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Performance comparison of supercritical CO_2 versus steam bottoming cycles for gas turbine combined cycle applications

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Abstract

Supercritical CO2 (sCO2) power cycles have been proposed as an alternative to steam bottoming cycles for natural gas combined cycle applications. This paper presents a performance comparison of sCO2 versus steam bottoming cycles for heavy-duty and aeroderivative gas turbine combined cycles.

For large heavy-duty gas turbines such as the GE H-Class, the baseline is a three-pressure reheat (3PRH) steam cycle. Comparisons between a cascaded sCO2 bottoming cycle versus a 3PRH steam bottoming cycle are performed at maximum bottoming cycle fluid temperatures of 600°C and 700°C. When assuming very high sCO2 expander and pump isentropic efficiencies of 95%, the maximum pressure in the sCO2 cycle needs to exceed 300bar to outperform 3PRH steam. If the sCO2 expander and pump isentropic efficiencies are reduced to the usual level obtained with steam-based turbomachinery, the sCO2 cycle performance is lower than 3PRH steam by at least 0.8%pts in combined cycle net efficiency. An exergy flow analysis is performed to understand the underlying reasons for the lower performance of sCO2 versus 3PRH steam. As expected, the sCO2 cycle has lower exergy flow losses than steam in the gas turbine waste heat recovery unit. However these reduced losses are not sufficient to balance the higher pumping exergy flow losses versus steam and the additional exergy flow losses introduced by the sCO2 recuperators.

For smaller aeroderivative gas turbines such as the GE LM2500, the baseline bottoming cycle is a two-pressure non-reheat (2PNR) steam cycle. The comparison is performed at a maximum bottoming cycle fluid temperature of 500°C. When assuming realistic sCO2 expander and pump isentropic efficiencies, the sCO2 cycle can outperform a 2PNR steam cycle at pressures above 200bar. In this case, the sCO2 cycle has significantly reduced exergy flow losses in the gas turbine waste heat recovery unit. This overcomes the higher pumping exergy flow losses versus steam and the additional exergy flow losses occurring with sCO2 recuperation.

In conclusion it is likely that sCO2 bottoming cycles can achieve higher performance than 2PNR steam bottoming cycles typically used in conjunction with small industrial and aeroderivative gas turbines. However, sCO2 cycles would need very high component efficiencies and operating pressures to achieve higher performance than 3PRH steam bottoming cycles typically paired with large heavy-duty gas turbines. Other factors differentiating between sCO2 and steam bottoming cycles, such as cost, footprint, operability and maintenance are not included in this study.

Introduction

Alternatives to steam bottoming cycles have been studied mostly for small industrial and aeroderivative gas turbines, where added maintenance, operation complexity and the use of water are undesirable attributes of steam bottoming cycles. For example, organic Rankine cycles (ORCs) have been commercialized for gas turbines used in pipeline compressor stations (1), where, amongst others, the working fluid should not freeze or an operator cannot always be present on site. In applications requiring very compact bottoming cycles such as offshore platforms or ships, supercritical CO2 (sCO2) cycles have been considered in particular (2). As the maximum temperature in ORCs is limited by the degradation temperature of the organic fluid (350°C for the most stable hydrocarbons), they have not been proposed as bottoming cycle for heavy-duty gas turbines (HDGT), which typically operate with a higher exhaust temperature than their aeroderivative counterparts. The exergy flow losses due to the high temperature difference between the exhaust gas and the maximum bottoming cycle temperature would result in very low efficiency versus three-pressure reheat (3PRH) bottoming cycles used in state-of-the-art HDGT combined cycle plants. This argument does not apply to sCO2 cycles, as carbon dioxide is an inert and stable fluid that can be heated beyond the temperatures used in modern steam bottoming cycles (600°C and above). The goal of this study is to understand if sCO2 bottoming cycles can be an efficient alternative to steam for HDGT and aeroderivative combined cycle plants.

The primary comparison criteria used here is performance, i.e. net combined cycle efficiency or net bottoming cycle power output. However, besides any performance increase, seen in this study, criteria such as footprint and cost need to be considered subsequently and might lead to the design of different sCO2 cycles than considered here. One cycle configuration is considered in particular in this study. It features dual expansion and dual flow split and has been identified in previous work (3)-(4) as the best compromise between high utilization of exhaust gas heat and high bottoming cycle first law efficiency, leading to high bottoming cycle net power output and combined cycle (CC) net efficiency. It should be noted that this configuration differs from the configurations considered for single-cycle applications (e.g. recompression cycle), where only the first law efficiency should be maximized. The considered cycle is shown on Figure 1 and uses one high temperature (HT) and one low temperature (LT) expander.

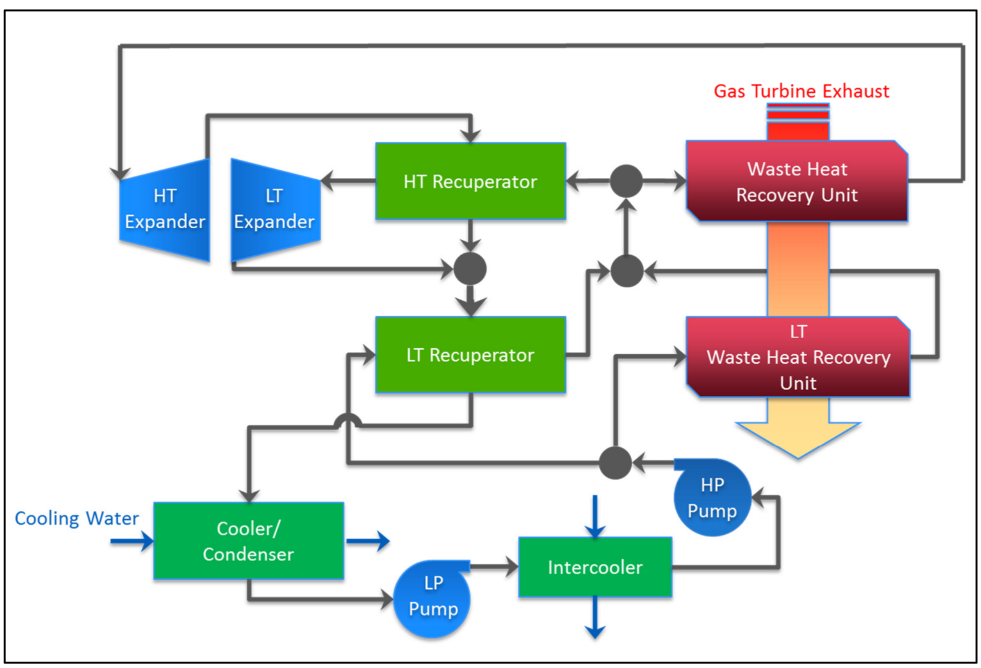

Figure 1 Considered sCO2 bottoming cycle

The temperature at the outlet of the HT expander is quite high and the available heat is used for heating one stream of high-pressure CO2 in an HT recuperator before being admitted to the LT

expander. The HT expander receives a second stream of high-pressure CO_2 from the high-temperature waste heat recovery unit (WHRU), where it is heated by exhaust gas to the maximum temperature in the cycle. After leaving the LT expander and the HT recuperator, the two streams that still have relatively high temperatures are merged and their available heat is used to preheat a part of the pressurized CO_2 from the feed pump in a LT recuperator. This heat exchanger is in parallel with a low-temperature WHRU, where the other part of pressurized CO_2 from the pump is preheated by cooling the exhaust gas to the stack temperature. The investigated configuration uses an intercooler in the feed pump to reduce the power consumption.
In (4) it was found that the sCO_2 bottoming cycle can show up to 0.7%pts better CC net efficiency than a plant using a 3PRH steam cycle. This CO_2 cycle runs without condensation with 32°C minimum temperature, a maximum pressure of 280bar, 92% turbine and 88% compressor efficiency. The baseline steam cycle is in an SGT5-4000F CC plant with an exhaust temperature of 580°C and a CC net efficiency of 58.4%. This is lower than current best-in-class efficiencies (60% to 61%) and significantly lower than future combined cycles considered for the next decade (62% up to 65%, (5)-(7)). This low baseline efficiency may be a reason for the result that sCO_2 bottoming cycles can give higher CC efficiencies than steam. Actually, another study (3) uses an H-Class GTCC with a higher exhaust temperature of 625°C as the steam benchmark and concludes that the sCO_2 dual-split bottoming cycle is not able to reach the performance of 3PRH steam bottoming cycle. The same study also considers the case of an LM6000 aeroderivative GT with a lower exhaust temperature of 471°C and a 2PNR bottoming cycle. In this case it concludes that the sCO_2 bottoming cycle can generate 9% more power than the steam cycle.
The goal of this work is to clarify the expected performance of this sCO_2 cycle in comparison with the best-in-class steam bottoming cycles that will be commercialized in the next decade. The first part of the paper will focus on the case of heavy-duty gas turbines with high exhaust temperature, while the second part will focus on aeroderivative gas turbines with lower exhaust temperature.

Comparison with three-pressure reheat bottoming cycles for heavy-duty gas turbines

Table 1 shows the two future 3PRH steam bottoming cycle baselines for the H-Class gas turbine.

Table 1 Steam 3PRH baseline combined cycle

GT Type	[-]	H-Class	
Configuration	[-]	2x1, 3PRH	
Case	[-]	Current exhaust temperature range	Theoretical higher exhaust temperature range
Exhaust temperature	[°C]	650-700	700-750
Steam maximum temperature	[°C]	600	700
CC net efficiency	[%]	62-62.5	>62.5

The boundary conditions used for the sCO_2 cycles are summarized in Table 2. The primary goal of this study is to understand if sCO_2 bottoming cycle can exceed the performance of steam bottoming cycles. Consequently a “theoretical” case needs to be obtained for sCO_2 and some boundary conditions use optimistic values. This is the case for the pump and expander isentropic efficiency, which are initially both set to 95%. The piping pressure losses are not considered and the slip streams needed for fuel heating as in the steam baseline are neglected.

Leakages and secondary flows are also neglected. Although these contribute to favorable values for sCO2 performance versus steam, care is taken throughout the study to ensure that the sCO2 cycle parameters are not unreasonable optimistic and, where possible, boundary conditions are set in a way to ensure fair and consistent modeling versus the steam baseline. Obviously, the same maximum working fluid temperature is used. The WHRU and the condenser overall heat transfer coefficients, noted UA, are constrained to be the same as in the steam baseline. The UA gives a first indication of the size of the heat exchanger equipment. This metric is used to avoid having much larger heat exchangers in either the steam or the CO2 cycle, resulting in a performance bias. As the heat transfer coefficients are not the same in the case of steam and CO2, a matching UA does not give exactly the same heat exchanger surface but at least ensures that the size of the equipment is comparable. In the case of a real installation, the UA and size of the condensers would result from thermo-economic optimizations performed for each cycle.

Table 2 "Theoretical" sCO2 bottoming cycles boundary conditions

Ambient	T	[°C]	15
	p	[bar]	1.01
	Relative humidity	[%]	60
Gas turbine	Minimum stack T	[°C]	70
CO2 EOS	na	na	Span-Wagner
Expanders	Isentropic efficiency	[%]	95
	Inlet pressure	[bar]	optimized
	Generator efficiency	[%]	98
	Mechanical efficiency	[%]	98
CO2 Pump	Isentropic efficiency	[%]	95
	Motor efficiency	[%]	98
Condenser	Tin water	[°C]	17
	Tout water	[°C]	23
	Subcooling	[°C]	0.5
	UA	[kW/°C]	same as steam baseline
	Hot side pressure drop	[% of p,in]	0.8
Intercooler	Tin water	[°C]	17
	Tout water	[°C]	23
	Tout CO2	[°C]	same as CO2 condenser outlet T
	Hot side pressure drop	[% of p,in]	0
Waste heat recovery unit (WHRU)	CO2 outlet T	[°C]	600 and 700
	Cold end approach	[°C]	Set to obtain same UA as steam HRSG baseline when adding WHRU and LT WHRU UAs
	Cold side pressure drop	[% of p,in]	5.4
HT Recuperator	Hot end approach	[°C]	4
	Cold end approach	[°C]	4
	Cold side pressure drop	[% of p,in]	0.7
	Hot side pressure drop	[% of p,in]	1.7
LT Recuperator	Cold end approach	[°C]	4
	Cold side pressure drop	[% of p,in]	0.7
	Hot side pressure drop	[% of p,in]	1.7
Low temperature WHRU	Minimum approach	[°C]	Set to obtain same UA as steam HRSG baseline when adding WHRU and LT WHRU UAs
	Hot side pressure drop	[% of p,in]	0
	Cold side pressure drop	[% of p,in]	0.7

Because of the ambient temperature and the constraints on the condenser size, the CO2 at the outlet of the cooler/condenser is in a liquid state and the investigated cycles are Rankine cycles with a relatively low power needed to compress the CO2 compared to Brayton cycles. The specifications for the recuperators result in very high effectiveness and surface area, which supports a high cycle efficiency but may not represent a thermo-economic optimum. The auxiliary loads in the sCO2 case are derived from the steam baseline values. The split ratio and intercooling pressure in the sCO2 cycle are optimized to maximize the net power output.
Figure 2 shows the performance of the “theoretical” sCO2 bottoming cycle operating at 600°C and 700°C maximum working fluid temperature relative to the 600°C and 700°C steam baselines, respectively. At 600°C, the maximum operating pressure in the sCO2 cycle (either at the inlet of the HT or LT expander, depending on the pressure drop assumptions) needs to exceed 250bar for the sCO2 cycle to show better performance than 600°C steam operating at a maximum pressure of 170bar (thermo-economic optimum). The maximum sCO2 efficiency is reached at 300bar, where the sCO2 CC net efficiency is 0.3%pts better than steam. Given the optimistic boundary conditions used, a realistic design would most probably be less efficient than steam at 600°C. At 700°C maximum working fluid temperature, the minimum sCO2 pressure needed to exceed steam performance is 290bar with a maximum improvement of 0.7%pts at 400bar. This pressure level is quite high for power production application and may result in engineering and sourcing challenges. As a reference, the highest pressure mentioned in research programs dealing with advanced ultra-supercritical coal power plant is around 350bar (8).

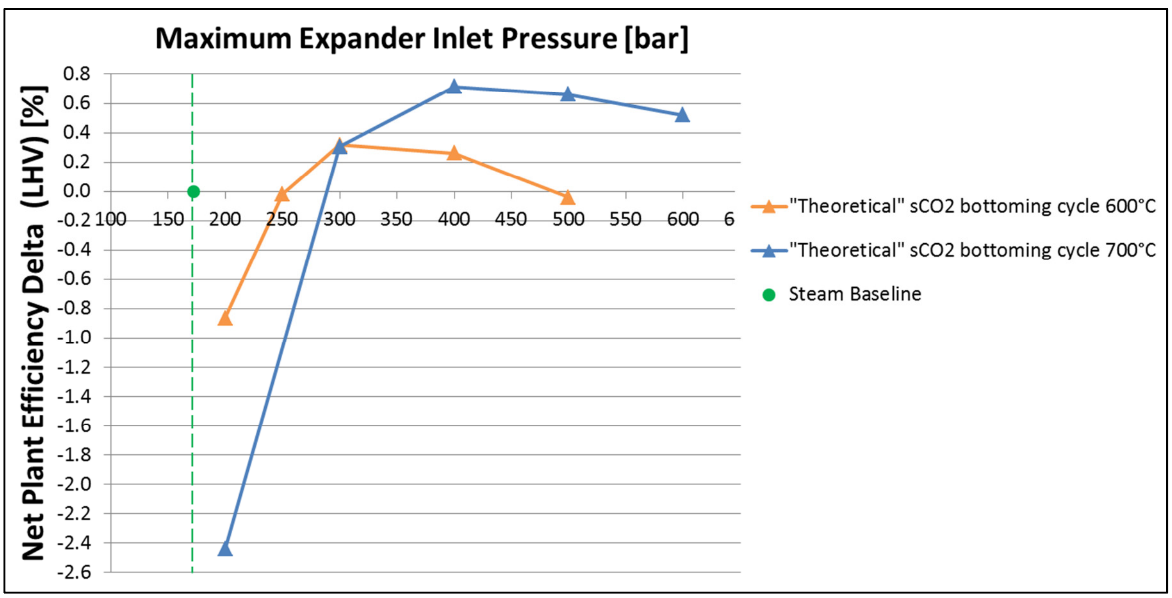

Figure 2 Relative CC net efficiency of steam and sCO2 bottoming cycles at 600°C and 700°C maximum working fluid temperature

Next, the sensitivity to CO2 turbine and pump efficiency is investigated. Given the minimal “theoretical” efficiency gain at 600°C, only the 700°C case is further considered. The sCO2 turbine and pump isentropic efficiencies are reduced to usual levels found in a steam bottoming cycle. As shown in Table 3, this revision of the settings reduces the overall efficiency (including isentropic, mechanical and generator/motor efficiencies) of the pump by 16%pts and the one of the expander by 2%pts. Figure 3 shows the impact of the revision on the net plant efficiency.

The optimum pressure level stays at 400bar, but there is a strong decline of net power at higher pressure caused by the revised pump efficiency.

Table 3 Revision of the expander and pump boundary conditions

All in %	"Theoretical" case	Revised boundary conditions
	Expander	
Overall eff.	**91**	**89**
	Pump	
Overall eff.	**93**	**77**

The net CC efficiency of the sCO2 cycle at the optimum pressure is 0.8%pts lower than that of a plant with the 700°C steam cycle when using the same pump and expander efficiencies. However, with the currently available materials it might be uneconomical to build a steam cycle with a maximum temperature of 700°C, because material cost increase and lifetime reduction might cancel out any benefit of higher plant efficiency. Therefore an additional steam case is run, where the maximum temperature is limited to 600°C. Even in this case, with the sCO2 cycle still operating at the maximum temperature of 700°C, the 600°C steam bottoming cycle shows 0.4%pts higher net CC efficiency.

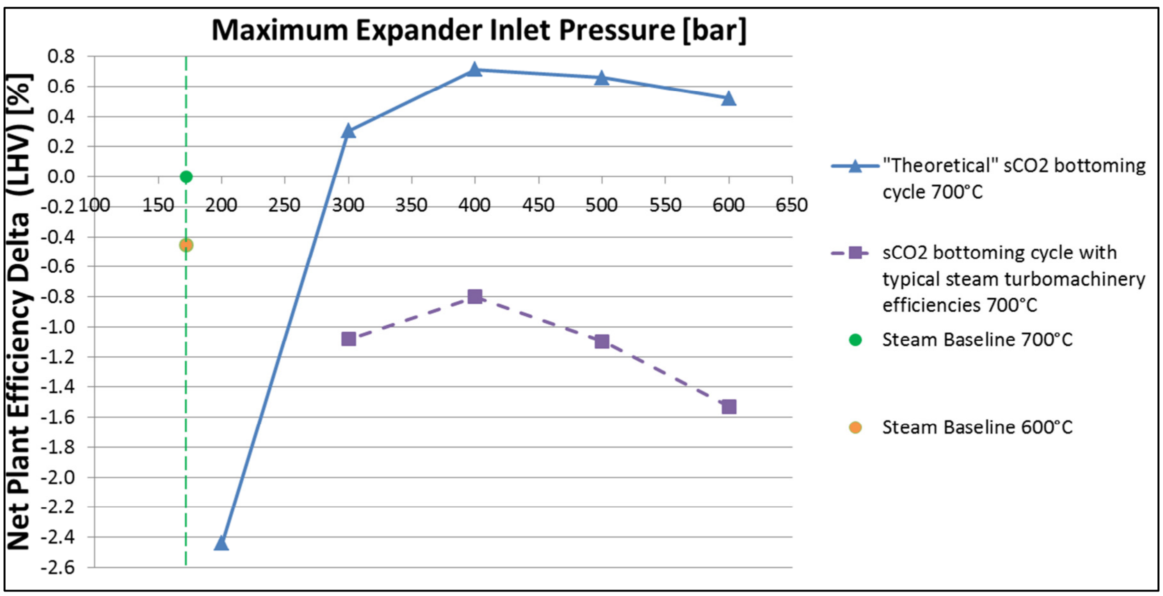

Figure 3 Impact of sCO2 expander and pump efficiencies on the relative CC net efficiency of steam and sCO2 expansion bottoming cycles at 700°C maximum working fluid temperature

If a detailed design would be performed for an optimized CO2 bottoming cycle, the expander and pump efficiencies would probably lie between the "theoretical" case and the revised steam-like efficiency cases considered in this study. Nevertheless, even with such an optimized design, the sCO2 cycle performance is unlikely to exceed that of the steam cycle at any pressure level below the highest pressure mentioned in research programs dealing with advanced ultra-supercritical coal power plant.

In order to understand the underlying reasons for the lower performance of the sCO2 in comparison with the steam cycle, three specific cycle configurations shown in Figure 3 are compared in further detail: the 700°C sCO2 cycle at 400bar with typical steam turbomachinery

efficiency levels, and the steam bottoming cycles operated at 700°C and 600°C. Table 4 gives an overview of the key parameters for each cycle.
The utilization of the exhaust gas heat by the sCO2 cycle is poorer than in the steam cases, as shown by the higher stack temperature and the lower WHRU thermal duty for sCO2. This lower utilization is partially compensated for by the higher sCO2 cycle first law efficiency, when compared to the 600°C steam case. This increase, however, is not sufficient to result in a higher net power output and subsequently higher net CC efficiency. When compared to the 700°C steam case, even the first law efficiency of the sCO2 cycle is lower. A positive aspect for sCO2 is a much lower condenser inlet volume flow, which will result in a more compact component for water-cooled plants.

Table 4 Relative comparison of key bottoming cycle parameters at 600°C and 700°C maximum working fluid temperature

		Steam, 700°C, 172bar	Steam, 600°C, 172bar	sCO2, 700°C, 400bar
Net CC efficiency delta	[% pts]	0.00	-0.45	-0.80
WHRU thermal duty	[%]	100.0	101.5	96.9
Difference to reference stack T	[°C]	0	-11	+29
Bottoming cycle first law efficiency delta	[% pts]	0.00	-1.50	-0.26
Net electrical power	[%]	100.0	98.0	96.4
Expanders electrical power	[%]	100.0	98.3	123.6
Pumps electrical power	[%]	100	113	1632
HT expander inlet volume flow	[%]	100.0	98.9	118.5
Condenser inlet volume flow	[%]	100.0	107.5	0.2

A comparative exergy analysis is performed to further illuminate the differences between sCO2 and steam bottoming cycles. This analysis focuses on the bottoming cycle with the boundaries and exergy flows shown in Figure 4.

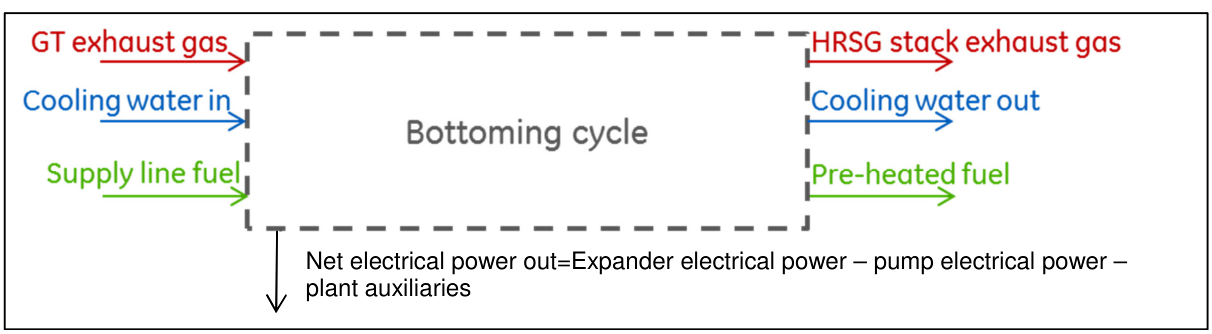

Figure 4 System boundary in the exergy flow analysis

Figure 5 shows the distribution of the incoming exergy flow in terms of net electric power, exergy flow losses and outcoming exergy flows. The total incoming exergy flow is lower in the sCO2 case, because fuel heating is not considered for CO2 but for steam. The net electric power output is lower in the sCO2 case than in both steam cases due to higher exergy flow losses and higher outcoming exergy flow. The exergy flow out is lower for sCO2 because no heated fuel leaves the cycle unlike in the steam case. If fuel heating is neglected in the steam cases, sCO2 has higher exergy outflow because of a higher stack temperature.
In Figure 6 and Figure 7 the exergy flow losses are compared in detail. The exergy flow losses occurring in the WHRU (including the LT WHRU for sCO2) are lower in the sCO2 case due to a better temperature match during the heat exchange between exhaust gas and the supercritical working fluid without isothermal evaporation. For the condensers the losses are similar in all cases, once the intercoolers losses are added to the main condenser losses in the sCO2 case. The losses during expansion are only slightly higher in the sCO2 case because of the larger gross power. The total exergy flow losses accounted up to this point are lower for the sCO2 case than in the steam cases. However, this changes when considering the pump exergy losses, which are much higher in the sCO2 cases. The losses of sCO2 compared to steam are further increased by the exergy flow losses occurring in the recuperators, which are not present in the steam cases.

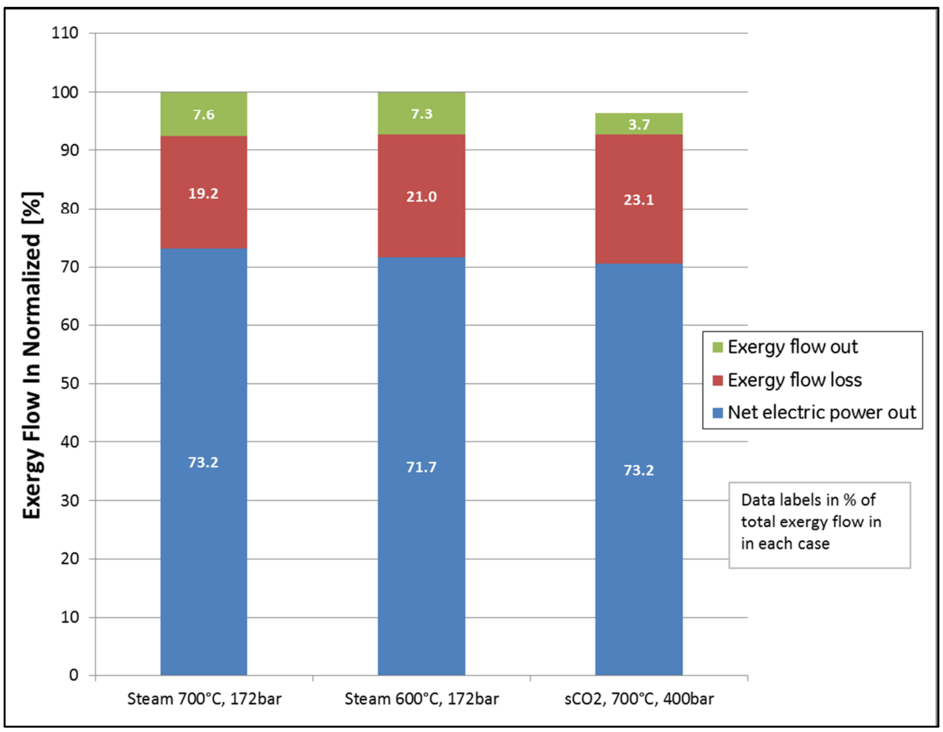

Figure 5 Distribution of net electric power, exergy flow losses and outcoming exergy flows

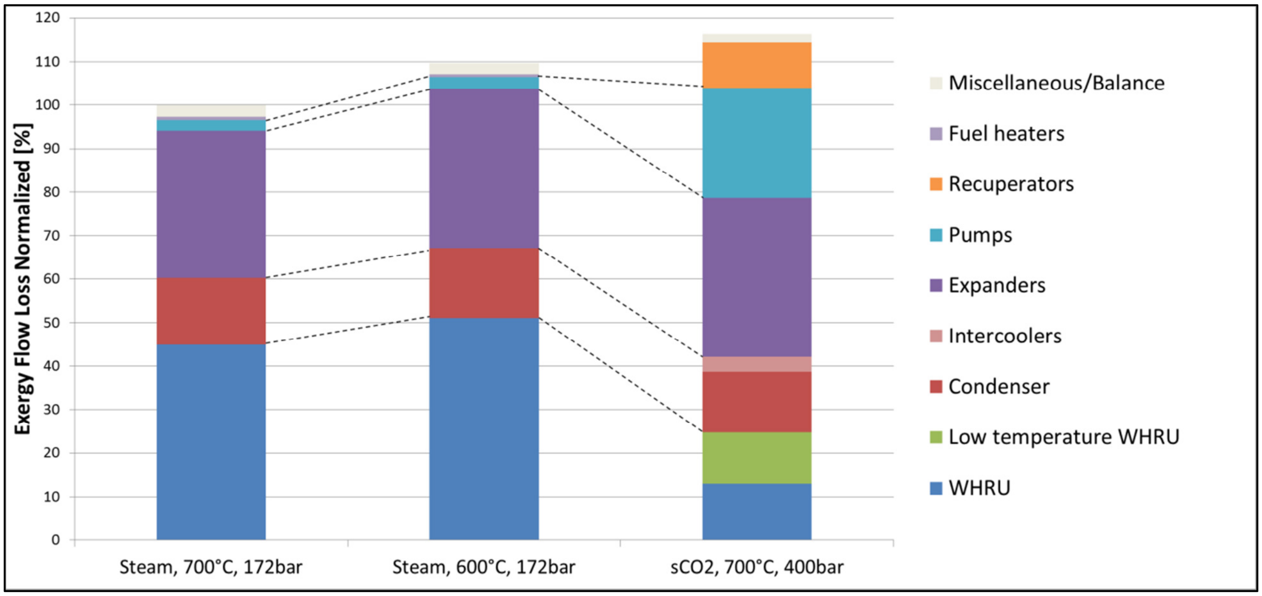

Figure 6 Comparison of exergy flow losses at 600°C and 700°C maximum working fluid temperature

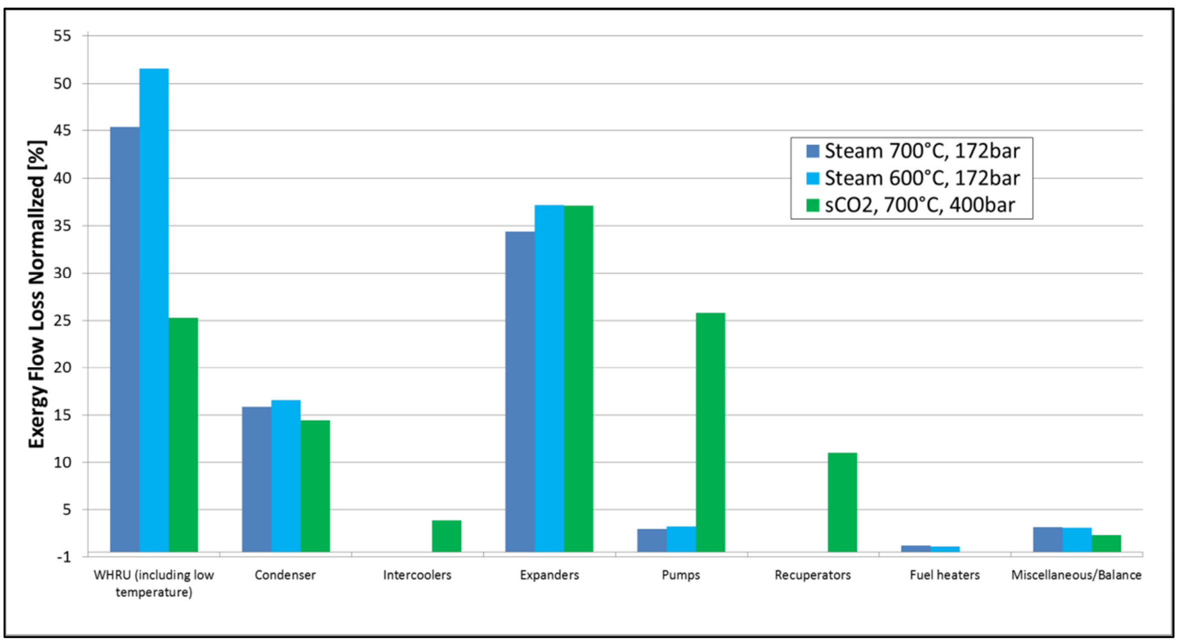

Figure 7 Comparison of exergy flow losses per component at 600°C and 700°C maximum working fluid temperature

Comparison with two-pressure non-reheat bottoming cycles for aeroderivative gas turbines

In the second part of this study, the same type of comparison between sCO2 and steam bottoming cycles is performed for a combined cycle based on an LM2500+G4 aeroderivative gas turbine. The exhaust temperature of aeroderivative gas turbines is typically lower than for their heavy-duty siblings, as they operate with higher pressure ratio. At lower exhaust temperature and smaller plant size, the 3PRH steam cycle configuration becomes less economical than a less efficient but simpler two-pressure non-reheat (2PNR) bottoming cycle. The steam baseline parameters used in this part of the study are given in Table 5. The maximum working fluid temperature in the bottoming cycle is now 500°C.

Table 5 Steam 2PNR baseline

GT Type	[-]	LM2500
Configuration	[-]	1x1, 2PNR
Exhaust temperature	[°C]	525-550
Steam maximum temperature	[°C]	500
CC net efficiency	[%]	52.5-53

The sCO2 bottoming cycle parameters, shown in Table 6, are also different to the previous larger scale. An air-cooled condenser is used (more common at smaller scale), and intercooling of the pumps is not considered. The expander and pump efficiencies have been adjusted due to smaller turbomachinery sizes. In general the specifications correspond to a more realistic near-term design than that in the large-scale sCO2 case in Table 2.

Table 6 sCO2 boundary conditions at 500°C maximum working fluid temperature

Ambient	T	[°C]	15
	p	[bar]	1.01
	RH	[%]	60
Gas turbine	Minimum stack T	[°C]	70
CO2 EOS	na	na	Span-Wagner
Expanders	Isentropic efficiency	[%]	85 to 90
	Inlet pressure	[bar]	optimized
	Generator efficiency	[%]	98
	Gearbox efficiency	[%]	98
CO2 Pump	Isentropic efficiency	[%]	75 to 80
	Motor efficiency	[%]	98
Condenser	Type	[-]	ACC
	Air inlet temperature	[°C]	15
	Air outlet temperature	[°C]	25
	dTmin	[°C]	Set to match steam UA
	Hot side dP	[%]	0.8
	Auxiliary consumption	[% of thermal duty]	1.32
	Subcooling	[°C]	0.5
Waste heat recovery unit (WHRU)	CO2 outlet T	[°C]	500
	Cold end approach	[°C]	Set to obtain same UA as steam HRSG baseline when adding WHRU and LT WHRU UAs
	Cold side pressure drop	[% of p,in]	5.4
HT Recuperator	Hot end approach	[°C]	4
	Cold end approach	[°C]	4
	Cold side pressure drop	[% of p,in]	0.7
	Hot side pressure drop	[% of p,in]	1.7
LT Recuperator	Cold end approach	[°C]	4
	Cold side pressure drop	[% of p,in]	0.7
	Hot side pressure drop	[% of p,in]	1.7
Low temperature WHRU	Minimum approach	[°C]	Set to obtain same UA as steam HRSG baseline when adding WHRU and LT WHRU UAs
	Hot side pressure drop	[% of p,in]	0
	Cold side pressure drop	[% of p,in]	0.7

The performance comparison is shown in Figure 8. For a small aeroderivative gas turbine bottoming cycle, sCO2 can outperform steam operating at the maximum pressure of 40bar (optimized to maximize CC net efficiency) when the maximum pressure is higher than 200bar, which is lower than in the heavy-duty GT case. An optimum of 0.5%pts additional CC net efficiency gain over the steam baseline is reached at 250bar. Despite a more near-term design with less optimistic boundary conditions than in the previous comparison for large heavy-duty GTs, the sCO2 cycle shows superior performance to a steam bottoming cycle for aeroderivative gas turbines.

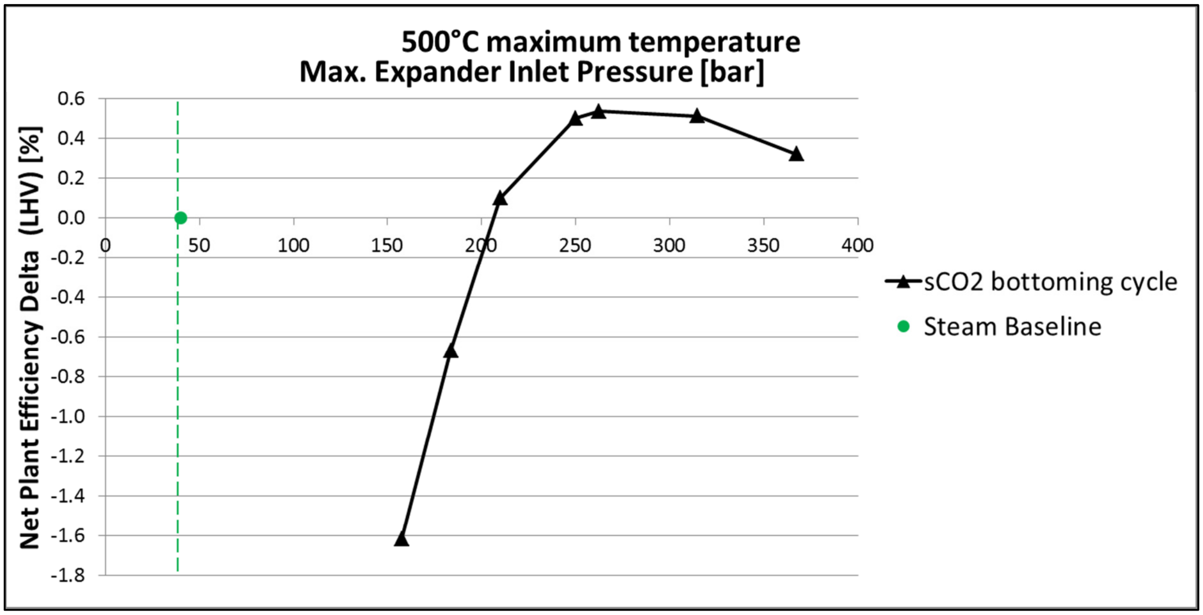

Figure 8 Relative CC net efficiency of steam and sCO2 bottoming cycles at 500°C maximum working fluid temperature

Cycle data listed in Table 7 show higher sCO2 performance for the optimum pressure level of 250bar compared to steam. As in the previous HDGT case, the utilization of the exhaust gas heat is poorer for sCO2 than for steam and the stack temperature is higher. However, a higher efficiency compensates for the lower utilization and the net power output of the sCO2 cycle is 3.5% higher than steam, corresponding to 0.5%pts CC efficiency increase. The expander and condenser volumetric flow rates are much lower.

Table 7 Comparison of key bottoming cycle parameters at 500°C maximum working fluid temperature

		Steam, 500°C, 40bar	sCO_2, 500°C, 250bar
Net CC efficiency delta	[% pts]	0.00	0.50
WHRU thermal duty	[%]	100.0	96.5
Difference to reference stack T	[°C]	0	+23
Bottoming cycle 1st law efficiency delta	[% pts]	0.00	+2.07
Net electrical power	[%]	100.0	103.5
Expanders electrical power	[%]	100	135
Pumps electrical power	[%]	100	3884
HT expander inlet volume flow	[%]	100.0	49.6
Condenser inlet volume flow	[%]	100.0	0.2

An exergy flow analysis shows in Figure 9 and Figure 10 why the sCO_2 cycle performs better than the 2PNR steam cycle. While the incoming exergy flow in the steam and sCO_2 cycles is the same (no fuel preheating), lower exergy flow losses than steam (-4%pts) lead to higher power output and efficiency of sCO_2, despite higher stack temperature losses.

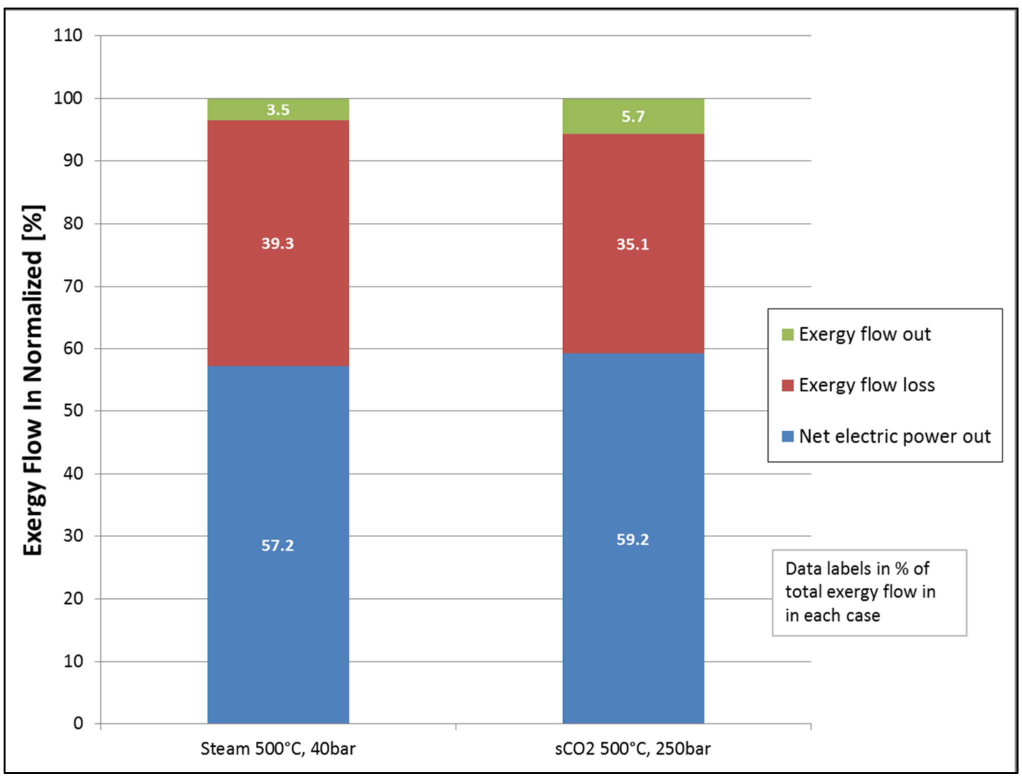

Figure 9 Distribution of net electric power, exergy flow losses and outcoming exergy flows at 500°C maximum working fluid temperature

These exergy flow losses are dissected in Figure 10. Unlike in the previous HDGT case, the reduction in WHRU exergy flow losses of the aeroderivative GT sCO2 cycle is large enough to make up for the higher pumping and additional recuperator losses and result in overall lower exergy flow losses than for steam.

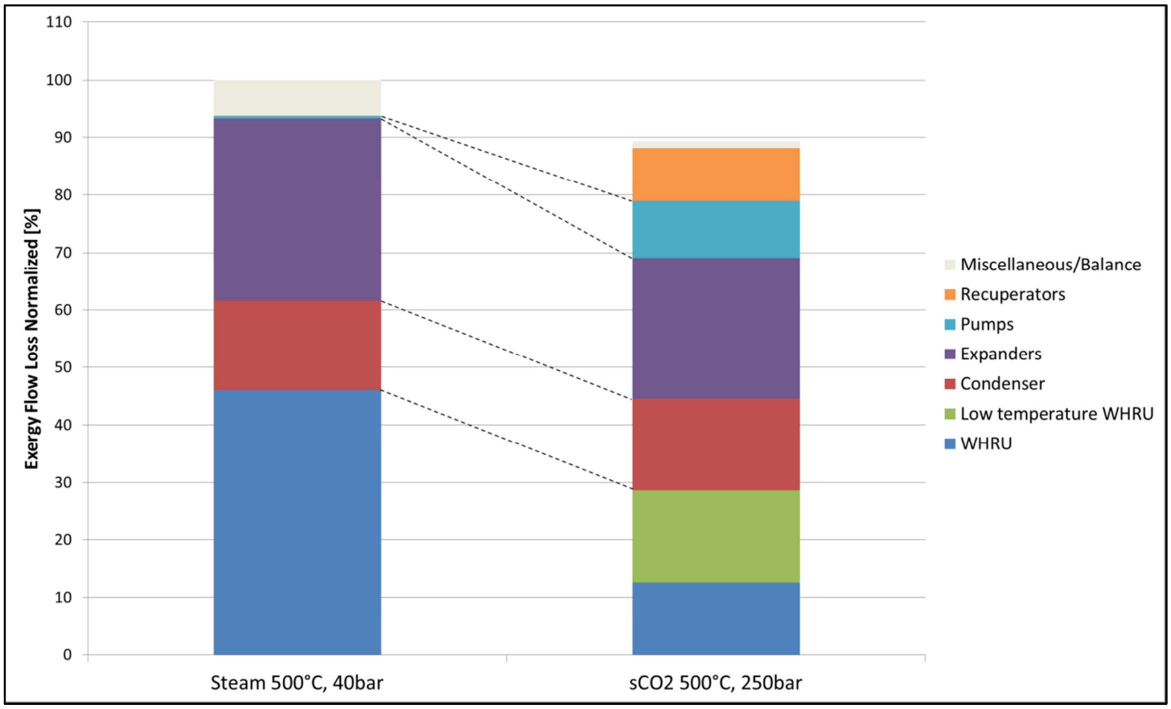

Figure 10 Comparison of exergy flow losses at 500°C maximum working fluid temperature

Conclusion

The optimized sCO2 bottoming cycle investigated in this study does not exceed the performance of a three-pressure reheat steam bottoming cycle recovering heat from future heavy duty gas turbines at maximum working fluid temperatures from 600°C up to 700°C, unless it operates at pressure in excess of 300bar, uses intercooling and has 95% efficient turbomachinery. The reason is that lower exergy flow losses in the sCO2 waste heat recovery unit, as compared to steam, are not sufficient to counter balance the higher exergy flow losses in the pumps and recuperators.

In contrast, when compared with a two pressure non-reheat steam cycle recovering heat from an aeroderivative gas turbine at a maximum working fluid temperature of 500°C, the reduction in WRHU exergy flow losses is more than enough to cover the higher pumping and recuperation losses. Consequently, sCO2 can outperform steam bottoming cycle for aeroderivative gas turbines at more feasible pressure levels (250bar) and with reasonable sCO2 expander and pump efficiencies at this scale.

Nomenclature

CC	Combined cycle
Eff.	Efficiency
GT	Gas turbine
GTCC	Gas turbine combined cycle

HDGT	Heavy-duty gas turbine
HP	High pressure
HRSG	Heat recovery steam generator
HT	High temperature
LP	Low pressure
LT	Low temperature
pin	Inlet pressure
UA	Overall heat transfer coefficient
Tin	Inlet temperature
Tout	Outlet temperature
WHRU	Waste heat recovery unit
3PRH	Three pressure reheat
2PNR	Two pressure non reheat

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Short biography and picture of the presenting author

Pierre Huck is a Lead Engineer in the Energy Systems Lab at GE Global Research in Munich, Germany. He joined GE in 2007. His work has been mostly dealing with cycle design and simulation for waste heat recovery from gas engines, aeroderivative and heavy-duty gas turbines. He has been particularly focused on organic Rankine cycle and supercritical CO2 bottoming cycles.