Evaluation of Gas Turbine Exhaust Heat Recovery Utilizing Composite Supercritical CO_2 Cycle

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ABSTRACT

Bottoming cycles are drawing a real interest in a world where resources are becoming scarcer and the environmental footprint of power plants is becoming more controlled. Reduction of flue gas temperature and power generation boost without burning more fuel are certainly very attractive, but how much can really be extracted from a Gas Turbine Unit (GTU) cycle to be converted to a combined configuration?

The current study presents a continuation of our previous work in which supercritical CO_2 (S-CO_2) bottoming cycles were considered as an alternative to traditional steam cycles for purposes of waste heat recovery. Prior evaluation regarding the option of combined gas and S-CO_2 cycles for different gas turbine sizes, gas turbine exhaust gas temperatures and configurations of bottoming cycle type allowed making conclusions about the advisability of the mentioned heat utilization technology and selecting the best embodiment of bottoming cycle taking into account heat source features.

In the present paper combinations of different S-CO_2 cycles are considered as bottoming parts for a specific middle power gas turbine. Such approach allows combining the advantages of different types of S-CO_2 cycles to increase the power production utilizing GTU waste heat. Also, a composite cycle approach is proposed to provide a positive effect from the combination of cycles and significantly decrease the number of components compared to what it would be for a sequential S-CO_2 cycles installation. The comparison of combined gas and supercritical CO2 cycles with traditional combined gas and steam cycles are presented in this paper. The specified comparison was performed in terms of additional power magnitude from the bottoming cycle installation and additional number of components.

INTRODUCTION

Gas turbines make a great contribution to global electricity generation. Analysis of the energy market shows that in the future the portion of electricity generated by gas turbines will only increase. The efficiency level of modern GTU (Gas Turbine Units) operating in a Brayton cycle is above 40%. The best way to increase GTU efficiency and decrease the environmental footprint is to recover the waste heat by adding a bottoming cycle. Traditionally, as the bottoming cycle, a steam turbine cycle on the basis of a Rankine cycle is used. The efficiency level of advanced combined cycles (GTU coupled with steam cycle) can exceed 60% [1]. However, due to the features of steam cycles (huge Heat Recovery Steam Generators, - HRSG -, and condenser, lots of auxiliary equipment, difficulties in scaling down steam systems, and so on) usually only high power units with a power level above 120 MW are configured for combined cycle service. Gas turbines of small and medium size are typically sold and operated as simple cycle units [2].

A bottoming S-CO_2 cycle makes possible to utilize residual heat from the GTU exhaust, producing additional energy and improving the overall efficiency of the system. It is evident that different types of bottoming S-CO_2 cycles have a different ability of heat utilization and its conversion into electricity. Also, S-CO_2 cycles may have a significantly high efficiency of heat-to-electricity conversion but may have a small temperature difference at the heater that significantly limits the GTU exhaust gases heat utilization degree. I.e., a high efficiency bottoming S-CO_2 cycle may give less of an electricity production addition to the whole system than one with a moderate efficiency but a higher S-CO_2 temperature difference at the heater.

Reasonability of bottoming cycle application after GTU is considered in papers such as [2, 3 and 5]. It is particularly stated in [2] that S-CO_2 technology can replace steam for bottom cycling on gas turbines by providing higher output power with lower installed cost and lower O&M costs. In comparison to heat recovery steam generators, the higher energy density of S-CO_2 reduces the system components size and cost, and provides significant advantages regarding system efficiency, footprint and ease of installation. Heat utilization from a variety of GTU with different exhaust gas parameters (mass flow rate, temperature, etc.) using standardized bottoming S-CO_2 units was presented in [3]. Different configurations of bottoming S-CO2 cycles applied after H-Class Siemens GTU and after GE LM6000 GTU were considered in [8]. The author of the report obtained ambiguous results according to which a bottoming S-CO_2 cycle is not reasonable for an H- Class GTU, but is reasonable for a GE LM 6000. Based on this, he concluded that all considered cycle configurations are suitable for low power, low GTUs exhaust temperatures.

In the aforementioned papers it was shown that bottoming S-CO_2 cycles make sense in terms of additional power production utilizing GTU exhaust gases but the cycles considered in those papers still have some potential for power production due to rather high residual temperature of exhaust flue gases (EFG) [2, 3] from one side and moderate efficiency of bottoming cycle itself [5] from another. It is evident that to accommodate this potential the residual temperature of the exhaust gases after the bottoming S-CO_2 cycle(s) should be as low as possible and the thermal efficiency of these cycles should be as high as possible. It should be noted that flue gases at relatively small temperatures become quite aggressive for the contiguous components of a power unit. Thus, for liquid fuels the minimum allowable temperature of flue gases is 120 – 125 °C and for natural gas it is 80-100 °C.

Unlike our previous paper [3] which was devoted to different GTU heat utilizations using standardized, relatively simple, bottoming S-CO_2 units, this paper describes the problem of heat utilization from another perspective: how much additional power can

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be produced from specific GTU when their heat is utilized by a bottoming S-CO_2 cycle complication.

A single, medium power GTU GE LM6000 PH DLE/Sprint [2] was selected as a topping GTU for the study. The exhaust gas parameters were fixed and correspond to the data presented in the table 1.

Table. 1 Selected set of GE LM6000-PH DLE parameters

Parameter	Unit	Magnitude
GTU thermal efficiency	%	42.3
GTU power	MW	53.26
Exhaust gas mass flow rate	kg/s	138.8
Exhaust gas temperature	°C	471
Exhaust gas pressure	MPa	0.15*
Bottoming steam cycle power	MW	14

* Since the exhaust gas pressure was unknown it was assigned as 0.15 MPa.

Since this GTU has different revisions with slightly different parameters, it was decided to use the parameters of the revision of this GTU that were considered in [5]. Usage of similar conditions allows us to compare our results with the results from [5] whose authors obtained a power increment of 15.2 MW in their best case.

Taking into account that the GTU has diverse fuel capabilities [4] the EFG temperature magnitude of 90 °C was used in the current study as the minimal allowable temperature of EFG in assumption of natural gas fuel utilization. However it does not mean that the EFG temperature will be equal to 90 °C in each considered cycle, because not every S-CO_2 cycle is able to give such a low residual EFG temperature.

The studies of combined and bottoming S-CO_2 cycles were performed with the use of the heat balance calculation tool AxCYCLE™ [6, 7]. S-CO_2 properties correspond to NIST data [6].

APPROACH

The first idea was to utilize heat from the GTU by a series of S-CO_2 cycles to find the most productive combination. This series of cycles was created as a sequential installation of different combinations of S-CO_2 cycles as shown in Fig. 1. According to this approach the high temperature EFG after the GTU goes to a first high temperature bottoming S-CO_2 cycle and then to a second low temperature one.

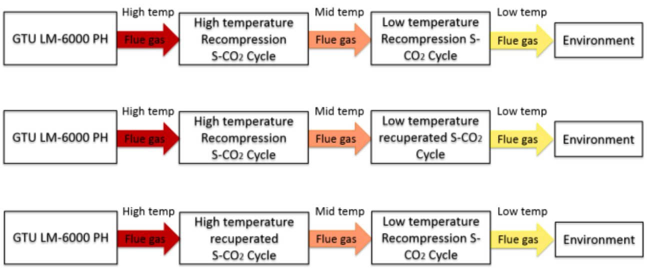


Fig. 1 Some examples of bottoming S-CO_2 cycles combinations

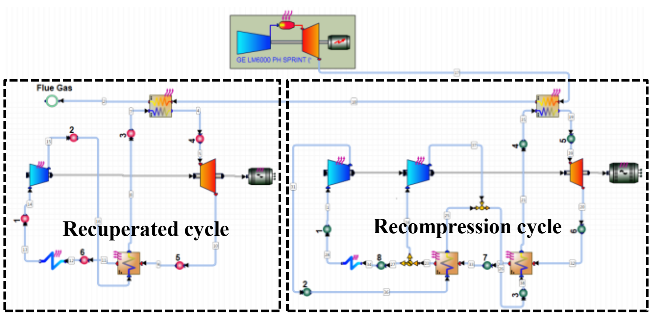
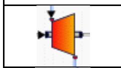
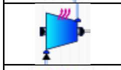
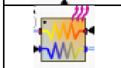
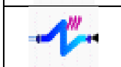
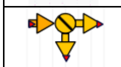
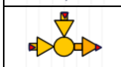
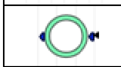


Fig. 2 Combination of recompression S-CO_2 cycle and recuperated S-CO_2

The first combination which was made is shown in Fig. 2. This is a combination of a recompression S-CO_2 cycle (high temperature) and a recuperated S-CO_2 cycle (low temperature). The background for this combination creation was an idea to use the highly efficient recompression cycle for the high temperature EFG utilization while taking into account the rather high flue gas residual temperature after the Intermediate Heat Exchanger (IHX) of this cycle. Moreover, the recuperated cycle was installed to utilize the low temperature EFG residual heat and increase the total power production.

The designations which are used in Fig. 1 and in further figures are presented in Table 2.

Table. 2 The designations in the figures

Picture	Meaning
	Turbine
	Compressor
	Alternator
	Recuperator
	Intermediate Heat Exchanger (IHX)
	Cooler (heat sink)
	Control splitter
	Mixer
	Environment or external fluid mass inlet/outlet
	Process point (used to build the process diagrams)

The design parameters of the S-CO2 cycle components and the main cycle parameters are given in Table 3. The efficiency values for turbine and compressor were obtained by our preliminary estimations utilizing AxSTREAM™ [9, 10, 11, 12].

Table. 3 Design parameters of S-CO_2 cycle components

#	Parameter	Unit	Value
1	Turbine efficiency	-	0.9
2	Compressor efficiency	-	0.85
3	Flue Gas/CO_2 HEX efficiency	-	0.9
4	Pinch for CO_2/CO_2 recuperator	°C	5
5	Generator efficiency	-	0.99
6	Temperature at cooler outlet	°C	33

The thermodynamic parameters of the cycles were optimized without taking into consideration the residual heat of the flue gas, allowing to obtain rectified cycles thermal efficiency. Thus, for the recompression S-CO_2 cycle the recompression ratio (ratio of the mass flow rate that goes to a cooler to the mass flow rate that goes to a recompressor), the pressure after the turbine and the pressure after the compressor were optimized as well. The pressure after the turbine and the pressure after the compressor were optimized for the recuperated cycle. The results of the thermodynamic parameters optimization according to Table 3 are presented in Table 4.

It should be noted that the bottoming recompression cycle produces its maximum power at a pressure after compressor of 32 MPa with a fixed recompression ratio and a 5 °C pinch in the recuperators. However, the recuperated cycle efficiency has another behaviour as it shows a slight efficiency rise even at values above

32 MPa. The obtained pressure value is slightly high and usually investigators assign a value not higher than 30 MPa [2, 3, 5] but our task in this paper is to reveal the maximum possible power of bottoming S-CO_2 cycles which utilize EFG heat of the specific GTU. Taking into account a conceptual character of the paper the value of pressure was not restricted therefore the value of 32 MPa for pressure after the compressor was accepted for the further calculations of both recompression and recuperated cycles.

Table. 4 Optimized thermodynamic parameters of S-CO_2 cycles

#	Parameter	Unit	Value	
			Recomp.	Recup.
1	Recompression ratio	-	0.683	-
2	Pressure after turbine	MPa	8.35	8.35
3	Pressure after compressor	MPa	32	32

After the standalone optimization of pressures and recompression ratio the cycles were connected in series downstream of the GTU as shown in Fig. 2. Then the optimization of the mass flow rates in each cycle was performed to get the maximum additional power possible. Firstly, the mass flow rate of the recompression cycle was optimized at fixed recuperated cycle mass flow rate. The dependency of the bottoming S-CO_2 cycles total power on the mass flow rate in the recuperated cycle is shown in Fig. 3. As it can be seen from this figure the optimum mass flow rate for the recompression cycle equals 135 kg/s. At the fixed optimum value of the mass flow rate in the recompression cycle the optimization of the mass flow rate of the recuperated cycle was performed (Fig. 4). The figure shows that the optimum mass flow rate for the recuperated cycle equals 95 kg/s. It should be pointed out that the total power of the bottoming cycles is much more sensitive to the mass flow rate in the recuperated cycle.

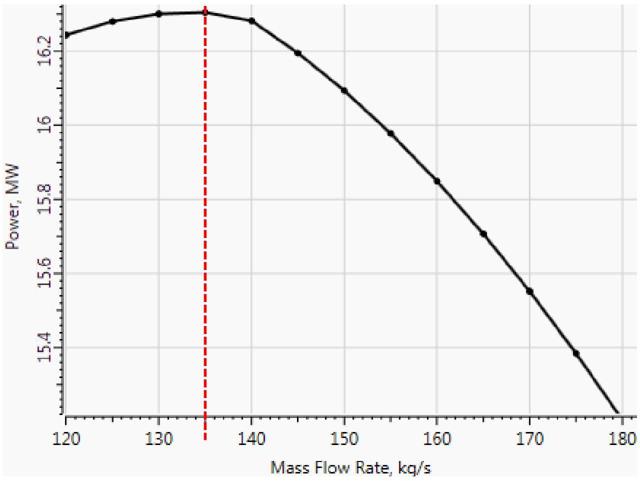


Fig. 3 Total power of the bottoming cycles vs. mass flow rate in the recompression cycle

The processes of the cycles with the optimized mass flow rates in TS coordinates are presented in Fig. 5. The purple circuit represents the recompression cycle (points 1a-2a-3a-4a-5a-6a-7a-8a-1a). It consists of the following processes: 1a-2a is the compression in the main compressor; 2a-3a is the heat recovery in the low temperature recuperator; 3a-4a is the heat recovery in the high temperature recuperator; 4a-5a is the heat reception in the first IHX; 5a-6a is the expansion in the turbine; 6a-7a is the heat return in the high temperature recuperator; 7a-8a is the heat return in the low temperature recuperator and 8a-1a is the heat sink in the cooler. In turn, the blue circuit represents the recuperated cycle (points 1b-2b-3b-4b-5b-6b-1b). Respectively, the recuperated circuit includes the following processes: 1b-2b is the compression in the compressor; 2b-3b is the heat recovery in the recuperator; 3b-4b is the heat reception in the second IHX; 4b-5b is the expansion in the turbine; 5b-6b is the heat return in the recuperator; 6b-1b is the heat sink in the cooler. The total power production of this sequence of the bottoming S-CO_2 cycles is 16.3 MW. For comparison in work [8] the best power production was 15.2 MW only. Existing bottoming steam cycles give 14 MW of power for this GTU exhaust. Our estimations of a single bottoming recuperated cycle gives 12.94 MW of power for the same conditions. The comparison presented above shows that sequential S-CO_2 cycles installation on GTU exhaust gives about 20 %, respectively, 8 % (relatively) more power than the existing steam solution or, respectively, other S-CO_2 concepts. However, this approach has an evident drawback – an increased number of additional components.

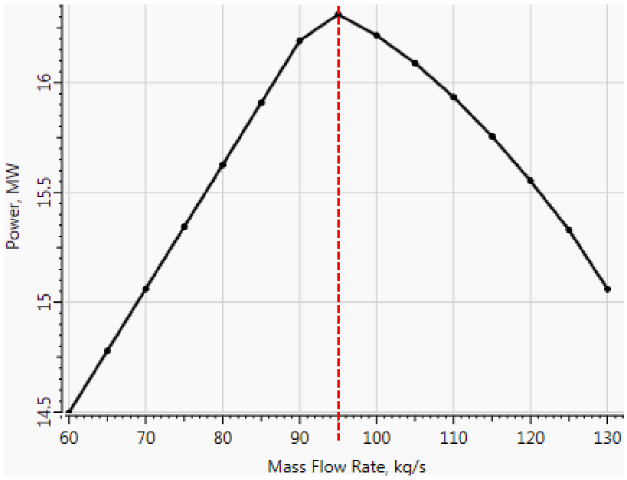


Fig. 4 Total power of the bottoming cycles vs. mass flow rate in the recuperated cycle

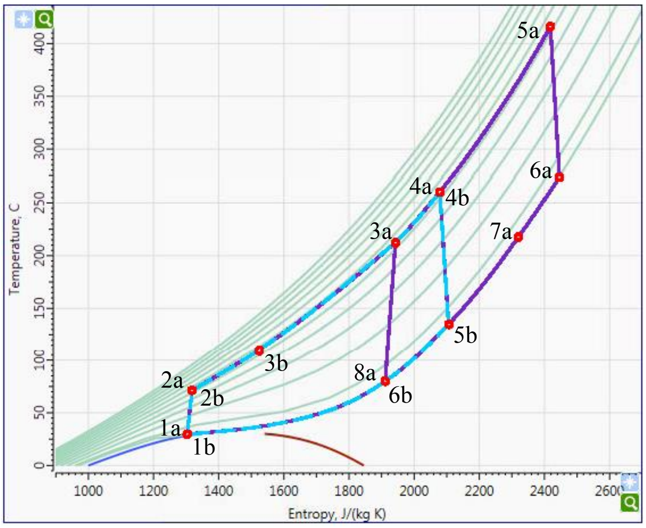


Fig. 5 Recompression and recuperated cycles processes diagram in TS coordinates

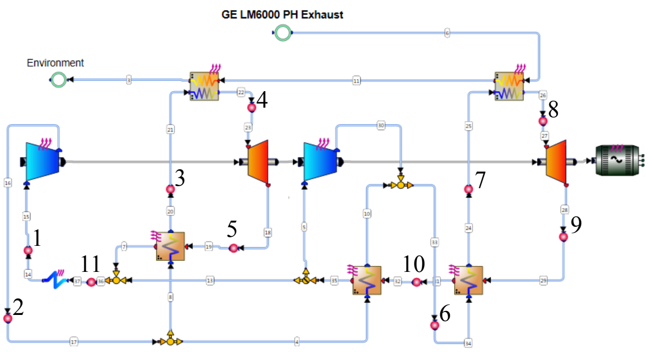


Fig. 6 Composite recompression-recuperated bottoming cycle

scheme

Fortunately, the cycles in Fig. 5 superimpose at the compression and heat sink processes. Taking into account this superimposition it seems reasonable to combine these cycles by using a single compressor and a single cooler for both cycles as shown in Fig. 6. Such a composition leads to a significant decrease in the number of components. In this particular case the generator, compressor and cooler were excluded from the bottoming system. Moreover, the composition even causes some positive effect on the bottoming cycle performance. Namely, the total power of the bottoming system was increased from 16.3 MW to 16.49 MW due to the exclusion of the second generator which had an efficiency of 0.99. The processes of this composite cycle in TS coordinates are presented in Fig. 7. The circuit in Fig. 7 is particularly the same as the circuits shown in Fig. 5, but now they are not isolated from each other but connected together by splitter and mixer at proper state points. The state points numbering corresponds to the numbering in Fig. 6. Thus, we obtained a composite cycle that has a rather high additional power production (about 31% of GTU power) with a moderate number of additional components. The net efficiency of the system GTU + bottoming composite cycles is 55.4 % and produced a total power of 69.75 MW.

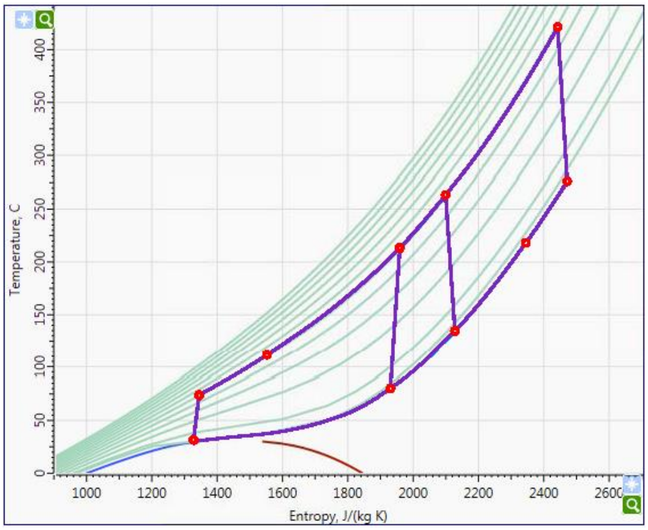


Fig. 7 Composite recompression-recuperated bottoming cycle processes in TS diagram

It should be noted that the common shaft configuration with two compressors and two turbines presented in Fig. 6 is hard to be practically implemented. The aspects of compressor and turbine matching are not considered in the scope of this study. Such a study will be performed later and might be presented in one of our further papers.

CALCULATIONS AND COMPARISON OF THE BOTTOMING CYCLES

Utilizing the approach described above several composite cycles were modelled (Fig. 8 – Fig. 10) and several simple cycles were also calculated for comparison. The mass flow rate for each considered cycle was optimized in terms of maximum power production. The conditions and components parameters were fixed and correspond to the data in tables 1, 3 and 4. However, the other parameters like the mass flow rate and temperatures for each cycle modification were respectively optimized to get the best possible power production.

In order to make references on cycles to be considered more convenient we gave to them some simple names, like Cycle 1, Cycle 2, etc. In accordance to this convention the cycle scheme shown in Fig. 2 was named Cycle 0, the scheme on Fig. 6 was referred to as Cycle 1.

Cycle 2 was made by some simplification of Cycle 1. The simplification was the exclusion of the recuperator from the recuperated part of the composite cycle (Fig. 8). The simplification led to some decrease of the bottoming cycle performance. As a result, the net power of Cycle 2 is 16.13 MW while the total power of the GTU + Cycle 2 system is 69.39 MW with a thermal efficiency of 55.1 %.

The next modification was an addition of a recuperated cycle to Cycle 2. The obtained cycle was named Cycle 3 (Fig. 9). The added recuperated cycle is marked out by red dashed rectangle in Fig. 9. The low temperature recuperator of the initial recompression cycle serves both for the recompression and recuperated cycles as well as the other components, like compressors, cooler, etc. As a result of this, only the single turbine and HEX were added to Cycle 2 to implement the additional recuperated cycle. As it can be seen, Cycle 3 includes recompression, recuperated and simple $S\text{-}CO_2$ cycles with only two compressors, single cooler and two recuperators. Only the number of turbines, HEXs and splitters were increased. The power of Cycle 3 is 16.45 MW with a thermal efficiency for the whole system of 55.36 %.

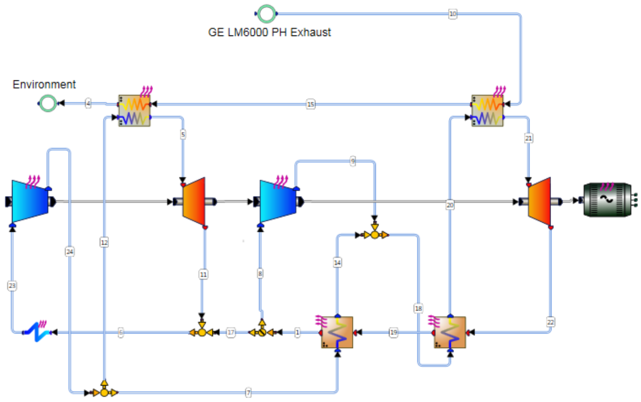


Fig. 8 Composite bottoming Cycle 2 scheme

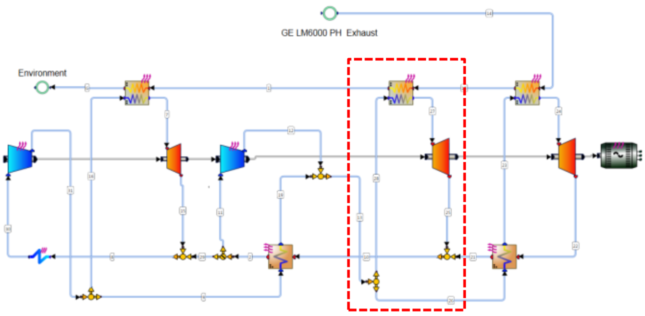


Fig. 9 Composite bottoming Cycle 3 scheme

Cycle 4 is based on Cycle 3's scheme complication. The modification is the addition of a reheat and a low pressure turbine. Additional components are marked out by a red rectangle in Fig. 10. According to this scheme, the $S\text{-}CO_2$ after first HEX at a pressure 32 MPa goes to the High Pressure (HP) turbine, expands to 14 MPa and recovers heat in the additional recuperator before being heated in the additional HEX and expands to 8.35 MPa in the Low Pressure (LP). The other part of the scheme is identical to Cycle 3. Such a modification leads to increasing the power production and obtaining 17.05 MW of additional power compared to the power of the GTU by itself. The thermal efficiency of the whole system is 55.84 %.

In order to estimate how much the reheat in the recompression cycle might give without including the recuperated and simple cycles Cycle 5 was developed (Fig. 11). The reheat part of this cycle is similar to that part in Cycle 4. However, the mass flow rates and temperatures were optimized to reach the maximum possible power for the defined conditions. The power of this cycle is 13.31

MW and the thermal efficiency of the whole system is 52.87 %.

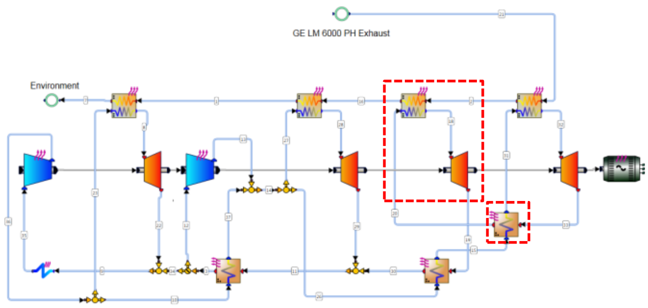


Fig. 10 Composite bottoming Cycle 4 scheme

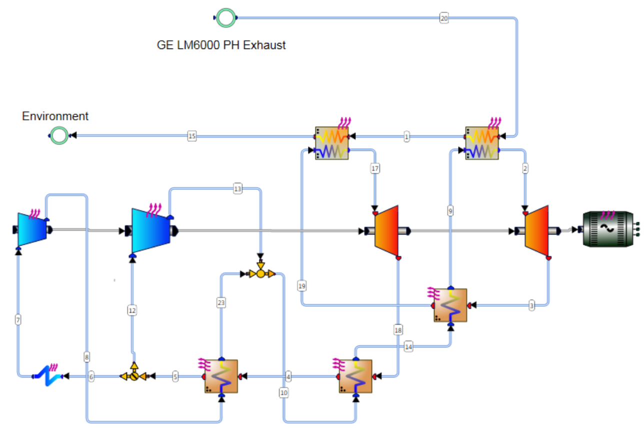


Fig. 11 Recompression bottoming cycle with reheat scheme (Cycle 5)

In order to make the comparison of the cycles more comprehensive a couple of conventional cycles, like recompression S-CO_2 and recuperated S-CO_2, were modelled and added to the list of cycles studied. The first one was designated as Cycle 6 (Fig. 12) and the second one as Cycle 7 (Fig. 13). Cycle 6 gives 11.85 MW of power. In turn, Cycle 7 gives 12.94 MW. The respective system thermal efficiency magnitudes of these cycles are 51.71 % and 52.58 %.

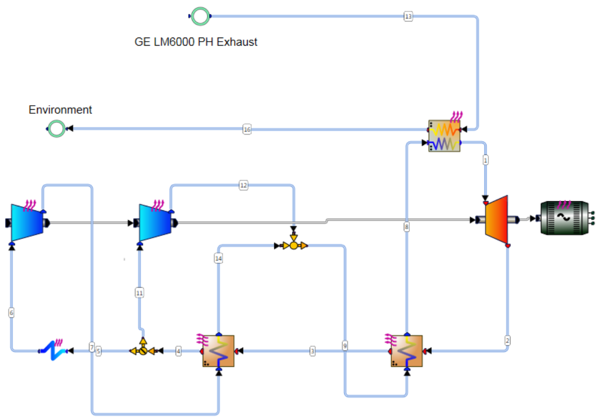


Fig. 12 Recompression S-CO_2 bottoming cycle scheme (Cycle 6)

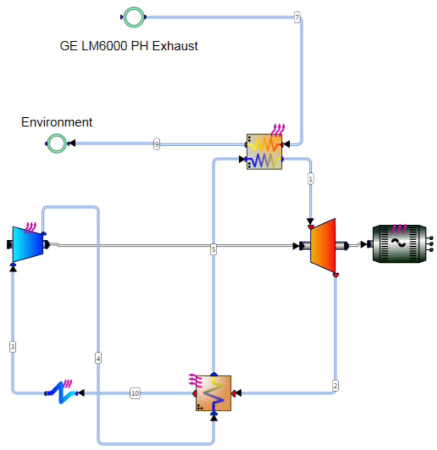


Fig. 13 Recuperated S-CO_2 bottoming cycle scheme (Cycle 7)

A comparison of the essential features of the considered cycles is shown in Fig. 14 – 17. The relative power increment of the system that can be obtained if the respective bottoming cycle is installed after the GTU is presented in Fig. 14. These values were obtained using the following formula:

$$Relative\ Power\ Increment = \frac{Bottoming\ Cycle\ Power}{GTU\ power} 100\%$$

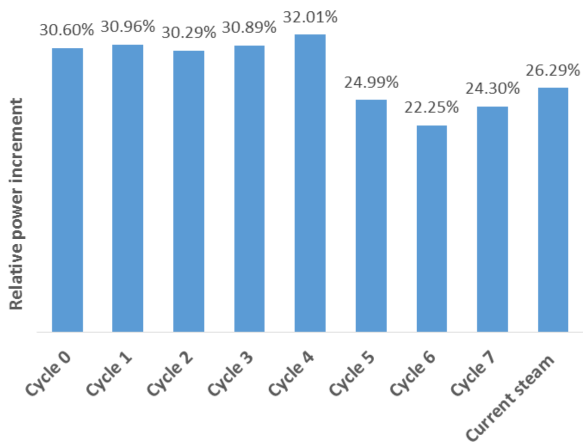


Fig. 14 Relative power increment of the system

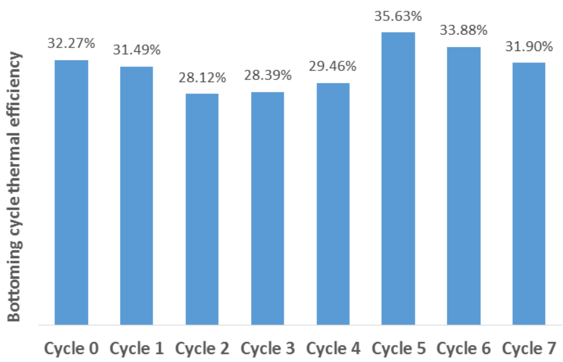


Fig. 15 Bottoming cycle thermal efficiency

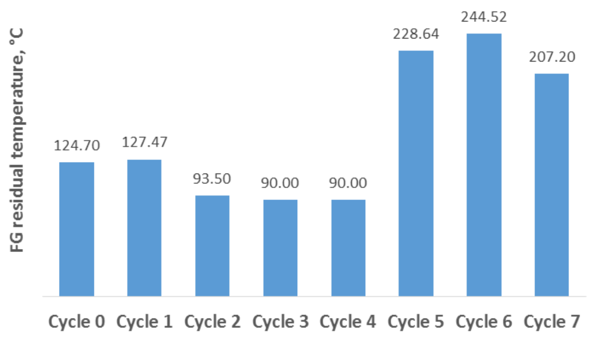


Fig. 16 FG residual temperature

It is easy to see that Cycle 4 gives the highest power increment among the cycles studied. It should be noted that the difference between Cycle 4 and the cycles from 0 to 3 is not very significant. However, comparing it with Cycle 6 (recompression S-CO_2) the difference is about 10 % (absolute) which is a much more significant difference. The steam bottoming cycles currently available produce about 6 % less power (absolute) than Cycle 4 presented in this study.

Fig. 15 shows the thermal efficiency of the bottoming cycle which was found using the following formula:

$$Thermal\ efficiency = \frac{Bottoming\ cycle\ power}{Captured\ heat\ from\ FG} 100\%$$

The bottoming cycle thermal efficiency values are rather small due to the moderate temperatures of the live CO2 and to the combination of high efficiency cycles with low efficiency ones. Curiously enough, the cycles that produce the lowest power (Cycle 5 and Cycle 6) have the highest efficiency. This happens because these cycles capture a lesser amount of FG heat and even the higher efficiency does not compensate for the heat losses. Fig. 16 shows FG residual temperature values for each considered cycle and allows to judge about lost heat for each of them. Namely, the higher residual temperature the higher amount of lost heat. The conclusion is in good agreement with the conclusions obtained in [3] and [5].

The performance data of the modelled cycles are presented in table 5. It should be noted that the bottoming steam cycle was not modelled due to a lack of some data about it. However, some performance information was known and added to the table. The table cells with unknown parameters were greyed. Cost estimation was not performed in current study, but to take, at least indirectly, into account a complexity of the bottoming cycle the number of components was included in the table and was obtained by counting the following components: cooler, HEX, recuperator, compressor, turbine and alternator. Splitters, mixers and any other auxiliaries were not accounted for.

Table. 5 Optimized thermodynamic parameters of S-CO_2 cycles

Cycle	Unit	Cycle 0	Cycle 1	Cycle 2	Cycle 3	Cycle 4	Cycle 5	Cycle 6	Cycle 7	Current steam
Bottoming cycle power	MW	16.30	16.49	16.13	16.45	17.05	13.31	11.85	12.94	14.00
Total power	MW	69.56	69.75	69.39	69.71	70.31	66.57	65.11	66.20	67.26
Relative power increment	%	30.60%	30.96%	30.29%	30.89%	32.01%	24.99%	22.25%	24.30%	26.29%
Bottoming cycle thermal efficiency	%	32.27%	31.49%	28.12%	28.39%	29.46%	35.63%	33.88%	31.90%	
Whole system thermal efficiency	%	55.25%	55.40%	55.11%	55.36%	55.84%	52.87%	51.71%	52.58%	~54.00%
FG residual temperature	°C	124.70	127.47	93.50	90.00	90.00	228.64	244.52	207.20	
Components number	pcs	14	11	10	12	15	11	8	6	

In terms of power increment Cycle 4 is the best candidate for installation at a GTU GE LM6000 PH exhaust. The magnitude of the additional power is 17.05 MW absolute, which is 32.01 % relatively to 53.26 MW of the GTU. The payback for such a performance increase is the bottoming cycle complication. Cycle 4 has 15 components total. This is a highest number from the calculated cycles presented here. It should be pointed out components number is varying significantly from cycle to cycle and is presented in Fig. 17. Matching the relative power increment (Fig. 14) and components number (Fig. 17) it can be seen that Cycle 2 produces almost the same power (16.13 MW) as Cycle 4 with significantly less components (10 vs 15).

Comparing our results with the results from [5] where authors obtained 15.2 MW of bottoming cycle power, any of the developed composite cycles produce an additional 0.93 MW to 1.85 MW of power.

It should be noted that the reasoning presented above is of an indirect nature without a cost estimation of the bottoming cycles. However, the obtained results allow concluding that it is possible to obtain more additional power utilizing S-CO_2 cycles with different levels of complexity instead of the currently used steam configurations. Final decision of some composite cycle reasonability should be performed after comprehensive economic analysis.

Also, it is necessary to mention that the proposed composite cycle approach allows accommodating S-CO_2 technology to temperature sensitive heat sources. This approach looks more vital and economically more promising in relation to sequential S-CO_2 cycles installation.

Composite cycle schemes are not limited to the considered ones and it is possible to develop a wide variety of composite cycles for different applications.

It is obvious that the composite cycle approach might be applied not only to GTU exhaust but also too any temperature sensitive heat source. This approach might help solve the problem of S-CO_2 cycle applicability to any temperature sensitive heat sources.

The reasonability of composite cycle application to high power GTU like Siemens H-class GTU or any other temperature sensitive heat source might be considered separately in one of our future papers.

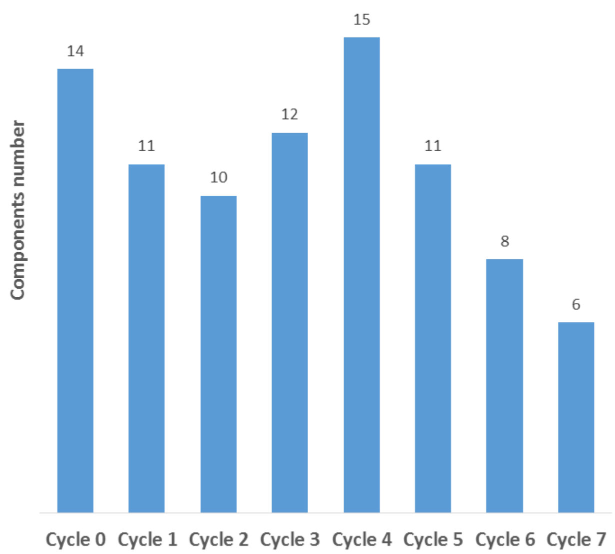


Fig. 17 Components number of considered bottoming cycles

CONCLUSIONS

- The problem of GTU GE LM6000 PH exhaust flue gas heat utilization by bottoming S-CO_2 cycles from the perspective of maximum possible power increment was considered.
- Sequential recompression and recuperated S-CO_2 cycles instal-

lation at the GTU exhaust was estimated. Estimations showed that it is possible to produce 16.30 MW in addition to the 53.26 MW of GTU power.

- If the sequentially installed cycles have the same pressure after the turbine and same the pressure after the compressor with the same heat sink, the thermodynamic processes in a compressor, a cooler and a low temperature recuperator from these cycles might be superimposed. Taking this into account the composite cycle approach was proposed. According to this approach different sequentially installed S-CO_2 cycles might be merged to have a common compressor, cooler, recuperator, etc. It makes possible to develop brand new composite cycles based on well-known relatively simple S-CO_2 cycles, such as recompression, recuperated, simple Brayton cycles and any other combinations. The main advantage of the composite S-CO_2 cycle approach is a significant reduction of the number of components when several relatively simple cycles are installed sequentially after temperature sensitive heat source.
- Several composite S-CO_2 cycles were developed and compared with sequential recompression and recuperated S-CO_2 cycles installation case, with standalone recompression, standalone recuperated and current steam bottoming cycle. A total of seven S-CO_2 cycles were calculated and compared with each other and with the current steam cycle standard. As a result, the composite cycle (Cycle 4) which contains a recompression cycle with reheat, a recuperated cycle and a simple Brayton cycle has the highest power from all considered cycles. The power is 17.05 MW, which is 32.01 % relatively to 53.26 MW of the GTU. It is 3.05 MW higher than what the current steam bottoming cycle produces. However, taking into account the number of components of the considered cycles this cycle might not be as attractive. For example, the composite cycle which consists of a recompression cycle and a simple Brayton (Cycle 2) produces almost the same power (16.13 MW) as Cycle 4 with a significantly lesser number of components (10 vs 15). Final decision about some specific composite cycle reasonability should be performed after comprehensive economic analysis.
- The proposed composite cycle approach allows accommodating S-CO_2 technology to temperature sensitive heat sources. This approach looks more vital and economically more promising relatively to sequential S-CO_2 cycles installation.
- The composite cycle approach might be applied not only to GTU exhaust but to any temperature sensitive heat source. This approach might help solve the problem of S-CO_2 cycle applicability to any temperature sensitive heat sources.
- Composite cycle schemes are not limited to the considered ones and it is possible to develop a wide variety of composite cycles for different applications.

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