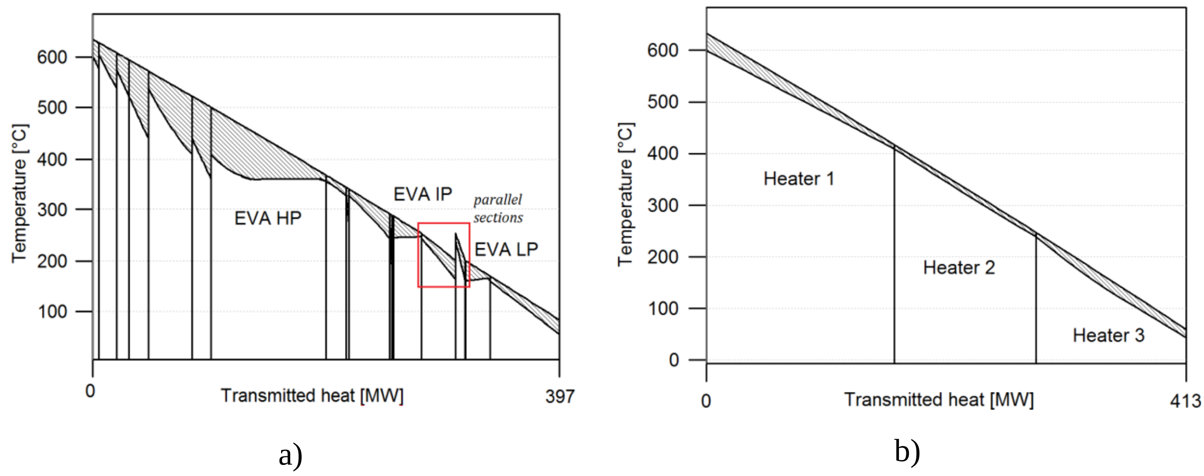

Figure 4. Temperature - thermal power diagram. a) Triple-pressure and reheat steam Rankine cycle; b) Dual-rail sCO_2 cycle.

In summary, the proposed BU configurations are penalized by the additional heat transfer needed to accomplish internal recuperation or to provide thermal power to the cascaded ORC gensets, as for configuration **III** and **IV**. The dual-rail cycle proves to be the only one which can compete in terms of conversion efficiency with a state-of-the-art steam Rankine cycle, and it can be a valid alternative to the latter given its layout simplicity and ease of adaptation to thermal sources of different temperature levels. This is shown in Figure 5, which compares the GTCC conversion efficiency achievable with a dual-rail sCO_2 cycle as bottoming unit against that attained by the reference GTCC power plants defined in Section 2 for the three case studies under consideration. The dual-rail cycle allows for a higher conversion efficiency when the topping unit consists of two industrial gas turbines SGT-800 (case study #2) or in the aeroderivative SGT-A65 (case study #3). The gain is 1.4 percentage points of efficiency for the power plant with the lowest capacity, and half percentage point for case study #2. The difference in the dual-rail sCO_2 cycle layout with respect to the case of heat recovery from the SGT-800H lies only in the inlet temperature and pressure of the two turbines. This scalability and flexibility of the dual-rail cycle with respect to the thermal source temperature can be very beneficial for small scale applications where modularity may provide a mean to minimize the CAPEX of the power plant without penalizing its performance.

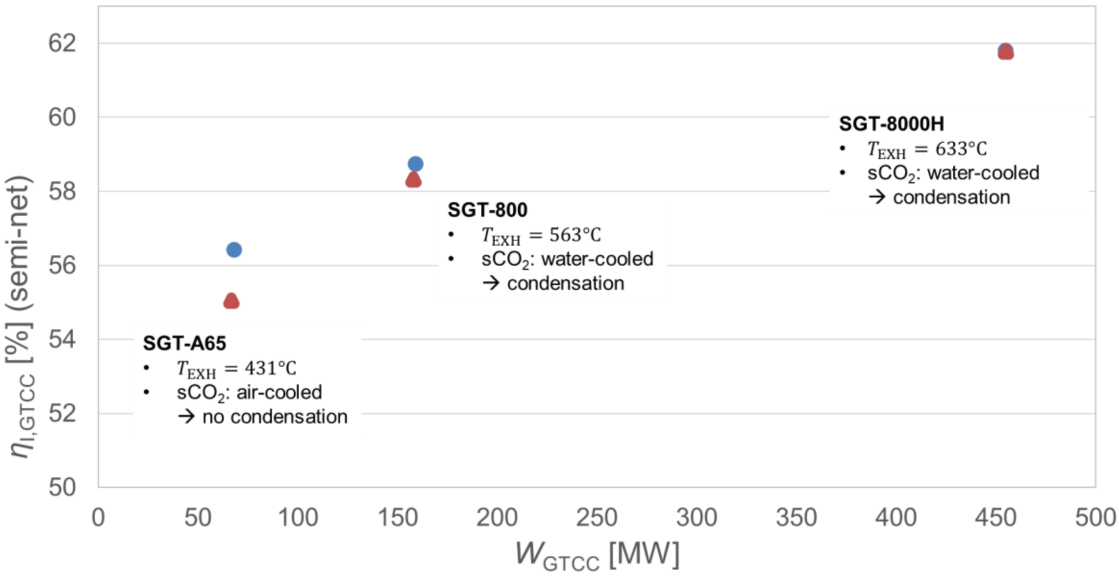

Figure 5: Conversion efficiency of CCGT power plants as a function of GT power capacity and technology selected for the bottoming unit: sCO_2 dual rail cycle (blue circular marker) versus steam Rankine cycle (red triangular marker).

6. CONCLUSIONS

A comprehensive study on combined cycle power plant configurations featuring exemplary Siemens gas turbines and non-conventional bottoming cycles has been carried out. The bottoming units are based on sCO_2 cycle and ORC configurations. The best configurations have been compared to benchmark configurations based on state-of-the-art steam bottoming cycles.

For all the considered power capacities, the highest energy conversion efficiency has been obtained with the dual-rail sCO_2 cycle configuration (Figure 3d, Figure 5 for the efficiency) because of its superior thermal coupling with the gas turbine exhaust. The efficiency advantage is sizable for the considered industrial gas turbines and negligible for the heavy-duty gas turbine. As known, the optimal thermodynamic performance can be achieved with the highest level of internal recuperation. In turn, the cost of the additional heat transfer surface does not scale linearly with the increase in efficiency, thus CAPEX issues might prevent the realization of the concept and only a study tackling the difficult issue of predicting the industrial cost of the equipment in a future scenario would allow to evaluate the actual benefit of adopting the dual-rail sCO_2 cycle configuration. In general terms, the turbine of the sCO_2 cycle is expected to be considerably less expensive than the corresponding steam turbine. However, the cost of the recuperators of the sCO_2 cycle might be very large and it is difficult to predict as they are not industrialized yet. Arguably, one benefit of the sCO_2 cycle configuration if compared to the traditional steam solution is that the same conversion efficiency can be achieved by optimizing the cost of the primary heat exchanger together with that of the recuperator. The material of the recuperator is less expensive because it is operated at lower temperature, therefore the heat transfer surface of the primary heat exchanger can be reduced and that of the recuperator increased with a possible cost advantage. Other possible benefits include smaller footprint, layout simplicity, scalability and adaptability to thermal sources at different temperature levels.

Medium-size combined cycle power plants adopting the dual rail sCO_2 cycle configuration are already pursued for commercial purposes and compete with solution based on traditional steam bottoming units and ORC waste heat recovery. For large capacity applications, the adoption of novel concepts for the bottoming cycle does not yield advantages, if current gas turbine technology is considered. However, previous studies, see, e.g. (Angelino, 1968), indicate that the performance of the sCO_2 cycle sharply increases with increasing temperature levels. A definite advantage might emerge if it will be proven that CO_2 is less corrosive than water at these high temperatures.

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