

## PowerEnergy2015-49439

### EVALUATION FOR SCALABILITY OF A COMBINED CYCLE USING GAS AND BOTTOMING SCO<sub>2</sub> TURBINES

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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, power generation boost without burning more fuel and even production of heat for cogeneration applications are very attractive and it becomes necessary to quantify how much can really be extracted from a simple cycle to be converted to a combined configuration.

As supercritical CO<sub>2</sub> is becoming an emerging working fluid [2, 3, 5, 7 and 8] due not only to the fact that turbomachines are being designed significantly more compact, but also because of the fluid's high thermal efficiency in cycles, it raises an increased interest in its various applications. Evaluating the option of combined gas and supercritical CO<sub>2</sub> cycles for different gas turbine sizes, gas turbine exhaust gas temperatures and configurations of bottoming cycle type becomes an essential step toward creating guidelines for the question, "how much more can I get with what I have?".

Using conceptual design tools for the cycle system generates fast and reliable results to draw this type of conclusion. This paper presents both the qualitative and quantitative advantages of combined cycles for scalability using machines ranging from small to several hundred MW gas turbines to determine which configurations of S-CO<sub>2</sub> bottoming cycles are best for pure electricity production.

#### 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 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]. But due to the features of steam cycles (huge 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].

Thus the search for alternatives to the bottoming steam cycle for existing and brand new gas turbines of different sizes, and exhaust gases temperatures is a very real task. One of the very promising way for this purpose is the consideration of Supercritical CO<sub>2</sub> (S-CO<sub>2</sub>) cycles based on closed-loop Brayton cycle and its modifications:

- Carbon dioxide is an ideal working fluid for closed-loop cycles. It is a low-cost, non-toxic, non-flammable, non-corrosive, readily available, stable fluid and there are a lot of available test data in a wide range of parameters.

- The high density of S-CO<sub>2</sub> enables the creation of a relatively compact power cycle.
- Supercritical CO<sub>2</sub> cycles have efficiency levels which are higher than those of a Helium Brayton system and are comparable with steam cycles. At higher temperatures, S-CO<sub>2</sub> cycle efficiency may exceed that of a steam cycle, but steam cycles tend to have the higher efficiency when initial cycle temperature is below approximately 450°C [3].

The bottoming S-CO<sub>2</sub> cycle makes it 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<sub>2</sub> cycles have a different ability of heat utilization and its conversion into electricity. Also, a S-CO<sub>2</sub> cycle may have a quite 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 efficient bottoming S-CO<sub>2</sub> cycle may give less of an electricity production addition to the whole system than one with moderate efficiency but with higher S-CO<sub>2</sub> temperature difference at the heater.

Reasonability of bottoming cycle application after GTU is considered in such papers as [2, 8]. It is particularly stated in [2] that S-CO<sub>2</sub> 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<sub>2</sub> reduces system component size and cost, and provides significant advantages regarding system efficiency, footprint and ease of installation. Different configurations of bottoming S-CO<sub>2</sub> 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 bottoming S-CO<sub>2</sub> cycle is not reasonable for H Class GTU, but reasonable for GE LM 6000. Based on this, he concluded that all considered cycle configurations are suitable for low power, low GTUs exhaust temperatures.

This paper considers three configurations of bottoming S-CO<sub>2</sub> cycles for exhaust heat recovery from existing GTU, which include the simple recuperated cycle, the pre-compression cycle, and recompression cycle [3]. The goals of the study are to define an optimal cycle configuration, optimal thermodynamic parameters of S-CO<sub>2</sub> cycle, provide a means for estimating bottoming cycle power output for a wide range of existing GTU and present suggestions for scaling/standardization of bottoming S-CO<sub>2</sub> cycles.

The study of combined and bottoming S-CO<sub>2</sub> cycles were performed with the use of the heat balance calculation tool AxCYCLE<sup>TM</sup> [4, 5]. S-CO<sub>2</sub> properties correspond to NIST data [6].

## DETERMINATION OF OPTIMAL BOTTOMING CYCLE CONFIGURATION

Three different S-CO<sub>2</sub> cycles were analyzed to determine the best bottoming cycle configuration for a wide range of GTU. At the first stage of study, the heat source for all cycles was simulated as a conventional gas turbine engine with equal exhaust parameters – exhaust gas temperature 550°C and exhaust gas mass flow rate as 100 kg/s. Specified flue gas parameters correspond to a middle-range class of GTU because it is expected that this GTU will be priority for equipment with alternative bottoming cycles.

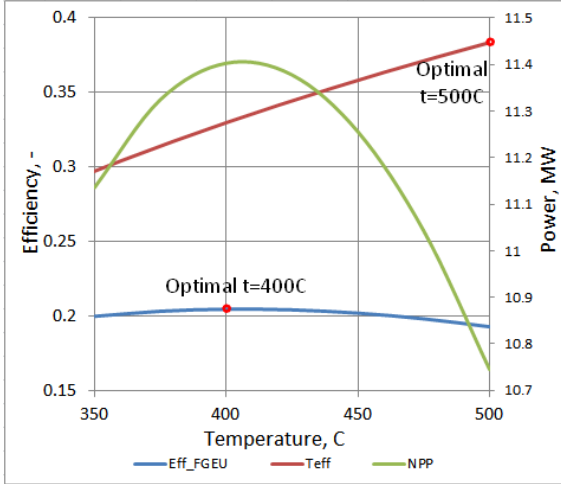
Within the scope of the current study, the question of finding the correct quality criterion for the bottoming cycle estimation was raised. Usually researchers don't pay sufficient attention to this question and use either bottoming cycle efficiency and output or overall performance of a combined cycle. It is quite possible that such an approach is correct for some cases, but such a solution is inadmissible when considering a universal bottoming cycle and its optimal parameters for a wide range of GTU. For example, in overall combined cycle performances the GTU characteristics (its efficiency or heat rate, exhaust gas temperature, and others) have a great influence and a particular bottoming cycle will be a good choice for that specific GTU and a poor choice for another unit. The use of bottoming cycle output as a quality factor in the current task is inconvenient because it is closely related to upper GTU too. However, it is a good quality factor for a task with fixed heat source conditions. Similar conclusion was also obtained in [8]. The choice of bottoming cycle thermal efficiency could be reasonable for fixed or circulating heat sources like a nuclear reactor, where heat that was not transferred to CO<sub>2</sub> cycle is not a loss [3]. For the current study, the **effectiveness of flue gas thermal energy use** was proposed as quality criterion even though other approaches like DiPippo's exist. The effectiveness considered here is defined by the following equation:

$$Eff_{FGEU} = \frac{NPP}{E_{FG}} = \frac{NPP}{M_{FG}(h_{FG1} - h_{FG2})},$$

where  $E_{FG}$  = heat energy of flue gas that theoretically could be used on the assumption of gas cooling to an ambient temperature;  $NPP$  = net power production of bottoming cycle;  $M_{FG}$  = flue gas mass flow;  $h_{FG1}$ ,  $h_{FG2}$  = flue gas enthalpy at GTU outlet and flue gas enthalpy at ambient temperature (20°C) and pressure, respectively.

Advantages of the proposed assessment criterion versus bottoming cycle thermal efficiency are presented in Figure 1. Charts were created for the following conditions:

- Flue gas mass flow 100 kg/s, temperature 550°C;
- CO<sub>2</sub> cycle type – simple recuperated (Figure 2), initial CO<sub>2</sub> temperature varies from 350 to 500°C.



**Figure 1: Comparison of quality criteria**

As is evident from Figure 1, different criteria give different optimal values of initial CO<sub>2</sub> temperature and  $Eff_{FGEU}$  allows more power from flue gas. The thermal efficiency of the CO<sub>2</sub> cycle therefore shows the effectiveness of energy transformation inside the cycle and doesn't account for the final flue gas temperature (after heat transfer to the bottoming cycle). For the current example, the bottoming cycle operation with a lower initial temperature and with lower thermal efficiency is more attractive because it creates a more complete recovery of exhaust heat. This conclusion is correct for recuperated CO<sub>2</sub> cycle configuration but it will change for other cycle embodiments.

The first embodiment is an assembly of GTU with a closed recuperated S-CO<sub>2</sub> cycle. As is evident from Figure 2, the bottoming S-CO<sub>2</sub> cycle includes a compressor (1-2), a CO<sub>2</sub>/Flue Gas intermediate heat exchanger (IHX) (3-4), a turbine (4-5), a recuperator (5-6) and a CO<sub>2</sub> cooler (6-1). The t-s diagram of the recuperated S-CO<sub>2</sub> cycle is shown in Figure 3.

The recuperator is used in the cycle to preheat the working fluid (process 2-3) before the main CO<sub>2</sub> heater (process 2-3). It preserves part of the heat (between point 5 and 6 in the process diagram) inside the cycle and decreases the heat load on the CO<sub>2</sub> cooler. For the correct recuperator simulation in its internal model, the pinch approach was used to avoid negative temperature differences between hot and cold CO<sub>2</sub> flows inside the recuperator.

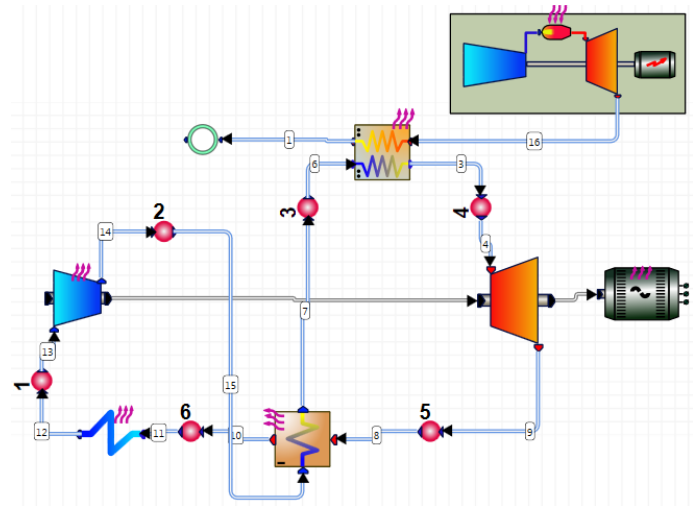
Design parameters of S-CO<sub>2</sub> cycle components and main cycle parameters are given in Table 1. In order to compare different bottoming cycles correctly these parameters were maintained constant for all the considered cycle configurations. Flue Gas/CO<sub>2</sub> heat exchanger (HEX) efficiency was determined using the following formula:

$$\eta_{HEX} = \frac{T_{fg1} - T_{fg2}}{T_{fg1} - T_{co21}}$$

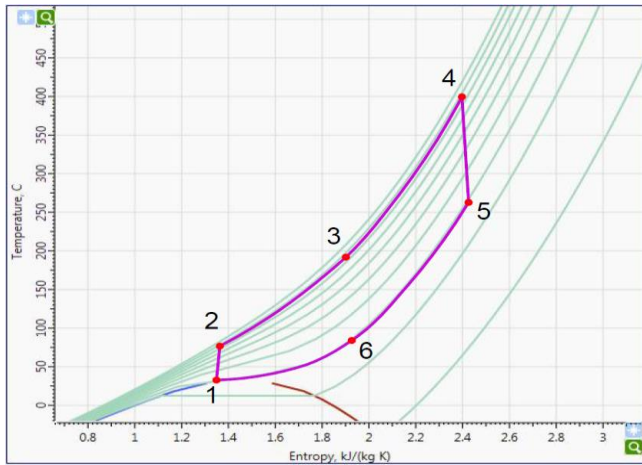
where  $T_{fg1}$  - flue gas temperature before HEX,  $T_{fg2}$  - flue gas temperature after IHX,  $T_{co21}$  - temperature of CO<sub>2</sub> before IHX.

Pressure magnitudes after S-CO<sub>2</sub> compressor and after S-CO<sub>2</sub> turbine were not the object of this study as they were determined from [3, 7]. According to [3], the S-CO<sub>2</sub> cycle has maximum efficiency at a top pressure of 30 MPa. Higher pressure magnitudes are not reasonable because they give less growth in efficiency but significantly increase structural requirements. The pressure after the turbine was determined according to the results of [7]. According to this study, the optimum pressure ratio for recompression cycle lays between 3.6 and 4.2. We decided to accept 3.75 as the chosen value. As a result, the pressure magnitude after the compressor was accepted as 30 MPa and the pressure after the turbine was calculated to be 8 MPa.

Different GTU have different temperatures and mass flow rates of exhaust gasses. The exhaust gas mass flow defines the CO<sub>2</sub> mass flow in the same way as the exhaust gas temperature bounds the maximal temperature in the bottoming cycle, at the defined heat exchanger efficiency, that has an essential influence on cycle performance. In order to draw a conclusion about the suitability of a recuperated CO<sub>2</sub> cycle for use as a bottoming cycle with a wide range of GTU, the series of calculations for different exhaust gas parameters was performed.



**Figure 2: GTU with bottoming recuperated S-CO<sub>2</sub> cycle**

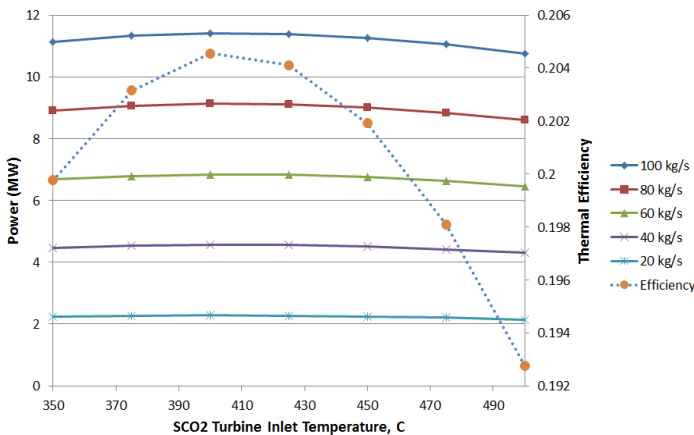


**Figure 3: T-s diagram for the recuperated S-CO<sub>2</sub> cycle – The bottom dome represents the saturation curves of liquid (blue, on the left) and vapor (red, on the right)**

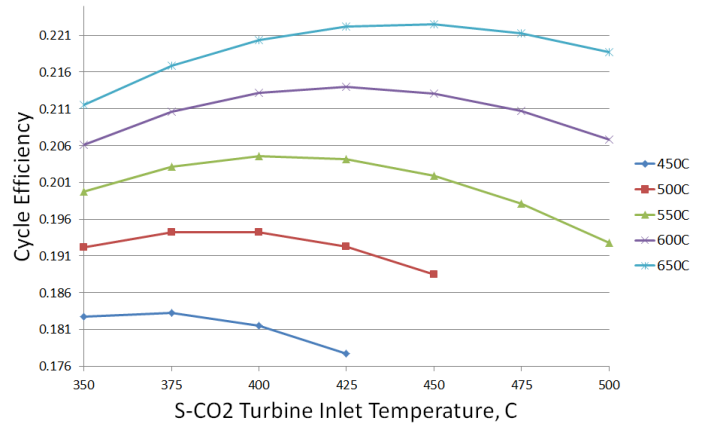
Calculated performances of a recuperated S-CO<sub>2</sub> bottoming cycle for different heat source conditions (flue gas mass flows and temperatures) are presented in Figure 4 and Figure 5.

**Table 1: Design parameters of S-CO<sub>2</sub> cycle components and main cycle parameters**

#	Parameter	Units	Value
1	Turbine efficiency	-	0.9
2	Compressor efficiency	-	0.85
3	Flue Gas/CO <sub>2</sub> HEX efficiency	-	0.9
4	Pinch for CO <sub>2</sub> /CO <sub>2</sub> recuperator	°C	5
5	Generator efficiency	-	0.99
6	Temperature at cooler outlet	°C	33
7	Maximal pressure	MPa	30
8	Pressure at turbine outlet	MPa	8



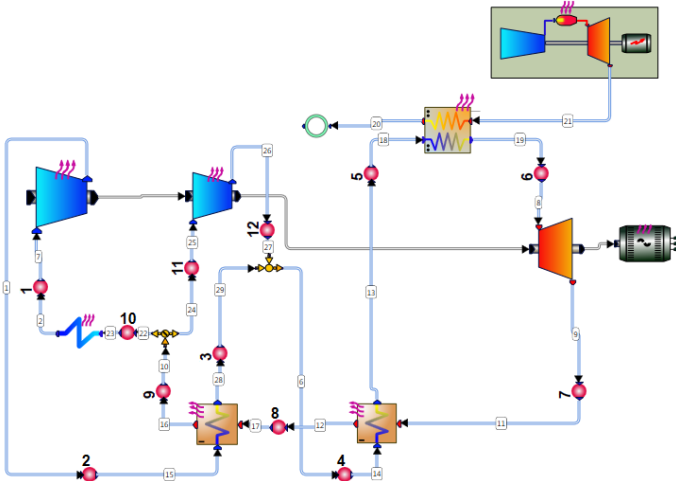
**Figure 4: Performances of recuperated S-CO<sub>2</sub> cycle (Thermal efficiency on the right Y-axis and net power production (NPP) on the left Y-axis) for different flue gas (FG) mass flows at several fixed CO<sub>2</sub> turbine inlet temperatures (X-axis, in Celsius) at fixed FG temperature**



**Figure 5: Performances of recuperated S-CO<sub>2</sub> cycle (Thermal efficiency on the Y-axis in %) for different flue gas temperatures (isolines shown) at several CO<sub>2</sub> turbine inlet temperatures (X-axis)**

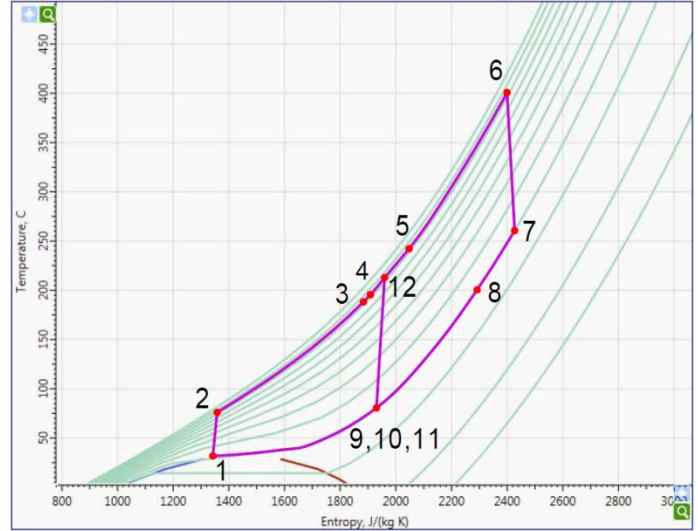
Figure 4 illustrates the influence of flue gas mass flow rate and the CO<sub>2</sub> temperature after the HRU (Heat Recovery Unit) on the bottoming cycle output and the effectiveness of flue gas thermal energy use. The gas temperature is fixed and equal to 550°C. As is evident from Figure 4, power curves for different gas mass flows are equidistant, meaning flue gas mass flow rate is only a scaling factor according to which each additional 20 kg/s of flue gas gives about 2 MW increase of bottoming cycle power. Also, it is clear that the optimal value of the initial CO<sub>2</sub> temperature exists inside of the used temperature range from 350 to 500°C. For the current case with a flue gas temperature of 550°C, the optimal initial temperature of the bottoming cycle is about 400°C. It should be noted that each flue gas mass flow rate curve has its maximum for initial temperatures of the bottoming cycle around 400°C with a slight increase as the FG temperature rises. This proves the correctness of the accepted efficiency criterion (Eff\_FGEU). In Figure 5, each curve represents the influence of CO<sub>2</sub> cycle performance from CO<sub>2</sub> turbine inlet temperature for specified value of flue gas temperature. The map covers a wide range of GTU with exhaust temperatures from 450 to 650°C. This map is correct for any flue gas mass flow rate. The effectiveness of flue gas thermal energy use in the bottoming recuperated CO<sub>2</sub> cycle is directly proportional to the flue gas temperature. As we would expect, a greater percentage of energy can be extracted by a recuperated bottoming cycle from a higher temperature exhaust.

The second examined embodiment is GTU coupled with a bottoming recompression S-CO<sub>2</sub> cycle. This cycle configuration was examined because according to [3] the recompression cycle is the most promising CO<sub>2</sub> cycle with the highest efficiency, while still retaining simplicity. The corresponding combined cycle is shown in Figure 6.



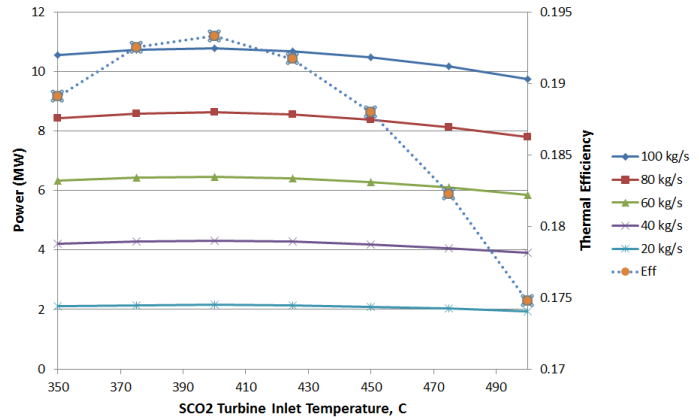
**Figure 6: GTU with bottoming recompression S-CO<sub>2</sub> cycle**

Unlike the simple recuperated S-CO<sub>2</sub> cycle, the recompression cycle includes two compressors (main compressor (1-2) and recompression compressor (11-12)), two recuperators (7-8) and (8-9), flow splitter (it splits the flow in point 9 into two flows) and mixer (it joins flows from the main and recompression compressors, point 4 has parameters of the mixed flow). In our simulation, the flow was divided between compressors, with 70% of the CO<sub>2</sub> compressed by the main compressor and the rest of the flow by the recompression compressor). This value was chosen taking into account data from [7]. According to [7] the ratio of the main compressor mass flow rate to the one of the turbine determines the parameters at the splitter (points 9, 10, 11) for fixed pressures after compressors and after turbine as well as a fixed temperature of CO<sub>2</sub> after cooler (point 1). Lower values for this ratio lead to higher cycle efficiency although the former should be limited to reach reasonable thermodynamic conditions at the second stage recuperator. Namely, at 5 °C pinch this ratio is equal to around 0.7 (70 %). The other design parameters of the cycle components and main cycle parameters are maintained the same as in the simple recuperated S-CO<sub>2</sub> cycle, as given in Table 1. The cycle process in T-S coordinates is presented in Figure 7.



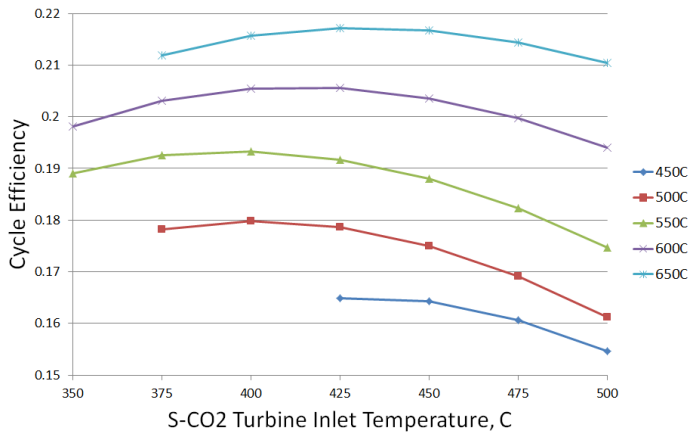
**Figure 7: T-S diagram for recompression S-CO<sub>2</sub> cycle**

Calculated characteristics for the recompression S-CO<sub>2</sub> cycle as bottoming coupled with GTU for different conditions of exhaust gas are given in Figure 8 and Figure 9. Charts in specified figures were generated for the same conditions as similar charts for recuperated cycle (for description see comments for Figure 4 and Figure 5).



**Figure 8: Performances of recompression S-CO<sub>2</sub> cycle (Thermal efficiency on the right Y-axis and NPP on the left Y-axis) at fixed flue gas (FG) temperature and different FG mass flows (isolines shown) at several CO<sub>2</sub> turbine inlet temperatures (X-axis)**

Compared to the recuperated S-CO<sub>2</sub> cycle, the recompression cycle has a narrower range of initial CO<sub>2</sub> temperatures. The minimal CO<sub>2</sub> temperature for the current minimal and maximum cycle pressures is slightly lower than 350°C. This is related to the operation of the high temperature recuperator. A decrease of initial cycle pressure or an increase of pressure at turbine outlet allows the extension of the cycle's operational range for low initial CO<sub>2</sub> temperatures.



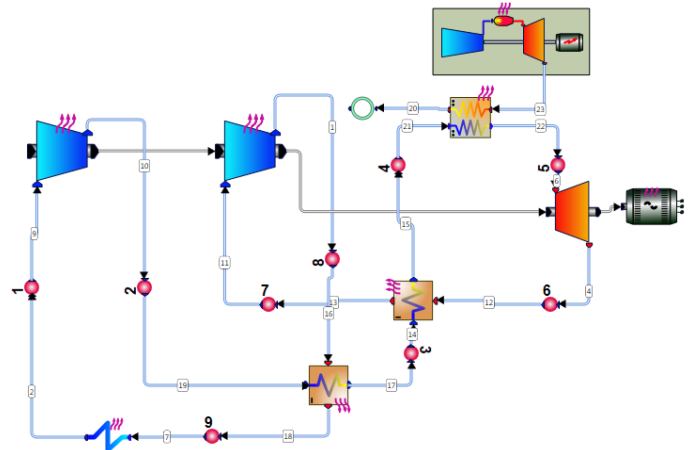
**Figure 9: Thermal efficiency (on the Y-axis) of the recompression S-CO<sub>2</sub> cycle for different flue gas temperatures (isolines) at several CO<sub>2</sub> turbine inlet temperatures (X-axis)**

The third examined embodiment is an assembly of GTU with a pre-compression S-CO<sub>2</sub> cycle. The corresponding cycle is shown in Figure 10 and its process in a t-s diagram is given in Figure 11. The pre-compression S-CO<sub>2</sub> cycle includes two compressors (low pressure compressor (7-8) and high pressure compressor (1-2)), IHX (4-5), a turbine (5-6), two recuperators (high temperature (6-7) and low temperature (8-9)) and a CO<sub>2</sub> cooler (9-1). Unlike the recompression S-CO<sub>2</sub> cycle, in the considered pre-compression cycle the flow does not divide into two flows, compressors (1-2) and (7-8), operate in different pressure ranges, and hot CO<sub>2</sub> flow passes through a low temperature recuperator (8-9) after compression in low pressure compressor (7-8). CO<sub>2</sub> pressure at the low pressure compressor outlet is 10 MPa. Other design parameters of cycle components and main cycle parameters correspond to data from Table 1. Calculated performances of a recuperated S-CO<sub>2</sub> bottoming cycle for different heat source conditions (flue gas mass flows and temperatures) are presented in Figure 4 and Figure 5.

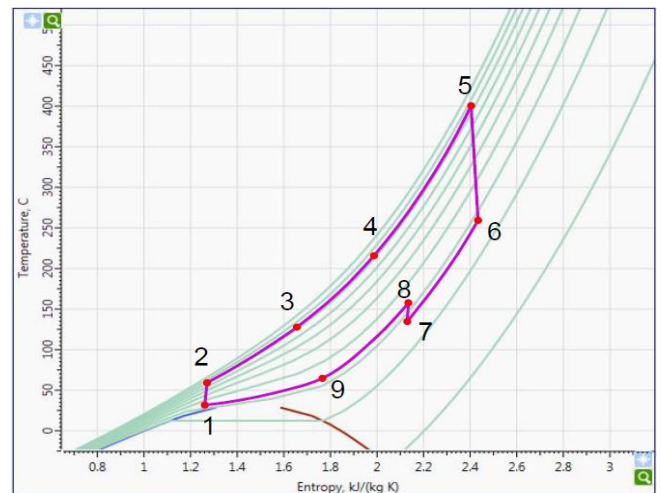
Table 1.

The pre-compression S-CO<sub>2</sub> cycle was researched because its scheme is simple enough and it has a lower working fluid temperature at the main CO<sub>2</sub> heater inlet (point 4 in Figure 11) than the recompression cycle. Due to lower CO<sub>2</sub> temperature at the heater inlet, the exhaust gas from the upper GTU can be cooled deeper and the amount of heat transferred to bottoming cycle heat will be greater.

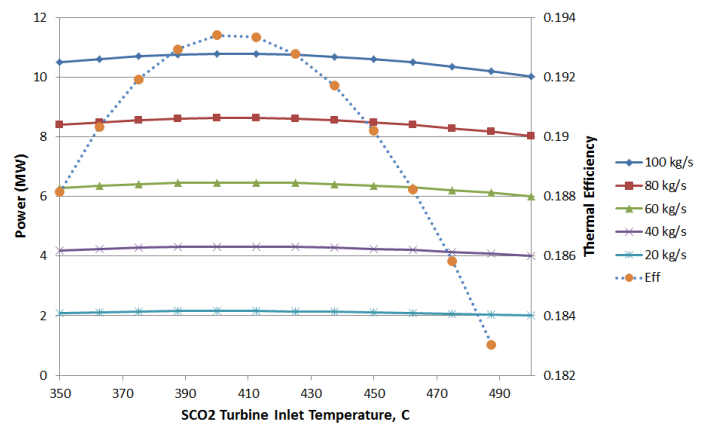
Calculated performances of the pre-compression bottoming cycle for different heat source conditions are given in Figure 12 and Figure 13. Charts in specified figures were generated for the same conditions as similar charts for previous cycles.



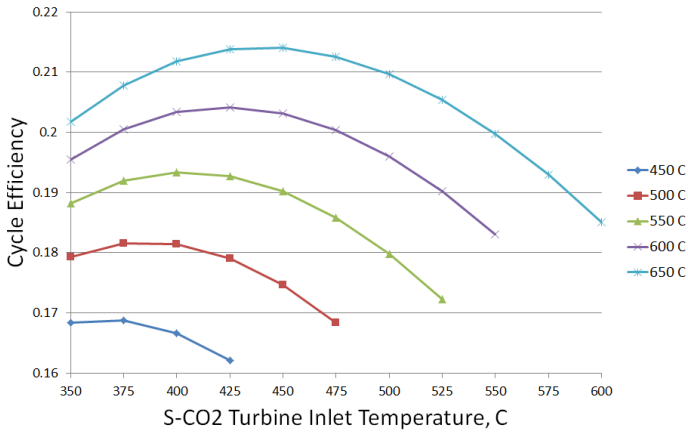
**Figure 10: GTU with bottoming pre-compression S-CO<sub>2</sub> cycle**



**Figure 11: T-S diagram for pre-compression S-CO<sub>2</sub> cycle**



**Figure 12: Thermal efficiency of pre-compression S-CO<sub>2</sub> cycle (on the right Y-axis and NPP on the left Y-axis) at fixed flue gas temperatures, for different mass flow (isolines) at several CO<sub>2</sub> turbine inlet temperatures (X-axis)**



**Figure 13: Thermal efficiency of the pre-compression S-CO<sub>2</sub> cycle (on the Y-axis) for several flue gas temperatures (isolines) at several CO<sub>2</sub> turbine inlet temperatures (X-axis)**

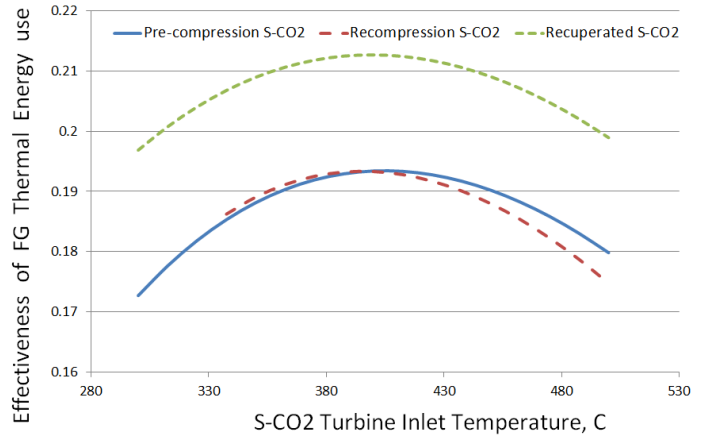
### CYCLES COMPARISON

Results of the performed study show that the simple recuperated S-CO<sub>2</sub> cycle is the most promising configuration of a bottoming cycle for power generation with GTU. This conclusion sounds strange enough because it is well known that recompression and pre-compression CO<sub>2</sub> cycles generally have higher efficiency levels than simple recuperated cycles, especially with high initial CO<sub>2</sub> temperatures. However, in our study we accounted not only for the efficiency of energy conversion inside the bottoming cycle but also for the maximum energy absorption from the gas turbine exhaust. A similar conclusion was reached in [8]. The recuperated cycle has the lowest values of CO<sub>2</sub> temperature at the inlet of IHX (point 3 in Figure 3) so it can take more energy from gasses. To make the results clearer we presented some characteristics for all the considered cycles in Figure 14 and Figure 15.

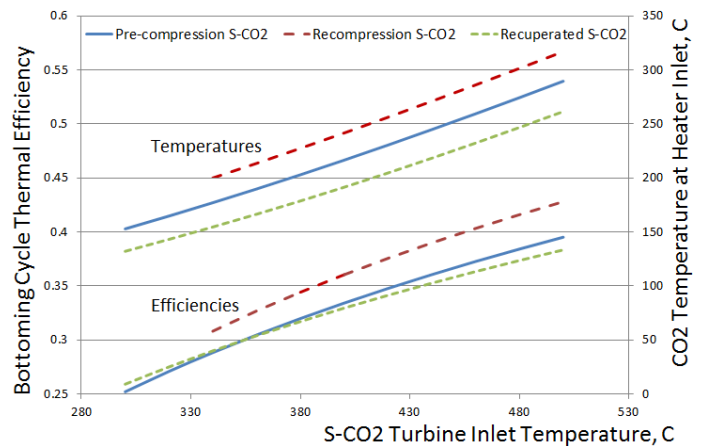
In Figure 14 the dependencies of effectiveness of flue gas thermal energy recovery from the initial temperature of CO<sub>2</sub> cycles are presented. Chart data was obtained for fixed GTU conditions (flue gas mass flow rate was 100kg/s and temperature = 550°C). The charts in Figure 14 show that power output of the recuperated cycle is about 10% higher than outputs of recompression and pre-compression cycles in the range of initial CO<sub>2</sub> temperatures.

Figure 15 illustrates the dependencies of S-CO<sub>2</sub> cycle thermodynamic efficiencies and CO<sub>2</sub> temperatures at main heater inlet on the initial CO<sub>2</sub> temperatures. The recompression cycle has the highest thermodynamic efficiency at any CO<sub>2</sub> temperature. The pre-compression cycle in terms of thermodynamic efficiency slightly beats the recuperated cycle when initial CO<sub>2</sub> temperatures exceed 350°C, but higher thermodynamic efficiency does not cover low heat absorption from the flue gas.

In addition, the recuperated cycle has the simplest configuration and the minimum set of equipment of the considered cycles.



**Figure 14: Effectiveness of flue gas thermal energy recovery for the different cycles studied for different CO<sub>2</sub> turbine inlet temperatures**



**Figure 15: S-CO<sub>2</sub> cycles thermodynamic efficiency (left Y-axis) and CO<sub>2</sub> temperatures before main heater (right Y-axis) for different CO<sub>2</sub> turbine inlet temperatures (X-axis)**

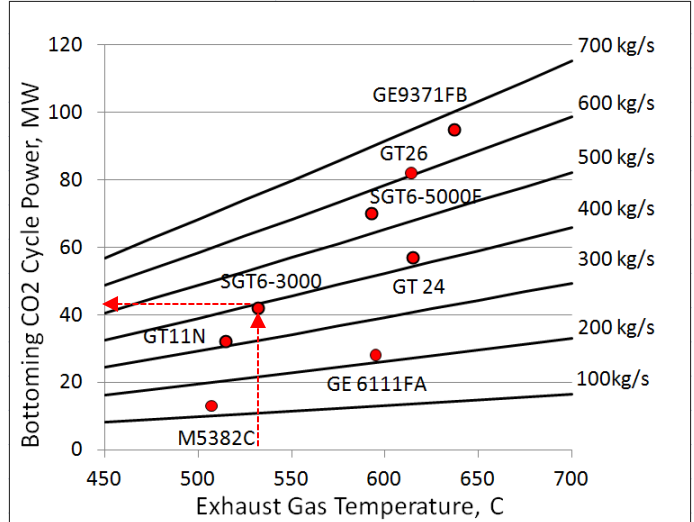
### POWER GENERATION BOOST OF EXISTING GTU

As a result of the performed study, it was concluded that the simple recuperated S-CO<sub>2</sub> cycle is the most promising bottoming cycle of the considered configurations. Therefore all of the following results refer to a combined cycle embodiment that includes GTU and simple bottoming recuperated S-CO<sub>2</sub>.

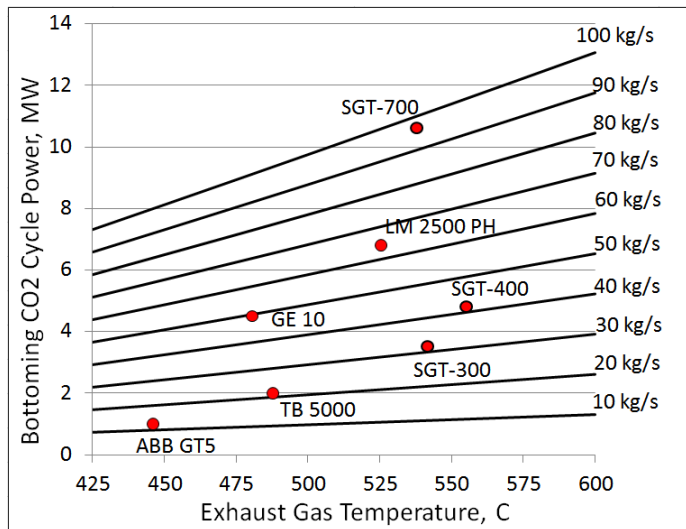
One of the goals of this work is to answer the question: "How much more can I get with what I have?". In the current chapter of this article we try to answer this question.

As has been made clear, the amount of power generated by a bottoming cycle depends on GTU exhaust gas mass flow rate, its temperature, the bottoming cycle configuration, and its parameters. The bottoming cycle has been fixed so its power is defined by flue gas parameters. On the basis of a series of thermodynamic calculations for different heat source conditions (GTU exhaust gas), maps for bottoming S-CO<sub>2</sub> cycle power were created. Specified maps are presented in Figure 16 and Figure 17. The pre-turbine CO<sub>2</sub> temperature in the recuperated

cycle was equal to 400°C. These maps can be applied to gas turbines of small and medium size with power up to 120 MW, flue gas mass flow rates from 10 to 700 kg/s, and flue gas temperatures from 425 to 700°C. We expect that GTU with corresponding flue gas parameters will be a priority for equipment with bottoming cycles. Points that correspond to the flue gas parameters of some existing GTU produced by leading manufacturers are marked on Figure 16 and Figure 17. For example, the SGT6-3000 unit has a flue gas mass flow of 382 kg/s and flue gas temperature of 532°C (see Figure 17). The coupling of this unit with the recuperated bottoming S-CO<sub>2</sub> cycle will give about 42 MW of additional power generation without burning more fuel. The proposed universal maps allow the calculation of the power production gain achieved by adding the bottoming cycle to the existing GTU (only mass flow and temperature of flue gas at GTU outlet are needed for this purpose).



**Figure 17: Bottoming S-CO<sub>2</sub> cycle power generation potential for some GTUs with high FG mass flow rate**

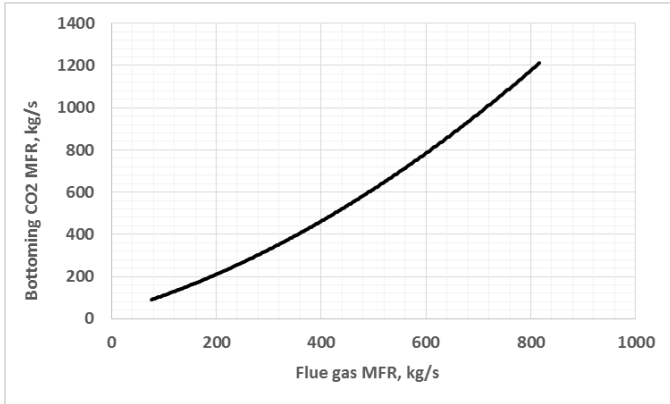


**Figure 16: Bottoming S-CO<sub>2</sub> cycle power generation potential for some GTUs with low FG mass flow rate**

**BOTTOMING CYCLE STANDARDIZATION**

The analysis of the efficiency curve behavior in Figure 5 indicates that the optimal bottoming cycle temperature at any flue gas temperature ranging from 450 to 650°C is around 400°C. Moreover, the bottoming CO<sub>2</sub> cycle power variation is not significant with a temperature deviation of about 400°C or above. This allows the creation of a single unified bottoming CO<sub>2</sub> cycle with a temperature of approximately 400°C for any flue gas temperature.

It should be noted that the proposed standardization is done at the thermodynamic level. This means that the proposed unified bottoming cycle will have the same turbine inlet temperature and pressure of CO<sub>2</sub>, as well as other thermodynamic parameters. However, different gas turbines not only have different flue gas temperatures but also different mass flow rates. The last facet prevents the creation of unified cycle components and equipment because they should be specifically designed for each specific flue gas mass flow rate. Figure 18 shows the mass flow rate (MFR) of the bottoming cycle at different values of flue gas MFR in a range from 77 to 816 kg/s. Such a MFR range corresponds to a power range from 22 to 380 MW of a gas turbine. The presented curve was obtained at a thermodynamically unified cycle with variation of MFR.



**Figure 18: Unified bottoming cycle CO<sub>2</sub> mass flow rate vs. flue gas mass flow rate for fixed CO<sub>2</sub> turbine inlet temperature of around 400C**

Naturally, the implementation of the unified cycle allows for an increase in net power production as well as efficiency.

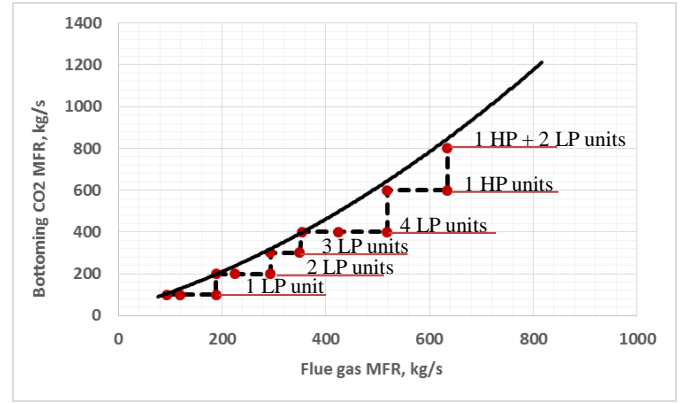
It is obvious that it is much more convenient to have some fully standardized components. Such an approach requires some quantitatively and thermodynamically standardized bottoming unit(s) which will help utilize any gas turbine exhaust heat. The estimation of efficiency and power increase with the use of fully standardized bottoming power units is presented below.

For the first attempt we assumed to have only two standardized bottoming units: 1) Low power (LP) unit with CO<sub>2</sub> MFR at about 100 kg/s (power around 10 MW); 2) High Power (HP) unit with CO<sub>2</sub> MFR at about 600 kg/s (power around 80 MW).

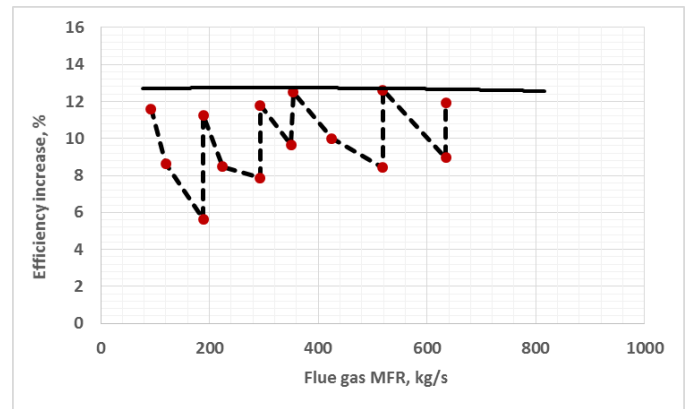
The mass flow rate of the bottoming CO<sub>2</sub> cycle variation at different flue gas mass flow rates is shown in Figure 19. Red points in Figure 19 mean some specific gas turbine unit. As we can observe, the use of unified units leads to some deviation from the curve shown in Figure 18. It is clear that the deviation of CO<sub>2</sub> MFR inevitably leads to a decrease in efficiency and a power growth (Figure 20 and Figure 21).

As can be seen in Figure 20 and Figure 21, the closer the standardized bottoming unit MFR to the thermodynamically unified curve, the higher the efficiency and power gain. Thus, the cost of standardization is efficiency and power growth. However, even with such potentially useful power losses we can obtain at least a 15% relative power increase compared to a standalone gas turbine unit.

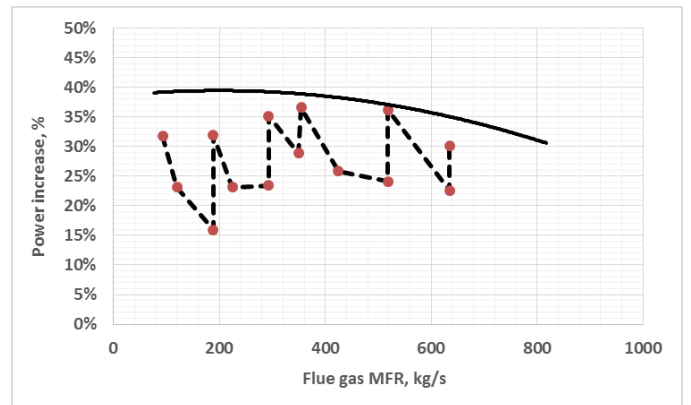
The advantage of complete (thermodynamic & MFR) standardization is that the customer can obtain 15% or higher additional power in relatively short terms with reasonable costs. However, if the customer needs maximum efficiency and power gain (up to 12.7% absolute increase of efficiency and up to 40% relative increase of power), and the terms and costs of manufacturing are a lower priority, an engineered thermodynamically unified cycle sized for maximum net power production should be considered.



**Figure 19: Covering diagram of the whole range of flue gas mass flow rates by completely standardized bottoming units**



**Figure 20: Completely unified bottoming CO<sub>2</sub> unit's absolute thermal efficiency increase vs. flue gas mass flow rate**



**Figure 21: Completely unified bottoming CO<sub>2</sub> unit's relative power increase vs. flue gas mass flow rate**

It should be noted that according to the presented approach some portion of the flue gas will be lost but this is the payment for cycle standardization. Of course, the bottoming system may be specifically designed for each particular GTU but if time of development and final investment are very strict then a unified unit may be a good compromise.

## CONCLUSIONS

The supercritical CO<sub>2</sub> power cycle is an emerging and promising technology for the recovery of heat energy of GTU exhaust gas.

- Bottoming cycle configuration selection should be performed with the consideration of heat source features.
- The simulation in AxCYCLE of three different configurations of S-CO<sub>2</sub> cycle showed that a simple recuperated cycle is the best one as the bottoming unit for GTU.
- The bottoming recuperated S-CO<sub>2</sub> cycle seems like quite a rational alternative to the bottoming steam cycle for small and middle size GTU even though thorough cost analysis would be required to validate this conclusion.
- The proposed universal maps reveal the gain in power generation from a bottoming recuperated S-CO<sub>2</sub> cycle addition to existing GTU. The specified maps are valid for a wide range of GTU.
- Results of the current study confirm the possibility of a standardization of bottoming cycles for use in combined power plants with top-cascading GTU.
- The advantage of complete (thermodynamic & MFR) standardization is that the customer can obtain from 15% to 40% additional power in relatively short terms with reasonable costs. However, if the customer needs maximum efficiency and power gain (around 40%), and the terms and costs of manufacturing are a lower priority, the engineered thermodynamically unified cycle sized for maximum net power production, or even a unique one, should be considered.

## ACKNOWLEDGMENTS

We wish to thank the many people from the SoftInWay Inc. team who generously contributed their time and effort in the preparation of this work. The strength and relevance of the material presented is a result of the inputs, and their insightful contributions are greatly appreciated.

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