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A GLOBAL APPROACH TO ASSESSING THE POTENTIAL OF COMBINED CYCLES USING SUPERCRITICAL TECHNOLOGY

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ABSTRACT

The utilisation of supercritical bottoming cycles in gas turbine based power plants is not new at all. A number of works have already been published discussing the real potential of this configuration as compared to the standard technology using subcritical steam turbines. Nevertheless, the interest of this paper relies on the global approach to cycle optimisation and to the assessment of the technical implications on the topping/bottoming systems. A further interest is the comparison of different layouts in the bottoming systems. In particular, two bottoming system options are considered in this paper. The first one makes use of supercritical steam turbines with single steam pressure and single reheat. The second one considers closed Brayton cycles using supercritical carbon dioxide as the working fluid.

The results of the paper show that there are some operating scenarios favouring the utilisation of supercritical bottoming cycles, mainly with steam and to a much lesser extent with carbon dioxide. The differences in performance are not dramatic but the simplicity and operability of these configurations have the potential to make a case for supercritical combined cycles.

INTRODUCTION

Current status of combined cycle power plants

The golden age of combined cycle power plants in Europe seems to have passed already due to the high prices of natural gas (in absolute figures and relative to coal) and the increasing concerns about the environment (carbon emissions) and the security of supply. This is reflected in the fact that, as of December 2015, there were no new orders for engines larger than 120 MWe in this region, in contrast to Middle East (39), Far East (23) and North (27) and Central (17) America [1]. The increasing share of renewables and the recent claims that a 100% renewable world is possible [2] are putting additional pressure on gas turbines, to the extent that "*recent estimates suggest that 51 GWe of the EU's generation capacity is currently mothballed and that 110 GWe of installed combined cycle capacity – 60% of Europe's total gas-fired capacity– is not recovering fixed costs and may face closure within the next three years*" [3]. In this scenario, gas turbine operators are seeing ancillary markets as the only choice to obtain revenues from their assets, but these markets are limited in size and hence they cannot accommodate all the existing plants. Moreover, low coal prices are favouring that coal power plants also compete in these markets, further limiting the opportunities for gas turbines.

Potential of supercritical bottoming cycles

The aforedescribed scenario depicts a situation where efficiency and operating costs (OpEx), in particular fuel costs, have been superseded by flexibility and capital costs (CapEx) as the main technical and economic drivers of combined cycle design. For standard subcritical steam technology this means:

- Dual pressure or even single pressure bottoming steam turbines which might reduce combined cycle efficiency from 60%+ to 50-55% [4].
- Triple pressure cycles with once-through heat recovery steam generators in the high pressure circuit. This enables similar combine cycle efficiencies but at the expenses of a high installation cost [5].

As a complement to these two options, this work explores the potential of supercritical bottoming cycles using simpler layouts than state-of-the-art combined plants. Two working fluids are considered: water/steam and carbon dioxide. The former is considered a less disruptive solution relying on the vast experience of the nuclear and coal industries with supercritical technologies, whilst the latter has recently captured the interest of the power industry as credited by the continuously growing scientific and industrial community built around it (e.g., the Supercritical CO_2 Power Cycle Symposium held every two years).

OBJECTIVES AND METHODOLOGY

Literature review

The utilisation of supercritical steam turbines in combined cycle applications has already been explored in the past. For instance, Bolland presents an interesting comparison of subcritical and supercritical, dual and triple pressure steam bottoming cycles using both the 1st and 2nd laws of Thermodynamics [6]. The analysis shows a potential reduction (2-4 percentages) of exergy losses in the HRSG when supercritical pressures are considered but no conclusive statements regarding the interest of the technology are made. Gülen also makes use of the 2nd law of Thermodynamics to explore the potential (entitlement) of bottoming supercritical steam cycles in [7], arriving to the conclusion that only modest gains with respect to conventional technologies should be expected. According to the author, there is not much interest in adopting this steam turbine technology in combined cycles.

The previous works are based on the standard integration of multiple pressure steam cycles. Franco considers less conventional layouts including single pressure supercritical cycles with double reheat, and incorporates an optimiser to obtain the highest efficiency [8]. The main conclusions are that single pressure supercritical bottoming systems are of interest for combined cycles in the range from 100 to 150 MWe, in particular if gas turbine exhaust temperatures keep rising as they have in the last decades. Rao and Francuz present an interesting comparison of various technologies that could enable higher efficiencies in combined cycle power plants, supercritical bottoming cycles included [9]. Nevertheless, they draw the opposite conclusion to Franco's, stating that supercritical steam turbines are interesting in fire boilers only.

Finally, there are some interesting works focusing on the simulation of once through boilers under dynamic operating conditions, [10] and [11].

Objectives and Methodology

As illustrated in the previous section, there is not consensus regarding the interest of supercritical steam turbines in combined cycle applications. Most authors agree that there are visible challenges, in particular regarding the different control strategies with respect to the standard technology, but there is disparity in the potential efficiency and flexibility gains.

This work aims to revisit this discussion with two main innovative features with respect to past works:

- Supercritical cycles operating on carbon dioxide are also considered, in addition to bottoming supercritical steam cycles.
- Rather than focusing on optimising the design parameters of the bottoming cycle, this work focuses on the specifications of the gas turbine (mainly pressure ratio).

This last feature is actually of great interest for the authors. Most of the existing works regarding gas and steam combined cycles (cited in the previous section) and gas and carbon dioxide cycles, for instance [12], make use of a reference factory engine and then explore the design space of the bottoming system to assess the highest efficiency attainable.

The approach in this work is just the opposite. A state of the art bottoming system is selected and then the gas turbine cycle that would yield highest combined cycle efficiency is evaluated.

This might seem like a contradictory approach, in particular because a combined cycle is actually designed as a master-slave system. The rationale though is that the currently available portfolio of gas turbines has been designed to achieve highest efficiency when coupled to a bottoming triple-pressure subcritical steam cycle [4]. If the latter cycle changes, then the gas turbine might need to change its design pressure ratio to get the most out of the combined power plant.

In order to perform this analysis, a combined modelling tool has been developed. Gas and steam cycles are modelled in Thermoflex whilst supercritical carbon dioxide cycles are simulated with an in-house tool implemented in Matlab. This last tool follows a similar approach to Thermoflex, making use of one-dimensional performance models for turbomachinery and heat exchangers and accounting for the non-ideal behaviour of the working fluid. Given the similar level of complexity of all the models used, the results are trusted to be comparable.

REFERENCE CASES – DESIGN SPECIFICATIONS

Gas turbine

A state of the art gas turbine is considered with specifications very similar to those of the General Electric 6FA engine. These are summarised in Table 1.

Parameter	*Value*
Inlet loss [%]	1.5%
Pressure ratio [-]	15.4
Turbine inlet temperature [ºC]	1325ºC
Compressor/turbine isentropic efficiency	85/90
Combustor pressure loss [%]	4
Gross output [MWe]	75.6
Gross efficiency [$\%_{LHV}$]	35.0%

Table 1. Gas turbine specifications.

Pressure ratio is then varied during the analysis in order to explore the impact on plant efficiency whilst the remaining design specifications remain constant.

Steam turbine

For the supercritical steam cycles, a single pressure and reheat layout has been considered, Figure 1, which is similar to that considered by the authors in [13]. Attention is drawn towards the deaerator loop (DA) within the waste heat recovery boiler (WHR). This component, which is mandatory in order to eliminate the non-condensable gases from the water/steam loop, is not needed in a sCO_2 system since this always works above atmospheric pressure.

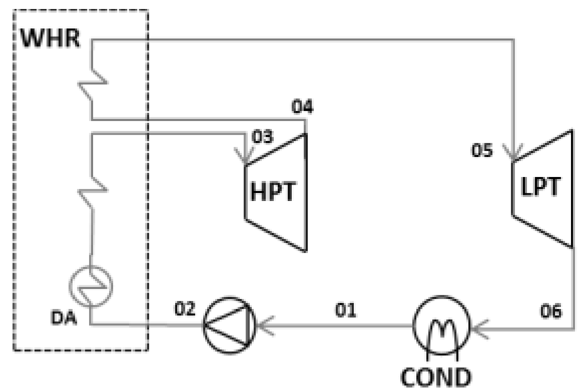

Figure 1. Schematic layout of the reference supercritical Rankine cycle.

The main design specifications are presented in Table 2. It is worth noting that the condenser pressure has been set to a value corresponding to ca. 32ºC saturation temperature, as this is the minimum temperature in the supercritical CO_2 cycles presented later. Even if this temperature might seem a bit too low, it is still well within the industrial practice.

Parameter	*Value*
Turbine inlet pressure p_{03} [bar]	250
Turbine inlet temperature T_{03} [ºC]	560
Reheat pressure p_{04} [bar]	54
Reheat temperature T_{04} [ºC]	560
Condenser pressure p_{05} [bar]	0.050
Heat recovery unit pinch point [ºC]	16.6
Pump/turbine isentropic efficiency	78/91

Table 2. Specifications of the supercritical Rankine cycle.

Carbon dioxide

Amongst the large number of possible carbon dioxide cycles to choose from [14], three are considered most appropriate: simple recuperated, precompression and recompression.

The simple recuperated cycle is presented in Figure 2. It comprises a compressor with inlet conditions very close to the critical point of CO_2, gas/gas recuperator (high pressure side), waste heat recovery unit, expander, (low pressure side of the) recuperator and water/gas cooler.

The main specifications of the cycle are summarised in Table 3. It is important to note that no pressure losses across the heat exchangers have been considered, which could be considered an oversimplification of the problem. Nevertheless, the main objective of this work is to assess the potential of supercritical bottoming cycles and the need/convenience to change the design of the topping gas turbine to fully exploit it. In this context, neglecting pressure losses is not expected to alter the conclusions (just the quantitative results).

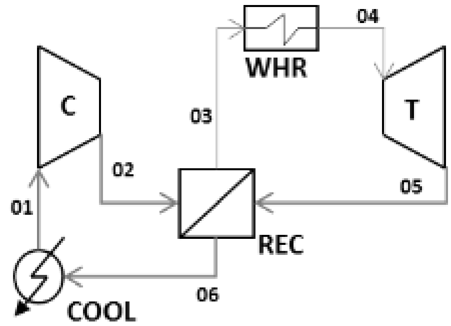

Figure 2. Schematic layout of the simple recuperated cycle.

Parameter	*Value*
Compressor inlet pressure p_{01} [bar]	73.5
Compressor inlet temperature T_{01} [ºC]	32
Compressor outlet pressure p_{02} [bar]	250
Recuperator effectiveness [%]	95
Heat recovery unit pinch point [ºC]	16.6
Turbine inlet pressure p_{04} [bar]	250
Turbine inlet temperature T_{04} [ºC]	560
Compressor/turbine isentropic efficiency	89/93
Pressure drops across cycle [%]	0

Table 3. Specifications of the simple recuperated cycle.

The second carbon dioxide cycle to consider is the precompression cycle, Figure 3. This cycle incorporates a low pressure compressor (C1) to enable a higher expansion ratio across the turbine (p_{05}/p_{06}) whilst still maintaining the main compressor (C2) working close to the critical temperature of carbon dioxide (T_{01}) and, simultaneously, avoiding very high pressures in the cycle (high p_{05}).

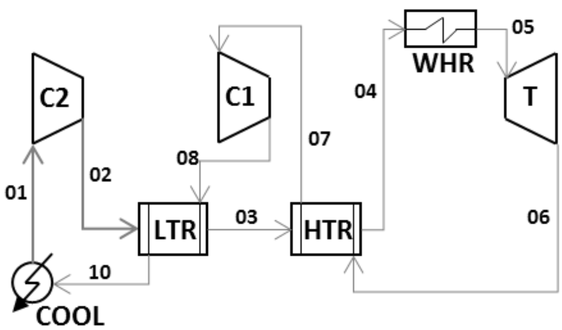

Figure 3. Schematic layout of the precompression cycle.

The low pressure side of the recuperator (which is now subcritical) is split in two, with a low pressure compressor in between them. This compressor raises the pressure of carbon dioxide to 96.1 bar before the flow is cooled down further in the low temperature recuperator and, later, the cooler.

The main features of this cycle follow:

- Higher expansion work without changes in the amount of heat added/rejected (thus higher cycle efficiency) [14].
- Higher compression work, not only due to the higher overall pressure ratio but also because the low pressure compressor operates far from the critical point. This effect is dominant and hence the specific work of this cycle ends up being lower than in the simple recuperated cycle (of the turbine inlet temperature considered).
- For the same turbine inlet temperature, lower temperature at recuperator outlet (T_{04}), which enhances waste heat recovery.

The main design specifications of the reference precompression cycle, presented in Table 4 below, are taken from the work by Kulhánek and Dostál [15].

Parameter	*Value*
LP compressor inlet pressure p_{08} [bar]	96.1
HP compressor inlet pressure p_{01} [bar]	123.5
HP compressor inlet temperature T_{01} [ºC]	32
Compressor outlet pressure p_{02} [bar]	250
LT/HT recuperator effectiveness [%]	80/95
Heat recovery unit pinch point [ºC]	16.7
Turbine inlet pressure p_{05} [bar]	250
Turbine inlet temperature T_{05} [ºC]	560
Compressors/turbine isentropic efficiency	89/93
Pressure drops across cycle [%]	0

Table 4. Specifications of the precompression cycle.

The last CO_2 cycle under consideration is the recompression cycle. This cycle features a compression process divided in parallel streams, Figure 4. The main compressor (C1) works very close to the critical point (p_{01}, T_{01}) whilst the recompressor (C2) works at a higher temperature and the same pressure ratio. The aim of this cycle is to enhance the performance of the low temperature recuperator (LTR) by reducing the mass flow rate on the high pressure side ($\dot{m}_{02}$), [14] [15]. This increases the efficiency of the cycle but at the cost of a lower specific work (with respect to the simple recuperated cycle).

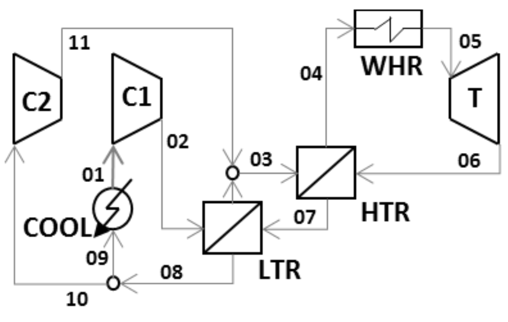

Figure 4. Schematic layout of the recompression cycle.

The main design specifications of the reference recompression cycle, Table 5, are taken from [15].

Parameter	*Value*
Compressor inlet pressure p_{01}, p_{10} [bar]	78.1
Main compressor inlet temperature T_{01} [ºC]	32
Compressor outlet pressure p_{02}, p_{11} [bar]	250
LT/HT recuperator effectiveness [%]	93/95
Heat recovery unit pinch point [ºC]	16.6
Turbine inlet pressure p_{05} [bar]	250
Turbine inlet temperature T_{05} [ºC]	560
Compressors/turbine isentropic efficiency	89/93
Pressure drops across cycle [%]	0

Table 5. Specifications of the recompression cycle.

RESULTS AND DISCUSSION

Reference gas turbine

This first section of the results presents the expected performance of the reference bottoming cycles (design specifications shown in Tables 2 to 5) when coupled to the gas turbine engine in Table 1. The main performance parameters of the resulting combined cycles are shown in Table 6.

Parameter	*H_2O*	*CO_2*		
		SR	*PR*	*RC*
Stack temperature [ºC]	214.7	343.7	408.6	393.3
WHR pinch point [ºC]	16.6	16.6	16.6	16.6
WHR efficiency [%]	67.6	35.44	27.36	29.29
Bottoming cycle efficiency[1] [%]	40.77	40.81	43.88	47.40
Bott. cycle specific work [kJ/kg]	1520	116.6	90.27	106.4
Turbine inlet mass flow [kg/s]	24.77	337.6	230.86	226.5
Turbine inlet flow [m^3/s]	0.323	2.225	1.522	1.493
Turbine exhaust flow [m^3/s]	697.8	5.921	3.250	3.804
Mass flow ratio $\dot{m}_{BC}/\dot{m}_{GT}$ [-]	0.12	1.03	1.10	1.08
Power ratio $\dot{W}_{BC}/\dot{W}_{GT}$ [-]	0.50	0.33	0.28	0.32
Bottoming cycle output[1] [kWe]	37654	25143	20841	24101
Combined cycle output[2] [MWe]	112.7	100.8	96.49	99.75
Combined cycle efficiency[1] [%]	52.63	46.66	44.67	46.18

[1] Thermal efficiency
[2] Gross output at generator terminals

Table 6. Results for the reference combined cycles (figures reported are thermal efficiencies and gross output).

The results obtained are in line with those expected from the literature review. The thermal efficiency of all three carbon dioxide cycles is higher than that of the steam cycle thanks to the known characteristics of supercritical CO_2 cycles, especially the large potential for internal heat recovery. Nevertheless, this higher efficiency cannot compensate for the very bad performance of the waste heat recovery unit that comes about because of the higher CO_2 temperature at the outlet of the high temperature recuperator. This is visible in the much higher stack temperature of all CO_2 configurations.

Further to this observation, it must also be noted that the existence of the deaerator loop (boiler) hardly has any impact

[1] Gross output at generator terminals.

on the performance of the waste heat recovery unit. Indeed, the DA is located halfway through the economiser at a temperature very close to the saturation temperature corresponding to 3.5 bar (which is the operating pressure of this component). It thus brings about a very small temperature drop in the gas turbine exhaust gases. For instance, for the reference case in Table 6, the hot gases are cooled down from 263.6ºC to 261.2ºC only (see Figure 5 in the next section). In accordance, the deaerator cannot be made responsible for the different performance of the WHR in cycles using steam and carbon dioxide.

One of the usual arguments in favour of carbon dioxide is compactness [14] [15]. This is best evidenced by the turbine volumetric flow comparison in Table 6. The much higher specific enthalpy rise of steam across the WHR implies that the mass flow rate of live steam produced by the heat recovery unit is ten times lower than for any of the carbon dioxide cycles, in spite of the much higher fraction of heat recovered by the steam cycle. In terms of volumetric flow, this difference is not compensated for at turbine inlet but it becomes clearly visible at turbine exhaust. Indeed, the much higher expansion ratio of the steam turbine brings about a drastic variation of specific volume as a consequence of which the volumetric flow rate leaving the turbine turns out to be ten times higher for steam. This means ten times larger exhaust area for the same leaving loss, what adds to the larger number of turbine stages required to accommodate the expansion.

As a final remark of the numerical analysis, it is interesting to look at the mass flow ratio $\dot{m}_{BC}/\dot{m}_{GT}$ which again reflects the much higher circulating mass flow rate of the cycles using carbon dioxide.

In summary, it is confirmed that non-condensing supercritical carbon dioxide cycles do not seem able to outperform single-pressure and reheat supercritical steam cycles which, even without using multiple steam pressures, achieve higher efficiencies by almost ten percentage points. At the same time though, these CO_2 cycles can be put forward as an interesting option to retrofit existing gas turbines with a compact, fairly simple bottoming cycle that can provide ten additional percentages of efficiency (gross) to the stand-alone gas turbine. It is worth noting in this regard that it is actually the simple recuperated cycle which yields highest combined cycle efficiency (contrary to what could be expected from more complex layouts).

These conclusions are in line with those obtained by other authors before: [12] and, especially, [14]. Indeed, Angelino stated that supercritical carbon dioxide is less efficient than supercritical steam for turbine inlet temperatures lower than 550ºC: "*at low temperatures (400-550ºC), a CO_2 cycle, although inferior to steam cycle with respect to efficiency, could prove economical on account of simplicity and compactness*". This statement, which was applied to stand-alone condensing CO_2 cycles, is even stronger when non-condensing cycles are considered, as in this work.

A further step is taken in the next section, where it is explored if there is any potential gain (with respect to combined cycle efficiency) remaining in the gas turbine by modifying pressure ratio only. Other works in the past have considered higher turbine inlet temperatures on the rationale that these are likely to characterise future gas turbines. Nevertheless, increasing turbine inlet temperature means higher capital costs and more demanding thermal cycling when the engine is subjected to frequent start/stops. Hence, a different approach has been considered whereby turbine exhaust temperature is increased by reducing pressure ratio. This design modification is expected to be more affordable and, ideally, bring about lower capital costs due to fewer turbomachinery stages and smaller footprint (higher specific work). This is the expected route to cost competitiveness of combined cycle power plants as discussed in the introduction.

Optimisation of gas turbine pressure ratio

This last section explores the potential gains that could be obtained if the pressure ratio of the gas turbine were left free to vary. In the process, all the remaining specifications of the engine remain constant; this applies to compressor/turbine efficiencies, inlet air mass flow rate and turbine inlet temperature amongst other less important parameters. The most interesting results are presented in Table 7.

Parameter	***H_2O***	***CO_2***		
		SR	***PR***	***RC***
GT pressure ratio [-]	8:1	14:1	12:1	11:1
GT exhaust temperature [ºC]	749.3	623.0	657.0	676.5
GT efficiency[1] [%]	30.57	34.61	33.70	33.10
GT output[2] [kWe]	78420	77045	78570	79053
Stack temperature [ºC]	113.0	343.7	408.6	393.27
WHR pinch point [ºC]	16.6	16.6	16.6	16.6
WHR efficiency [%]	87.75	37.06	32.05	35.33
Bottoming cycle efficiency[1] [%]	41.73	40.81	43.88	47.40
Mass flow ratio $\dot{m}_{BC}/\dot{m}_{GT}$ [-]	0.19	1.10	1.39	1.44
Power ratio $W_{BC}/\dot{W}_{GT}$ [-]	0.82	0.35	0.34	0.41
Bottoming cycle output[2] [kWe]	64345	27032	26356	32245
Combined cycle output[2] [MWe]	142.8	104.1	109.9	111.3
Combined cycle efficiency[1] [%]	55.65	46.75	45.00	46.59

[1] Thermal efficiency
[2] Gross output at generator terminals

Table 7. Results for the reference bottoming cycles and variable gas turbine pressure ratio.

First of all, a screening of the possible pressure ratios in a wide enough range reveals that the pressure ratios for peak gas turbine efficiency and specific work (kJ/kg) are 10:1 and 22:1 respectively. This explains the design pressure ratio of 15.4:1, Table 1, according to the known existence of a pressure ratio for peak combined cycle efficiency in between the two aforecited values [4].

Regarding the steam cycle, the performance of the combined cycle is favoured by very low pressure ratios. This is due to several reasons:

- The bottoming cycle is more efficient.
- Higher turbine exhaust temperatures enable higher WHR efficiency inasmuch as the feedwater inlet temperature is very low and the pinch point of this boiler is found

internally (i.e., somewhere along the flow path and not at one end as in fired boilers).

Reducing pressure ratio brings about lower stack temperatures and biases power production towards the steam cycle. In fact, it is observed that for the minimum pressure ratio accepted (8:1), the contribution of the bottoming cycle to power generation is close to that of the gas turbine ($\dot{W}_{BC} = 0.8 \cdot \dot{W}_{GT}$). This minimum pressure ratio is the pressure ratio corresponding to a turbine exhaust temperature of 750ºC, which can be considered as a limiting value for the high temperature piping of the heat recovery boiler [4] [5].

Regarding the carbon dioxide cycles, they all experience an efficiency increase when pressure ratio is changed but this is lower than 0.5 percentage points. It could then be said that there are no gains for carbon dioxide coming from fine-tuning the topping cycle to the bottoming cycle needs, the reason for this being the little effect that this has on the performance of the waste heat recovery unit. Indeed, a closer look at the stack temperature values in Tables 6 and 7 reveals the following.

In as far as the bottoming steam cycle is concerned, the pinch point of the boiler is in the evaporator. Thus, when turbine exhaust pressure is increased because of the lower pressure ratio, stack temperature drops what and thus WHR efficiency increases. This is shown in Figure 5 which presents the impact of changing pressure ratio of the gas turbine from 15.4:1 to 8:1 on the T-Q (temperature vs. heat exchange) diagram of the waste heat recovery unit. For the cases under consideration, reducing stack temperature from 214.7ºC to 113.0ºC raises efficiency from 67.60% to 87.75%. This is observed in Figure 6 which presents the impact of variable pressure ratio on the efficiency of the WHR unit. Therefore, for the case of steam, it is confirmed that η_{WHR} is very sensitive to pressure ratio.

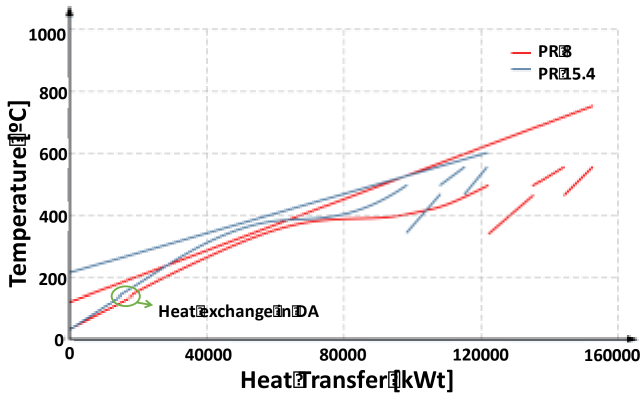

Figure 5. T-Q diagram of the WHR unit for a supercritical bottoming steam cycle with gas turbine pressure ratio of 15.4:1 (blue) and 8:1 (red).

In the bottoming carbon dioxide cycle, the pinch point of the WHR unit is located at the cold end of the heat exchanger (i.e., the stack). This means that stack temperature is actually constant as long as the specifications of the bottoming cycle remain constant. In particular, this temperature (T_{stack}) can easily be calculated as the temperature at the outlet of the high temperature recuperator of the CO_2 cycle plus the selected pinch point. In this work, the pinch point is set to 16.6°C (~9.5ºC) which corresponds to a WHR effectiveness of 92%. It is to note in this regard that effectiveness must not be mistaken for efficiency. Effectiveness quantifies the amount of heat recuperated with respect to the maximum heat recoverable, which depends on the inlet temperature of both fluids. Efficiency is a measure of the fraction of heat recovered with respect to the sensible heat carried by the exhaust gas stream, which depends on the gas turbine exhaust temperature and on ambient temperature.

Further to the discussion in the previous paragraph, it must be highlighted that even if stack temperature remains constant, this does not mean that WHR efficiency remains also constant when pressure ratio is changed. Actually, when pressure ratio is reduced, the gas turbine exhaust temperature increases and thus there is more sensible heat available at turbine exhaust. If the flue gases are left at the same temperature after heat recovery (same T_{stack}), the fraction of heat recovered is higher. This explains why reducing WHR efficiency increases at lower pressure ratios also for the carbon dioxide cycles.

The same reason explains why the sensitivity of η_{WHR} to pressure ratio is higher for steam. At lower pressure ratios, bottoming steam cycles benefit from both effects, more energy available at WHR inlet and lower stack temperature. On the contrary, carbon dioxide cycles benefit from the first effect only and thus the sensitivity of η_{WHR} to pressure ratio is lower.

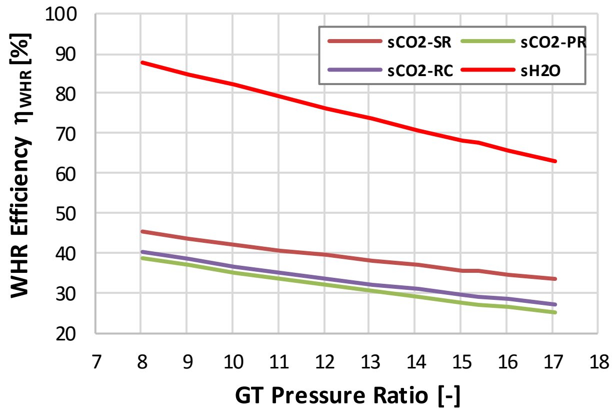

Figure 6. Impact of gas turbine pressure ratio on WHR efficiency.

A summary plot is presented in Figure 7, confirming the observations discussed in this and the previous sections. Supercritical carbon dioxide cycles seem to be able to provide an efficiency gain of some 10 percentages with respect to the stand alone gas turbine, regardless of the pressure ratio of choice. Actually, the moderate impact of lower pressure ratios on WHR efficiency is offset by the detrimental effect on gas turbine efficiency. This yields a flat plot which suggests that, irrespective of the gas turbine considered, the performance of a GT & sCO_2 combined cycle would be similar for a given set of design specifications of the bottoming system.

The case is different for GT & sH_2O cycles. For these, the results credit that very simple bottoming cycle layouts can potentially attain 55% thermal efficiency when coupled with moderate pressure ratio gas turbines. Interestingly, the dependence of η_{CC} on the pressure ratio of the gas turbine seems to fade out at pressure ratios lower than 10:1, meaning that there is no need to go for very low pressure ratio engines whose design is farther from state-of-the-art engines.

Moreover, at 10:1, the gas turbine considered in this work would have an exhaust temperature of about 700ºC which, even if high, would be compatible with the current manufacturing technology of both the turbine and the waste heat recovery boiler.

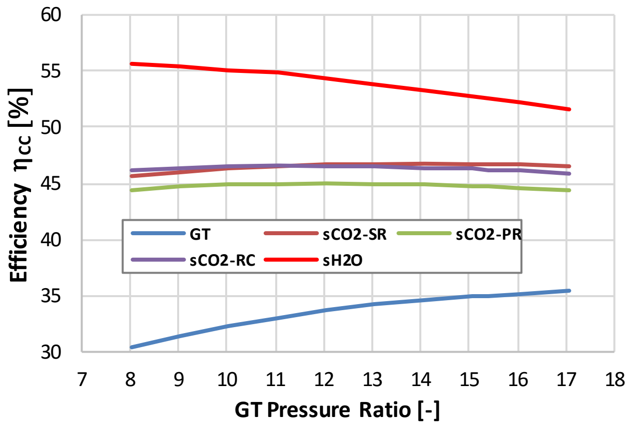

Figure 6. Impact of gas turbine pressure ratio on combined cycle efficiency.

Quick assessment of the potential gains from sCO_2 cycle optimisation

As a complement to the core study presented in this work and in order to incorporate other approaches available in the public domain (consisting mainly in exploring the potential efficiency gains coming from bottoming cycle optimisation), the effect of changing the pressure ratio of the bottoming sCO_2 cycle is evaluated. To this end, the simple recuperated layout is considered given that this is the sCO_2 layout yielding highest efficiency.

The results of this exploratory analysis are presented in Figure 8, where the inlet pressure to the CO_2 turbine has been increased from the reference value of 250 bar up to 350 bar and the pressure ratio of the gas turbine has been set to the optimum value for this cycle (14:1). The rest of specifications of the topping and bottoming systems remain as indicated in Tables 1 and 3. The main observations are:

- The efficiency of the Waste Heat Recovery unit increases due to the lower stack temperature that comes about because of the lower temperature of carbon dioxide at the high pressure outlet of the recuperator (inlet stream to the WHR). This lower temperature is caused by a larger expansion ratio across the turbine (note that turbine inlet temperature remains constant).
- The performance of the sCO_2 cycle benefits from the higher turbine inlet pressure and thus the corresponding efficiency (η_{BC}) increases, even if only slightly because turbine inlet temperature remains constant.
- In spite of the constant value of gas turbine efficiency, the combined cycle benefits from the better performance of the bottoming system and hence η_{CC} increases moderately.

Thermal efficiencies in the order of 48% are thus possible by merely increasing turbine inlet pressure to values between 300 and 350 bar. When accompanied by a similar increase in turbine inlet temperature, this configuration should easily enable thermal efficiencies in excess of 50%, as already suggested by Angelino in [14]. This global optimisation of the topping (pressure ratio) and bottoming (turbine inlet pressure and temperature) is currently under analysis by the authors.

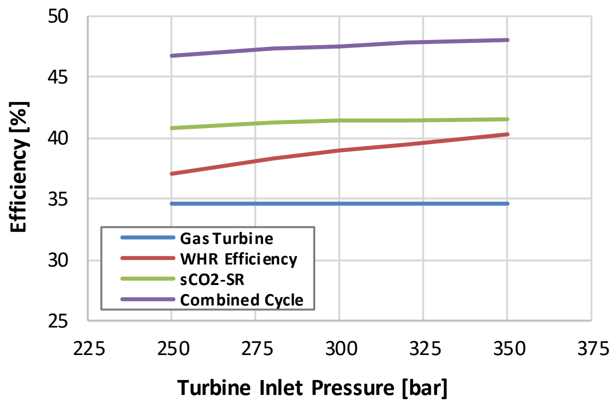

Figure 8. Impact of bottoming turbine inlet pressure on the performance of a GT & sCO_2 using a simple recuperated carbon dioxide cycle.

CONCLUSIONS

The analysis presented in this work aims to complement the existing literature assessing the potential of combined cycles using a gas turbine and either a supercritical steam or supercritical carbon dioxide bottoming cycles. A few works have already been published, most of which claim that the utilisation of carbon dioxide does not seem to provide enough operational benefits (flexibility, compactness) so as to compensate for the lower efficiency.

With such background, the interest of this work lies in the fact that it puts emphasis on the possible efficiency gains coming from variations in the gas turbine cycle in lieu of the usual approach considering an off-the-shelf engine. The main conclusions follow:

- In terms of balance between simplicity of the bottoming system (cycle layout, WHR) and combined cycle efficiency, supercritical steam cycles are a very interesting option for the current renewable-dominated energy scene. This is particularly true if low pressure ratio engines are developed (say pressure ratios in the order of 10:1) with the same firing temperature levels as contemporary engines.
- The main benefit of supercritical carbon dioxide cycles is arguably the smaller footprint. The turbomachineries are smaller (Table 6) and even if there are several heat exchangers, these are expected to be equivalent in size to the Waste Heat Recovery unit and condenser of a steam

cycle. A secondary benefit could be found in the lack of air infiltrations into the power cycle, thus lower maintenance and operating costs.

- Unfortunately, it seems difficult to make a case for GT & sCO_2 combined cycles. They are able to increase the efficiency of the stand-alone gas turbine by some 10 percentage points but even in this case they are still some 5 percentages below the GT & sH_2O counterpart.
- As a final remark, it is interesting to highlight that GT & sCO_2 cycles exhibit their best performance at pressure ratios that are close to (slightly below) those of commercial gas turbines (~15:1) whereas GT & sH_2O require much lower pressure ratios in the gas turbine to attain peak performance.

The final conclusion is hence that supercritical steam cycles can definitely make a case for combined cycle power plants in the 100÷150 MWe size aiming to operate in secondary markets (low CapEx). The confirmation of this requires an assessment of the dynamic performance of the main components of the bottoming cycle, which is the natural next step of this work. Regarding supercritical carbon dioxide, it seems a less interesting option than steam for the boundary conditions considered.

NOMENCLATURE

BC	Bottoming Cycle
C	Compressor
CapEx	Capital Expenditures
CC	Combined Cycle
COOL	Cooler
GT	Gas Turbine
HRSG	Heat Recovery Steam Generator
HTR	High Temperature Recuperator
LHV	Low Heating Value
LTR	Low Temperature Recuperator
$\dot{m}$	Mass flow rate
OpEx	Operating Expenditures
P	Pressure
PR	Precompression
RC	Recompression
REC	Recuperator
sCO_2	Supercritical carbon dioxide
sH_2O	Supercritical steam
SR	Simple Recuperated
T	Turbine/Expander (only in cycle layouts)
T	Temperature
$\dot{W}$	Power output
WHR	Waste Heat Recovery
η	Efficiency

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