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Modeling and Economic Analysis of SCO2 Power Systems Hybridized with a Gas Turbine

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ABSTRACT

This study examines the possibility of integrating supercritical carbon dioxide (SCO2) cycles with gas turbines and the resulting cost and performance implications. Cycles are designed based on common configurations in cost and performance baselines to provide good estimates for cost predictions. Process modeling software is used to calculate performance predictions using real gas properties derived from the REFPROP modeling method. The preheat, recuperation, and overheat (PRO) cycle and dual cascaded (DC) cycle are considered for hybridizing with the simple cycle gas turbine. The system cooling is matched to baseline conditions for comparison purposes. Capital expense, operating expense and the levelized cost of electricity (LCOE) are calculated for the different systems and compared to other baselines.

INTRODUCTION

This study presents findings of scoping study for combining a gas turbine (GT) Brayton power generation cycle with an SCO2 cycle. The motivation of this work is to compare and contrast two different cycle configurations that provide different advantages in applied integration and operation. In addition, the goal of the ongoing work is to build up a system with limited complexity and a preliminary techno-economic assessment of these systems at the 20-30 MW scale and the 900-1,200 MW scale. The goal of the investigation is to begin building up realistic models of these systems, define modes of operation, assess relative advantages to each approach, and provide a basis for future, more detailed investigations. In addition, proving the feasibility of combined cycle at the smaller scale allows for microgrid and distributed power applications to achieve low costs of electricity, previously only achievable for large gas turbine combined cycles (GTCC).

SYSTEM AND MODEL DESCRIPTIONS

The approach for modelling and techno-economic analysis is intended to align with approaches taken by DOE funded baselines to provide a basis for comparison. The large gas turbine systems were modelled after the efficiency and outlet temperature and mass flow of the H-class turbine baseline in the GTCC NREL baseline study.[1] The SCO2 baseline is also provided by NREL for the indirect SCO2 cycle.[2] At the smaller scale, published conditions from gas turbine original equipment manufacturers were used. Generally, ambient conditions and efficiencies were derived from baseline assumptions.

The system modelling was performed in Aspen Plus, with the REFPROP equation of state. This process software was primarily used to estimate the overall system efficiency at on-design conditions. In future work, the models will be given more definition, allowing for off-design modelling, assessment of startup and shutdown, partial operation modes, and other parameters needed to apply to a wide variety of applications beyond the scope of the current study. In general, the model maintained a constant gas turbine design across configurations and attempted to maximize efficiency of the overall system by tweaking flows and pressure ratios in the SCO2 system design.

Ambient conditions at the inlet to the gas turbines are taken to be 25°C. The cooler for the SCO2 system is taken to achieve 35°C in the SCO2 at the compressor inlet, 10 degrees above ambient. Recuperators have a 10°C approach temperature, and the air to SCO2 exchanger (WHR-EX) has a 15°C approach temperature. SCO2 turbines have an isentropic efficiency of 90% and SCO2 compressors have an isentropic efficiency of 80%. Heat exchangers are designed to have a 1% drop in pressure.

Gas Turbine with Dual Cascaded Cycle

The cascaded cycle recuperates the exhaust heat from one stage of expansion into a split flow from the compressor outlet. A dual cascaded performs another step of recuperation after the second turbine, providing heat for a third stage of expansion. The basic block flow diagram of a DC SCO2 cycle is shown in Figure 1. Intercooling may be added to the compressor, however it was not included in this iteration of designs to keep configurations simpler and better match baseline configurations for cost estimation. This cycle is similar to a combined cycle in that the SCO2 cycle is a bottoming cycle that requires heat from the exhaust of the GT to operate. If there is an operational advantage to operating the SCO2 cycle independently, its heat must be provided through duct burners or a similar technology within the WHR-EX system. Additionally, the SCO2 cycle is designed to the turbine exhaust temperature, so its efficiency is limited by the lower maximum temperature.

The flow from the main compressor is split three ways, each leading to a heat exchanger. The heat recovery from the gas turbine exhaust is added to the SCO2 through the WHR-EX exchanger. The high temperature recuperator (HTR) recovers exhaust heat from the first turbine into the flow that feeds the second turbine. The low temperature recuperator (LTR) provides heat from the second turbine exhaust into the flow that feeds the third turbine. All exhaust flows are combined and sent to the cooler to provide heat rejection.

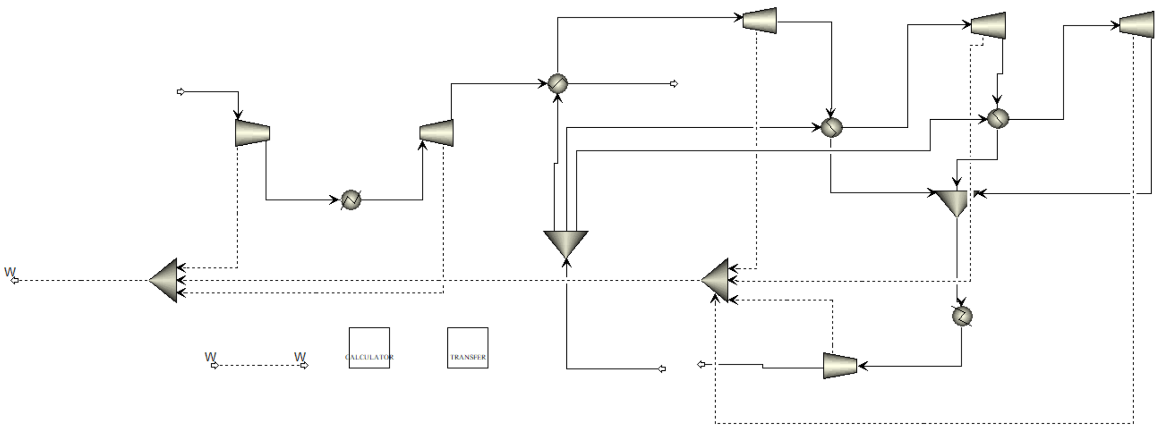

Figure 1. Block Flow of the Gas Turbine with SCO2 Dual Cascaded Cycle

Gas Turbine with Preheat, Recuperation, and Overheat

Another SCO2 cycle was examined for better operational independence. The PRO SCO2 cycle involves splitting flow between two major exchangers, the WHR-EX and the HTR. The basic block flow diagram is shown in Figure 2. The preheat is performed by the GT exhaust through the WHR-EX, the recuperation is performed by the HTR, and the overheat is performed by a primary heater after the high pressure SCO2 flow from the WHR-EX and the HTR are recombined. This system acts like a simple recuperated SCO2 cycle, but with reduction in the HTR duty, by shifting some of the cold, high pressure flow to the WHR-EX instead.

The flow is split to the two exchangers after the main compressor. Again, intercooling is possible in the cycle, but it is not examined at this time to keep the cycle configuration simple and to better match baseline cost data. Flow control splits the SCO2 between the HTR and the WHR-EX. The WHR-EX outlet temperature is limited by the GT exhaust temperature, and the HTR outlet temperature is limited by the SCO2 turbine exhaust temperature. When these two temperatures are brought together and mixed, the average temperature is about 510°C. To increase SCO2 cycle efficiency, a primary heater brings the combined flow temperature up to 680°C. This temperature was chosen based on an understanding of material limits in primary heater construction and SCO2 turbine inlet material limits. The current assumption is that the primary heater is gas-fired, however future versions could explore using thermal energy storage (TES) in conjunction with renewable energy to supply heat to the SCO2 cycle. In order to run the SCO2 cycle independent from the GT, the system could be designed to send the full flow the HTR with minimal pressure drop. This would reduce cycle efficiency. Instead, a good way to run the SCO2 PRO cycle independently would be to add duct burners to the WHR-EX to replace the heat provided by the gas turbine. Future studies will investigate the performance and operational advantage of these cycles under these modes of operation.

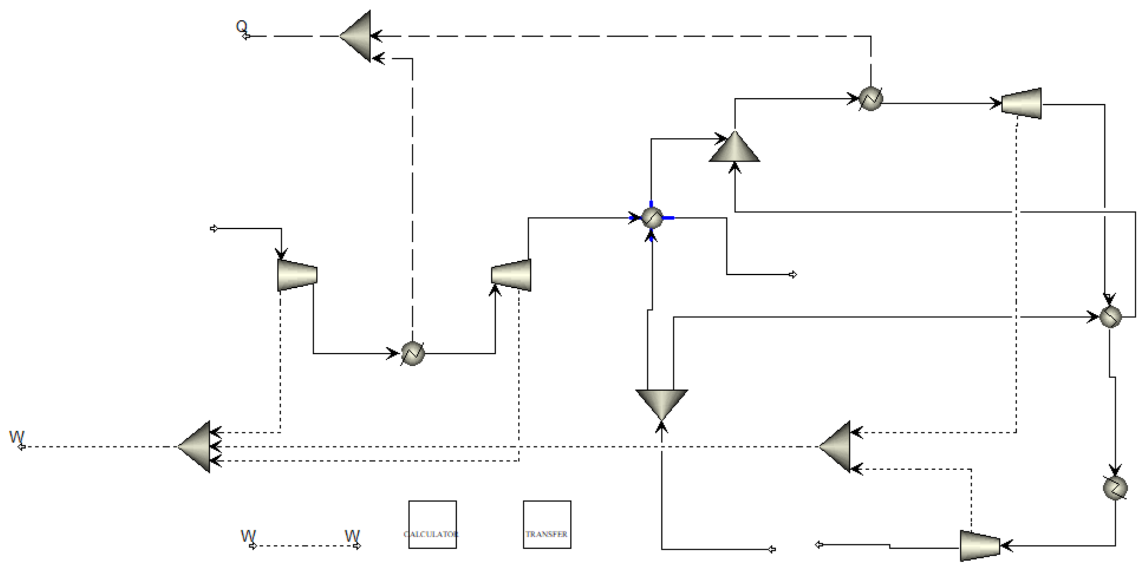

Figure 2. Block Flow of the Gas Turbine with SCO2 Preheat, Recuperation, and Overheat Cycle

PERFORMANCE RESULTS

The systems were modeled in Aspen plus, with performance results shown in Table 1. The cooler outlet pressure was 85 bar. About 1% of pressure loss was assumed in the exchangers. The outlet pressure of the SCO2 main compressor was varied to improve performance. The outlet pressure was 300 bar for the DC cycle and 380 bar for the PRO cycle. Mass flows were varied to minimize temperature differences in heat exchangers, while maintaining approach temperature limits.

In general, the combined cycles with DC cycles performed better in combined thermal efficiency. The H-Class turbine achieves the highest combined cycle efficiency of 59.8%. This is due to the higher turbine efficiency and SCO2 cycle efficiency versus a small GT. All systems perform above 50% thermal efficiency, which is especially impressive at the scale of the small GT. One interesting result of the modelling is the difference in power output across different cycle configurations. For the DC cycle, the power output is less than half of the power produced by the GT. For the PRO cycle, power outputs are roughly equal between the GT and the SCO2 cycles independently. This may provide an operational advantage if a certain power threshold must be met when the cycles are run independently. Also notable is that the PRO cycle efficiency when run independently is higher than the DC cycle. However, for combined operation, the DC cycle has a clear advantage in efficiency.

Table 1. Performance Results for the Microgrid and Utility-Scale System Models

	H-Class GT with DC Cycle	H-Class GT with PRO Cycle	Small GT with DC Cycle	Small GT with PRO Cycle
GT Efficiency (%)	43.8%	43.8%	39.1%	39.1%
GT Output Size (kW)	685,495	685,495	13,962	13,962
SCO2 Efficiency (%)	32.4%	39.2%*	31.9%	39.1%*
SCO2 Output Size (kW)	252,139	603,320	5,890	15,074
System Total Output Size (kW)	937,634	1,288,815	19852	29,036
Combined System LHV Thermal Efficiency (%)	**59.8%**	**55.0%**	**55.6%**	**52.4%**
*Note that the SCO2 PRO cycle efficiency is in standalone operation assuming heat through the WHR-EX is provided externally and not by the GT exhaust				

DC Cycle Heat Exchanger Results

For examining the ability of the SCO2 system to effectively move heat, the heat exchanger curves are provided. For brevity, the H-class heat exchanger curves are shown, but the differences seen in the SCO2 heat exchangers at the small scale will be discussed.

Figure 3 shows the results for the H-Class GT providing heat to the WHR-EX in the DC cycle. The hot flue gas inlet is at 596°C and the approach temperature of 15°C occurs at the hot end. The temperature difference at the cold end is about 18°C. For the small GT, the hot stream inlet is lower at 586°C. The outlet temperature of the hot stream is higher, with a 37°C difference between stream temperatures.

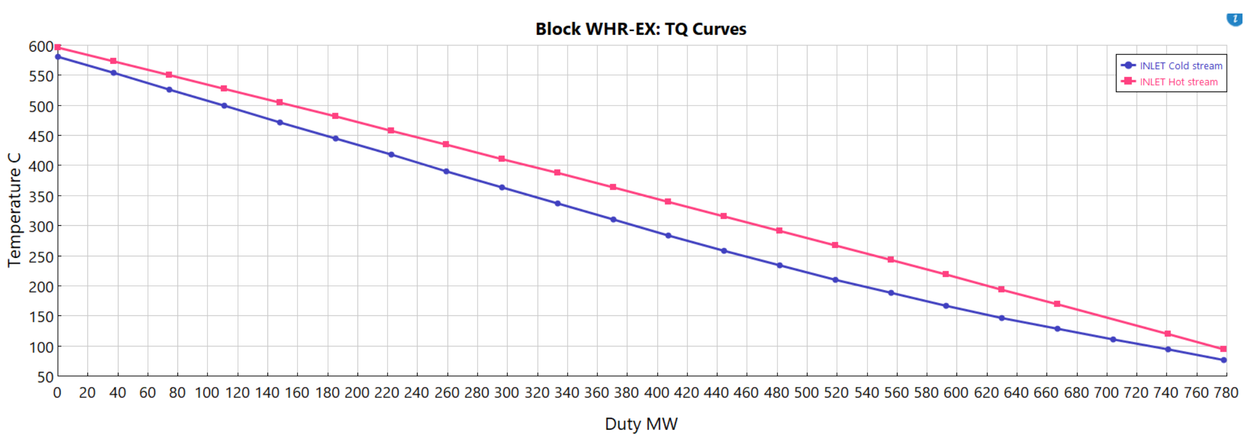

Figure 3. Temperature vs Duty in the WHR Exchanger for the H-Class GT+SCO2 DC Cycle

Figure 4 shows the results for the DC cycle HTR. The SCO2 inlet is at 430°C and the approach temperature of 10°C occurs at the hot end. The temperature difference at the cold end is about 12°C. For the small GT, the hot stream inlet is lower at 421°C. The outlet temperature of the hot stream is higher, with a 14°C difference between stream temperatures.

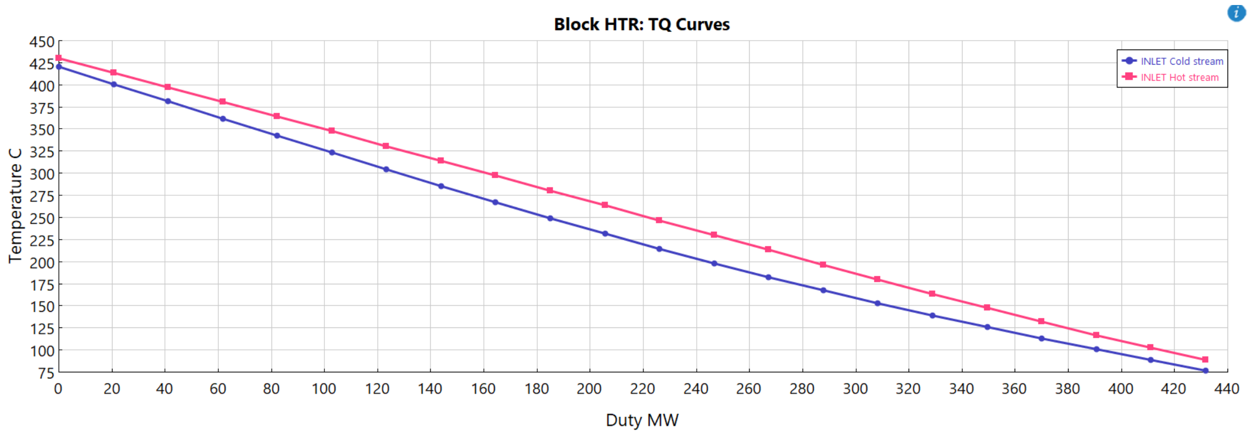

Figure 4. Temperature vs Duty in the HTR Exchanger for the H-Class GT+SCO2 DC Cycle

Figure 5 shows the results for the DC cycle LTR. The hot SCO2 inlet is at 285°C and the approach temperature of 10°C occurs at the cold end. The temperature difference at the hot end is close to the approach temperature at 11°C. For the small GT, the hot stream inlet is lower at 273°C. The approach temperature is also at the cold end of the LTR. The outlet temperature of the cold stream is lower, with a 13°C difference between stream temperatures. The cooler inlet temperature is 102°C for the H-Class DC SCO2 cycle and 101°C for the small GT DC SCO2 cycle.

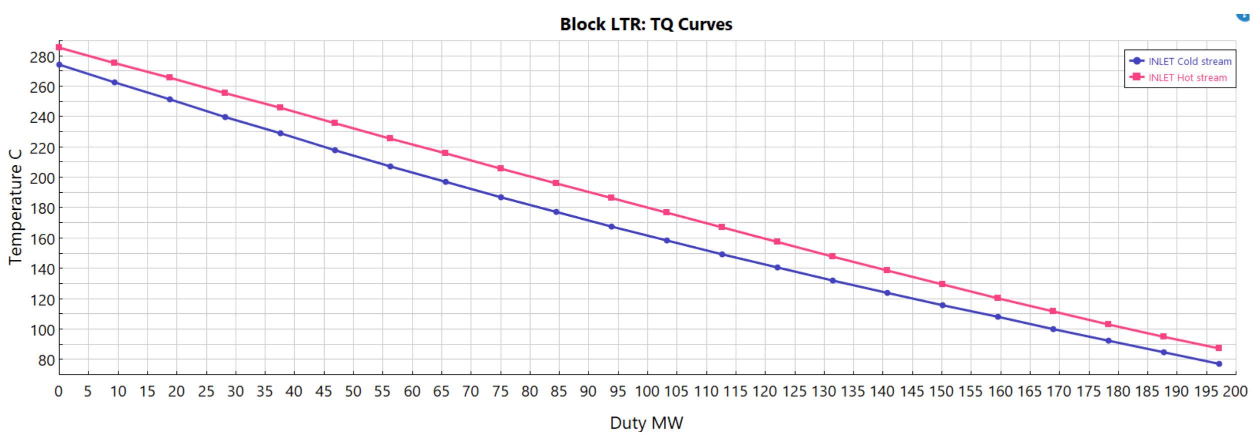

Figure 5. Temperature vs Duty in the LTR Exchanger for the H-Class GT+SCO2 DC Cycle

PRO Cycle Heat Exchanger Results

Figure 6 shows the results for the H-Class GT providing heat to the WHR-EX in the PRO cycle. The hot flue gas inlet is at 596°C and the approach temperature of 15°C occurs at the hot end, although a similar temperature difference is achieved at the cold end. For the small GT with PRO cycle, the hot stream inlet is lower at 586°C, with similar approach temperatures at either end of the exchanger.

Figure 6. Temperature vs Duty in the WHR Exchanger for the H-Class GT+SCO2 PRO Cycle

Figure 7 shows the results for the PRO cycle HTR. The SCO2 hot stream enters the HTR at 490°C and the approach temperature of 10°C occurs at the hot end. The temperature difference at the cold end is about 55°C. For the small GT, the hot stream HTR temperature conditions are the same as the H-Class scale.

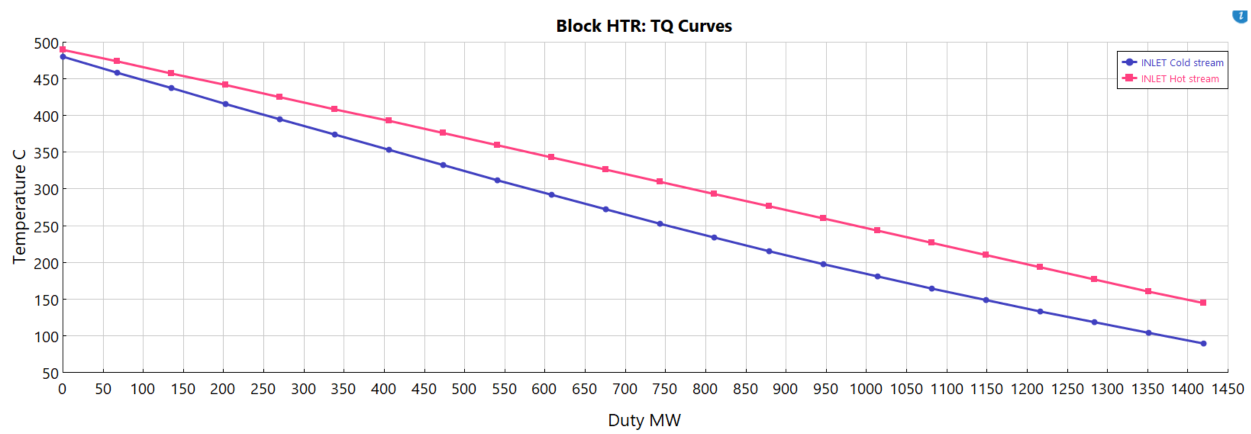

Figure 7. Temperature vs Duty in the HTR Exchanger for the H-Class GT+SCO2 PRO Cycle

Target Applications and Off-Design Performance

Matching other baselines for GT and SCO2 [1][2], a constant, baseload profile is adopted for the utility-scale cases. These cases run at full power and include an 85% capacity factor. For the microgrid scale, a more variable profile was adopted because a small, combined cycle on a microgrid will have to provide baseload and peaking capability. The Southwest Research Institute load profile was used as a reference case for a microgrid. Some characteristics of the profile are shown in Figure 8. The profile has higher peaks on a Monday-Friday basis that become more pronounced in summer months. Weekends see much smaller peaks in power usage. Additionally, there is more variation in load in the spring and autumn seasons than summer and winter. The load profile was scaled such that the maximum annual load matched the maximum capacity of the combined cycle output.

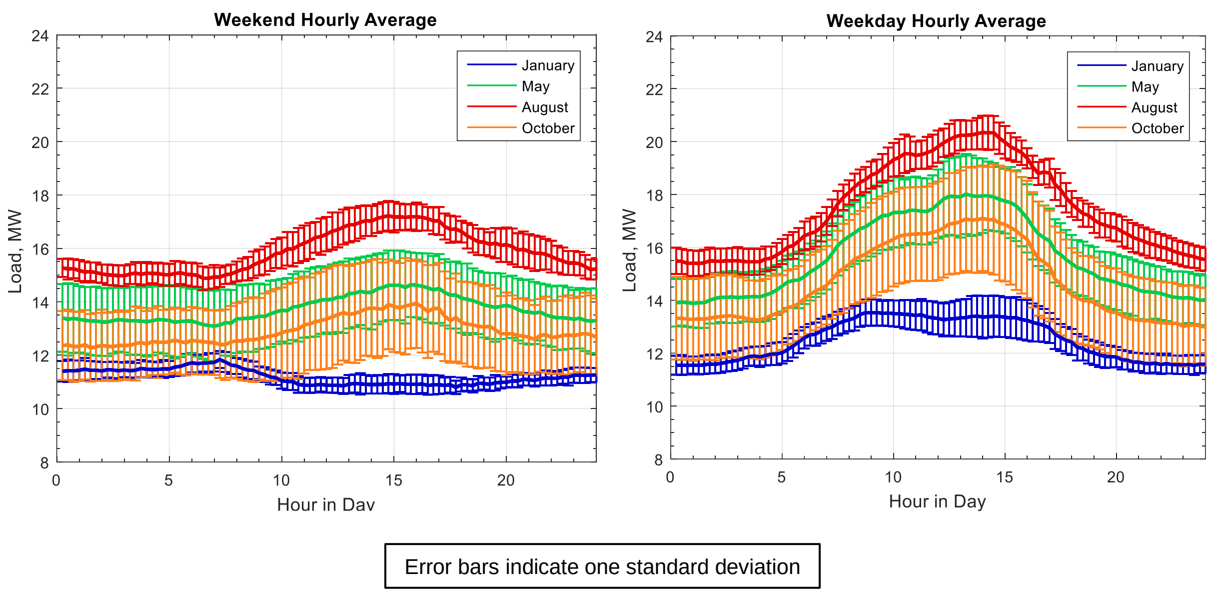

Figure 8. Select Load Data from the SwRI Microgrid Demand Profile

A simple estimate of off-design performance for the combined cycles at the microgrid-scale was adopted. The profiles for each cycle in part-load are shown in Figure 9. This was derived from the work of Allison, et al [3], which discussed the performance of a combined cycle GT with SCO2 at similar scale. Future work will further adapt the Aspen Plus models to create off-design models specific to these cases, and pair the systems with renewables and energy storage.

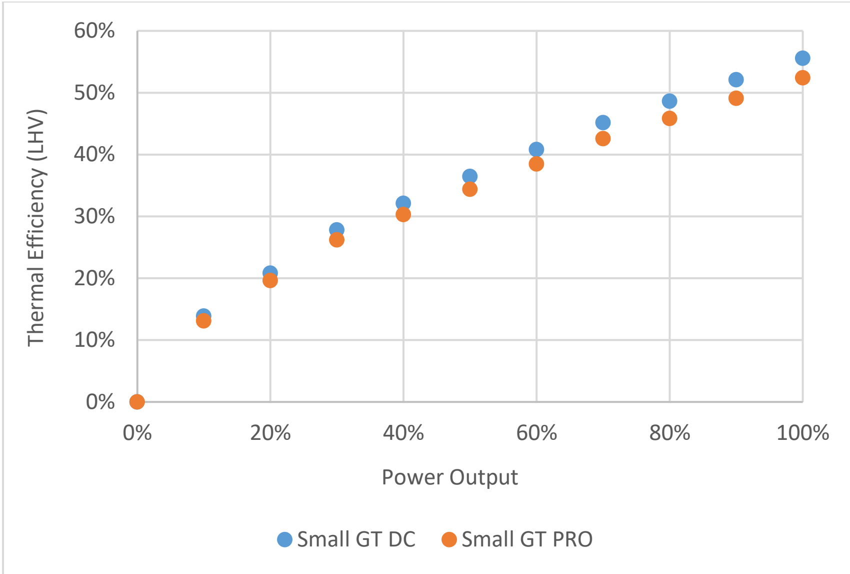

Figure 9. Estimate of Off-Design Performance for the Microgrid Combined Systems

TECHNOECONOMIC SOURCES AND INPUTS

Capital expenditure (CAPEX) and operating expenditure (OPEX) are generally derived from the DOE baselines [1][2]. For this study, the costs are reduced to a total plant cost per kilowatt basis and cost data is only used if it comes from a source that is close to the scale of the application. The DOE baseline for indirect SCO2 [2] contains cases for a 620°C system and a 760°C turbine inlet temperature systema, the description of material impacts for each temperature indicated that the 620°C baselines aligned more closely to the current study. This study also includes costs for coal firing and cleanup, which were removed from the cost basis for this study. Additionally, the SCO2 cycles feature a recompressor in the system configuration, which was also removed from the cost basis. The coal heater from the baseline cost was kept as the cost for the WHR-EX and primary heater. For natural gas firing, the equipment from the GTCC baseline [1] was used in the cost basis.

For the small GT and the SCO2 cycles at the microgrid scale, costs were estimated based on a study of an SCO2 system at the 5 MW scale.[3] The small GT cost information was sourced from an EPA study.[4] The CAPEX cost inputs are shown in Table 2. Based on the baselines the engineering procurement and construction (EPC) contractor costs are estimated to be 20%. Similarly cost estimates on OPEX were derived from the baselines. There is a fixed capacity cost which is applied to each system. There is also a variable OPEX cost that applies to the power produced. The parameters are shown in Table 3.

Table 2. Cost Inputs for Estimation of CAPEX

	Small GT with DC Cycle	Small GT with PRO Cycle	H-Class GT with DC Cycle	H-Class GT with PRO Cycle
GT System Capacity Cost ($/kW$_{AC}$)	1,510	1,510	771	771
SCO2 System Capacity Cost ($/kW$_{AC}$)	2,900	2,900	2,130	2,087
Combined Capacity Cost ($/kW$_{AC}$)	1,946	2,232	1,137	1,388

The fixed OPEX is multiplied by the system size. The variable OPEX was calculated by separating the power produced by each subsystem in the combined cycle and multiplying by the parameter. Fuel costs are derived from the DOE baseline[1] at bar (4.90 $/MMBTU$_{LHV}$).

Table 3. Cost Inputs for Estimation of OPEX

GT Fixed OPEX	$/kW	26
GT Variable OPEX	$/MWh	1.2
SCO2 Fixed OPEX	$/kW	113
SCO2 Variable OPEX	$/MWh	4.4

The analysis seeks to calculate the LCOE of each system. The approach is derived from the NREL baseline for photovoltaics[5], to facilitate integration with solar generation and TES in future studies. In the utility-scale analysis, the period of analysis is 30 years. 28.2% of the CAPEX is paid for upfront, with 71.8% financed at a fixed rate of 5% over 20 years. The inflation rate is taken to be 2.5%, and the real discount rate is 5.10%. The nominal discount rate, which includes inflation is 7.73%. The nominal rate is applied to all cash flows. The real discount rate applies to the LCOE. The LCOE is calculated by using the discount rates to get the net present value of all cash flows, divided by the generation of the system, and the net present value of the LCOE.

TECHNOECONOMIC RESULTS

The results for CAPEX, OPEX and fuel costs are presented in Table 4. Values vary due to varying capacities and annual power production profiles. The PRO Cycle is larger than the DC cycle in capacity, so it generally has higher CAPEX and OPEX. For CAPEX and OPEX, the GT with DC cycle is lower than the PRO cycle when normalized to capacity and power generation. The combined cycle capacity cost is shown in Table 2, and is lower for the GT with DC cycle. In addition, the calculated efficiency of the GT with DC combined cycle is better, improving OPEX. However, there may be operational advantages to the PRO cycle and future work will study operational strategies.

Table 4. Cost Results for Microgrid and Utility-Scale Systems

	Small GT with DC Cycle	Small GT with PRO Cycle	H-Class GT with DC Cycle	H-Class GT with PRO Cycle
CAPEX GT System	\$21.1 M	\$21.1 M	\$528.8 M	\$528.8 M
CAPEX SCO2 System	\$17.1 M	\$43.7 M	\$537.1 M	\$1,259.4 M
CAPEX Combined System	\$38.2 M	\$64.8 M	\$1,066.0 M	\$1,788.3 M
EPC and Owner's Costs	\$7.6 M	\$13.0 M	\$213.2 M	\$357.7 M
Total CAPEX	**\$45.8 M**	**\$77.8 M**	**\$1,279.2 M**	**\$2,145.9 M**
OPEX GT System	\$0.5 M	\$0.6 M	\$26.2 M	\$29.3 M
OPEX SCO2 System	\$1.2 M	\$2.4 M	\$59.2 M	\$110.4 M
Total OPEX	**\$1.6 M**	**\$3.0 M**	**\$85.4 M**	**\$139.7 M**
Annual Payment for 20-year Financing	\$2.6 M	\$6.7 M	\$195.1 M	\$291.6 M
Capacity Factor (%)	63.4%	63.4%	85%	85%
Power Exports (MWh)	110,301	161,330	6,981,623	9,596,516
Natural Gas Imports ($tonne_{NG}$)	19,821	30,761	890,444	1,330,768
Annual Fuel Cost (\$)	**\$4.3 M**	**\$6.7 M**	**\$195.1 M**	**\$291.6 M**

The LCOE of the system was calculated and the results are shown in Table 5 and Figure 10. In general, the bulk of cost in these systems is in the OPEX and fuel costs. The H-class GT with DC SCO2 system performs the best at \$40.8/MWh. This is competitive with the DOE baseline CCGT H-class system, which is estimated in their methods to be \$42.7/MWh (Case B32A).[1] All of the cases, even at small scales outperform the coal cases, which range from \$123.0/MWh (Case RhtIC620) to \$128.2 (Case Baseline620).[2]

Table 5. LCOE Breakdown for Microgrid and Utility-Scale Systems

	Small GT with DC Cycle	Small GT with PRO Cycle	H-Class GT with DC Cycle	H-Class GT with PRO Cycle
GT System LCOE	10.7	7.3	4.2	3.1
SCO2 System LCOE	8.7	15.2	4.3	7.4
EPC LCOE	3.9	4.5	1.7	2.1
OPEX LCOE	11.3	14.0	9.3	11.1
Fuel LCOE	29.9	31.8	21.2	23.1
Combined System LCOE (\$/MWh_{AC})	**64.5**	**72.8**	**40.8**	**46.7**

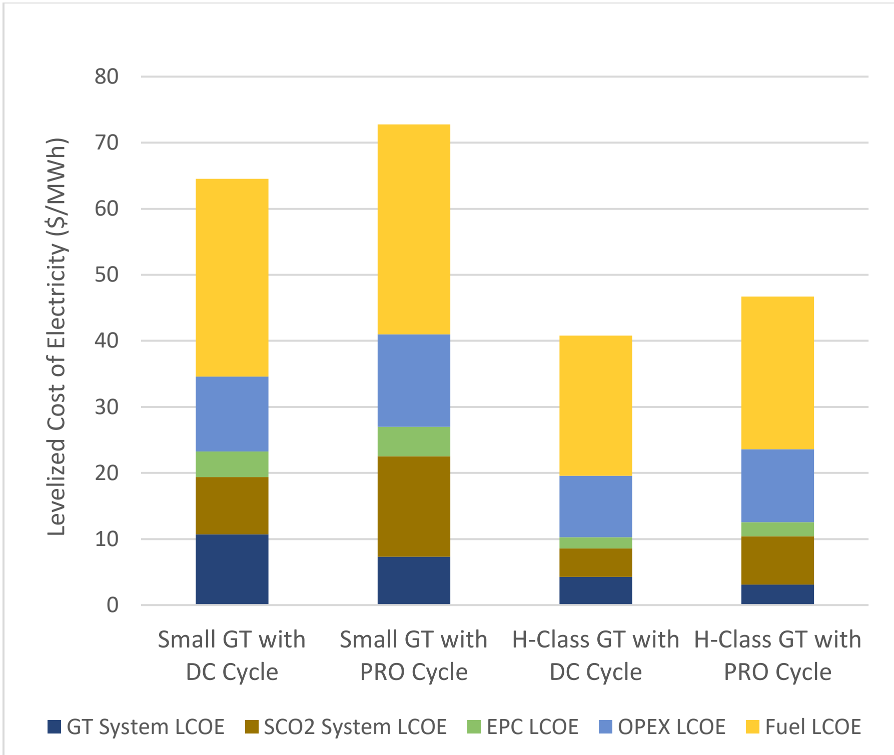

Figure 10. LCOE Results for the Microgrid and Utility-Scale Systems

The results for LCOE of an H-Class GT with SCO2 DC cycle align closely with the results for the DOE GTCC, except fuel which is about $2/MWh higher and accounts for the improvement in results in the current study. This may be due to the escalation factors assumed in the baseline, and future investigation will attempt to capture these effects.

CONCLUSION

The study investigated two different styles of combined cycle with GT and SCO2 at two different scales each. The utility-scale systems used an H-class turbine to provide heat to the SCO2 cycle. The microgrid-scale systems used a small GT in a similar fashion. The SCO2 cycles were either the DC or PRO configuration, with the DC configuration taking its heat from the WHR-EX, and the PRO configuration adding a primary heater to improve the SCO2 cycle turbine inlet temperature and overall cycle efficiency. The DC cycle resulted in SCO2 systems that were much smaller than the gas turbine system output. The PRO cycle was typically similarly sized in output to the GT. The H-class GT with a DC SCO2 cycle was the most efficient cycle. Technoeconomic analysis estimated the CAPEX, OPEX, and fuel costs of the systems. An LCOE was calculated for all four systems, with the H-class GT with a DC SCO2 cycle having the lowest LCOE at $40.8/MWh. All systems are competitive with baseline estimates, and the microgrid-scale systems offer a promising LCOE for that scale of system.

REFERENCES

[1] Schmitt, T., Leptinsky, S., Turner, M., Zoelle, A., Woods, M., Shultz, T., and James, R. "Fossil Energy Baseline Revision 4a", Vol. 1, October 2022, National Energy Technology Laboratory, Pittsburgh, DOE/NETL – 2023/4320

[2] White, C., Gray, D., Plunkett, J., Shelton, W., Weiland, N., and Shultz, T., "Techno-economic Evaluation of Utility-scale Power Plants Based on the Indirect SCO2 Brayton Cycle", September 2017, National Energy Technology Laboratory, Pittsburgh, NETL_PUB_21490

[3] Allison, T., Wygant, K., Wilkes, J., and Pelton, R. "Technoeconomic Comparison of SCO2 and ORC Systems for Waste Heat Recovery", *Proceedings of the ASME Turbo Expo 2022: Turbomachinery Technical Conference and Exposition*, Rotterdam, The Netherlands, June 13-17, 2022, ASME

[4] Darrow, K., Tidball, R., Wang, J., and Hampson, A., "Section 3. Technology Characterization – Combustion Turbines", *Catalog of CHP Technologies*, September 2017, U.S. Environmental Protection Agency

[5] Ramasamy, V., Zuboy, J., O'Saughnessy, E., Feldman, D., Desai, J., Woodhouse, M., Basore, P., and Margolis, R., "U.S. Solar Photovoltaic System Cost Benchmark: Q1 2022", September 2022, Golden, CO, National Renewable Energy Laboratory, NREL/TP-7A40-83586