Efficient Generation: Combustion Turbine Electric Generating Units
Technical Support Document

New Source Performance Standards for Greenhouse Gas Emissions from New, Modified, and Reconstructed Fossil Fuel-Fired Electric Generating Units; Emissions Guidelines for Greenhouse Gas Emissions from Existing Fossil Fuel-Fired Electric Generating Units; and Repeal of the Affordable Clean Energy Rule

Final Rule

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1 Introduction

Multiple design and operation and maintenance (O&M) parameters influence the efficiency of a stationary combustion turbine electric generating unit (EGU). This technical support document describes these factors for both simple and combined cycle combustion turbines. It describes designs and O&M practices that can improve the efficiency of combustion turbines, which reduces fuel consumption and therefore emissions of greenhouse gases (GHGs).

The following is a summary of some of the design parameters that impact the carbon dioxide (CO_2) emission rates of combustion turbine EGUs. This summary is not intended to be a comprehensive list of all design parameters that influence the CO_2 emission rates.

As the thermal efficiency of a combustion turbine increases, less fuel is burned per gross megawatt hour (MWh) of electricity produced by the turbine-generator and there is a corresponding decrease in CO_2 and other air emissions. Efficiency is reported as the percentage of the energy in the fuel that is converted to electricity.[1] Heat rate is another common way to express efficiency. Heat rate is expressed as the amount of British thermal units (Btu), or kilojoules (kJ), required to generate a kilowatt-hour (kWh) of electricity. Lower heat rates are associated with more efficient power generation. Efficiency improvements can be expressed in different formats; they may be reported as an absolute change in overall efficiency (*e.g.*, a change from 40 percent to 42 percent represents a 2 percent absolute increase). They may also be presented as the relative change in efficiency (*e.g.*, a change from 40 percent to 42 percent results in a 5 percent reduction in fuel use). The relative change in efficiency is the most consistent approach because it corresponds to the same change in heat rate. For most combustion turbine EGUs, as heat rates are reduced there are corresponding reductions in fuel extraction-related environmental impacts as well as associated thermal impacts on cooling water ecosystems.[2]

The electric energy output for an EGU can be expressed as either "gross output" or "net output." The gross output of an EGU is the total amount of electricity generated at the generator terminal. The net output of an EGU is the gross output minus the total amount of auxiliary (*i.e.*, parasitic) electricity used to operate the EGU (*e.g.*, electricity to power fuel handling equipment, pumps, fans, pollution control equipment, and other onsite electricity needs), and thus is a measure of the electricity delivered to the transmission grid for distribution and sale to customers.

1.1 Types of Combustion Turbines

Combined cycle EGUs are power plants using both a combustion turbine engine (*i.e.*, the "topping" or Brayton cycle) and a steam turbine (*i.e.*, the "bottoming" or Rankine cycle) to generate electricity. First, fuel is burned in a combustion turbine engine and the high-temperature exhaust is recovered by a heat recovery steam generator (HRSG) to create additional thermal output (*e.g.*, steam). Next, the steam is used as the working fluid in a Rankine cycle and expanded through a steam turbine to produce

[1] For example, a 40 percent efficient combustion turbine converts 40 percent of the energy in the fuel to useful output.

[2] Combined cycle EGUs using dry cooling or simple cycle EGUs do not have cooling water ecosystem impacts.

additional power.[3] Combined cycle units have significantly higher efficiencies compared to simple cycle turbines—combustion turbine engines where heat from the high-temperature exhaust is not recovered.

1.2 Design Efficiencies of Combustion Turbines

EGU efficiency generally increases with size. Larger EGUs tend to be more efficient because they are usually newer and use advanced technologies. Economy of scale allows the use of higher cost improvements to be more economic, but these are not inherent differences in efficiencies. For combined cycle turbines, as equipment size increases above approximately 2,000 MMBtu/h, the differences in these losses start to taper off.

1.3 Factors Impacting Combustion Turbine Efficiency

Multiple factors can influence the efficiency of a combustion turbine. Factors include, but are not limited to, fuel type, ambient conditions, and pollution control equipment.

1.3.1 Fuel type

Combustion turbines tend to be most efficient when burning natural gas. When burning other fuels, such as distillate oil or hydrogen, design efficiencies decrease.

1.3.2 EGU cooling system

Reducing the condenser temperature improves the efficiency of a steam turbine converting the thermal energy in the steam to electricity. Once-through cooling systems can achieve a lower condenser temperature and have an efficiency advantage over recirculating cooling systems (*e.g.*, cooling towers) and dry systems. However, once-though cooling systems typically have larger water-related ecological concerns than recirculating cooling systems. The International Energy Agency[4] (IEA) estimates that similar EGUs using the Rankine cycle with once-through cooling systems that use sea water and river water would have 2.4 percent and 1.5 percent lower heat rates (*i.e.*, more efficient), respectively, than an identical EGU using a recirculating cooling system. Conversely, an identical EGU using a dry cooling system would have a 5 percent higher (*i.e.*, less efficient) heat rate. Where the efficiency of once-through cooling systems is impacted by the cooling water temperature, the efficiency on recirculating cooling systems is influenced by the wet bulb temperature (*i.e.*, the air temperature accounting for the relative humidity) and dry cooling systems are impacted by the actual temperature of the air (*i.e.*, dry bulb temperature). Also, geographic location influences the type of cooling system that can be used (*e.g.*, EGUs in locations with limited water availability rarely, if ever, use once-through cooling and might elect to use dry cooling).

1.3.3 EGU geographic location and ambient conditions

Ambient temperatures and elevation at the site of a facility may potentially have an impact on EGU efficiency. Cooler ambient temperatures theoretically could increase the overall combustion turbine efficiency by decreasing the power required by the combustion turbine engine compressor. In addition, the efficiency of Rankine cycle of combined cycle combustion turbines can increase at lower temperatures from increasing the draft pressure of the HRSG flue gases and the condenser vacuum and

[3] https://www.ge.com/gas-power/resources/education/combined-cycle-power-plants
[4] Coal Industry Advisory Board to the International Energy Agency, "Power Generation from Coal," (Paris, 2010). Available at: https://www.iea.org/reports/power-generation-from-coal-2010.

by increasing the efficiency of the cooling system. However, higher ambient temperatures potentially lower the amount of fuel required to increase the temperature of the combustion air and reduce radiative losses. The IEA[5] estimates that net heat rate of a Rankine cycle increases by 0.15 percent for each 1 degree Celsius (^{o}C) increase in ambient temperature.

1.3.4 EGU duty cycle & load generation flexibility requirements

In general, operating an EGU as a base load unit is more efficient than operating an EGU as a load cycling unit at a lower duty cycle to respond to fluctuations in customer electricity demand.

1.3.5 Other factors

Pollution control devices require electricity to operate and increase the auxiliary (*i.e.*, parasitic) loads. This reduces the net efficiencies of EGUs. The auxiliary load requirements for nitrogen oxide (NO_X) controls are typically 0.5 percent of the gross output of the facility.

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[5] Coal Industry Advisory Board to the International Energy Agency, "Power Generation from Coal," (Paris, 2010). Available at: https://www.iea.org/reports/power-generation-from-coal-2010.

2 Potential Efficiency Gains at Simple Cycle EGUs

Simple cycle designs have been iterated over the years to improve efficiency, increase capacity, and reduce emissions. Efficiency improvements include increased firing temperatures, increasing compression ratios, and the addition of intercooling. According to *Gas Turbine World*, the design net efficiency of new simple cycle turbines can range from 32 to 40 percent. These are design efficiencies at specified conditions, and both power output and efficiency are impacted by ambient conditions. A combustion turbine operates on a fixed volumetric flow of air to the compressor at a given inlet guide vane position. At higher temperatures and elevations, the density of the air entering the compressor is lower, reducing the mass flow through the turbine and consequently less air is available for combustion. Since the combustion turbine maximum heat input is reduced, the combustion turbine engine output is less than the rated output. In addition, as the air inlet temperature increases, more work is required to accomplish the specified pressure rise. The increased work is provided by the combustion turbine and less is available to rotate the generator to produce electricity. At lower temperatures the opposite occurs—output and efficiency increase compared to design specifications. For every °C increase in ambient temperature, combustion turbine output is decreased 0.5 to 1 percent and the heat rate increases 0.15 to 0.4 percent.[6, 7] Humidity also impacts the output and heat rate of a combustion turbine. As humidity increases, the density of the air decreases which reduces the mass flow through the compressor. Higher humidity also results in a reduction in compressor efficiency, increasing the heat rate.[8]

One approach owners/operators of combustion turbines can take to reduce the capacity losses and increased heat rates due to higher ambient temperatures is precooling the combustion turbine inlet air.[9, 10] Owners/operators employ inlet air cooling techniques that generally fall into two categories: evaporative cooling and chilling systems. Evaporative cooling works by adding liquid water to the combustion air.[11] As the water evaporates, it cools the combustion air. Chilling systems use mechanical

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[6] Farouk, N., Sheng, L., & Hayat, Q. (2013). Effect of ambient temperature on the performance of gas turbines power plant. *IJCSI International Journal of Computer Science Issues*, *10*(1,3), 439-442. https://www.ijcsi.org/papers/IJCSI-10-1-3-439-442.pdf.

[7] Higher air pressure (at a constant temperature) results in a relatively slight decrease in the heat rate. Enge, Jason (August 2, 2022). *Ambient Factors Conditions and Combustion Turbine Performance.* Fossil Consulting Services, Inc. https://www.fossilconsulting.com/2022/08/02/ambient-factors-conditions-and-combustion-turbine-performance/.

[8] Enge, Jason (August 2, 2022). *Ambient Factors Conditions and Combustion Turbine Performance.* Fossil Consulting Services, Inc. https://www.fossilconsulting.com/2022/08/02/ambient-factors-conditions-and-combustion-turbine-performance/

[9] Pre-cooling of combustion turbine inlet air can be accomplished by using either an evaporative cooling or a chilling system. Evaporative cooling is limited by the wet bulb temperature and is most effective in areas with low humidity. Chilling systems can cool the inlet air below the dew point temperature but can have significant auxiliary loads. The auxiliary load of the chilling system can be reduced if absorption chillers are used instead of mechanical chillers. Compressed air injection is an alternate way to recover capacity that is lost due to high ambient temperatures (https://powerphase.com/turbophase-air-injection/).

[10] GTW (2021). *2021 GTW Handbook.* Volume 36. Page 79. Pequot.

[11] Water used in combustion turbines must be of high quality (*e.g.* demineralized water) to prevent deposits and corrosion from occurring in the turbine engine.

or adsorption chillers to reduce combustion air temperature.[12] One common type of cooling is inlet fogging, an evaporative cooling system where fine water particles (typically less than 20 microns) are sprayed into the inlet combustion turbine air, leading to lower inlet air temperatures and higher efficiencies.[13] Wet compression is a system like inlet fogging but with higher efficiency. In wet compression, an excess of fog is sprayed into the inlet air so that fog still exists after the air is fully saturated. Some of the excess fog droplets are not evaporated until they are carried into the compressor, which provides additional cooling and results in further power increases of the combustion turbine engine.[14]

General Electric (GE) has intercooling technology, called "SPRINT" or "SPRay INTercooling," that is paired with an LM6000 combustion turbine. The SPRINT™ technology uses demineralized water that is atomized with high-pressure compressed air and sprayed into the inlet of the low-pressure compressor and high-pressure compressor. This results in a higher mass flow through the compressor and increased power output. Moreover, this technology can result in high incremental output and improved efficiency as ambient temperatures rise.[15] The design output and efficiencies for three different LM6000 combustion turbines with and without SPRINT technology are outlined in **Figure 1**.

FIGURE 1: COMPARISON OF LM6000 COMBUSTION TURBINES WITH AND WITHOUT SPRINT™ TECHNOLOGY

Combustion turbine*	ISO Base Load (MW)	LHV Efficiency (%)
LM6000 PC	46.6	40.0%
LM6000 PC Sprint	51.1	39.8%
LM6000 PG	56	39.1%
LM6000 PG Sprint	57.2	38.7%
LM6000 PF	44.7	41.4%
LM6000 PF Sprint	50.0	41.4%
LM6000 PF+	53.9	40.8%
LM6000 PF+ Sprint	57.1	40.8%

[12] United States Environmental Protection Agency (EPA) (2022). *Available and Emerging Technologies for Reducing Greenhouse Gas Emissions from Combustion Turbine Electric Generating Units.* EPA Office of Air and Radiation. April 21, 2022. Accessed at https://www.epa.gov/system/files/documents/2022-04/epa_ghg-controls-for-combustion-turbine-egus_draft-april-2022.pdf.
[13] Meher-Homji, C., Mee, T. (2000). *Gas Turbine Power Augmentation by Fogging of Inlet Air*. Proceedings of the 28th turbomachinery Symposium (2000), Texas A&M. Accessed at https://oaktrust.library.tamu.edu/bitstream/handle/1969.1/163382/Vol28010.pdf?sequence=1&isAllowed=y.
[14] Savic, S., Hemminger, B., Mee, T. (2013). *High Fogging Application for Alstom Gas Turbines.* Proceedings of PowerGen. November 2013. Accessed at http://www.meefog.com/wp-content/uploads/High-Fogging-Alsotom-Mee_-2013-2.pdf.
[15] GE (n.d.) *SPRINT * SPray INTercooling for power augmentation*. Accessed at https://www.ge.com/gas-power/services/gas-turbines/upgrades/sprint.

* The "PC" version of the LM6000 is the general purpose "power conversion" option. The power generation (PG) version runs at higher speeds for increased output, and the power generation and flexibility (PF) version is optimized for increased efficiency of electricity production. The PF+ version has greater output, but a lower efficiency.

Siemen's intercooling technology, called Inlet Spray Intercooling (ISI), is available on the STG-A65 combustion turbine. **Figure 2** shows that at ISO conditions the primary benefit of intercooling is increased output.[16]

FIGURE 2: COMPARISON OF SGT-A65 COMBUSTION TURBINES WITH AND ISI

Combustion turbine	ISO Base Load (MW)	LHV Efficiency (%)
SGT-A65 DLE	59.6	43.2%
SGT-A65 DLE with ISI	64.9	43.3%
SGT-A65 WLE with ISI	70.8	41.4%

Demineralized water injection into the combustor through the fuel nozzles also provides NO_X control. NO_X emissions from the combustor have been shown to increase exponentially with increasing temperatures. Thus, water injected into the combustor flame area lowers the temperature and, consequently, reduces NO_X emissions.[17] Water injection is estimated to result in a 60 to 80 percent reduction of NO_X emissions for a diffusion flame design combustion turbine.[18, 19] Additionally, experimental results have shown that water injections can lower exhaust gas temperatures and reduce NO_X emissions by 70 percent and 57 percent, respectively.[20] Water injection also increases the mass flow rate and the power output, but the energy required to vaporize the water can reduce overall efficiency.

2.1 Steam Injection

Steam injection is like water injection, except that steam is injected into the compressor and/or through the fuel nozzles directly into the combustion chamber instead of water. Steam injection reduces NOx emissions and has the advantage of improved efficiency and larger increases in the output of the combustion turbine. Multiple vendors offer different variations of steam injection. The basic process uses a relatively simple and low-cost HRSG to produce steam, but instead of recovering the energy by expanding the steam through a steam turbine, the steam is injected into the combustion chamber and the energy is extracted by the combustion turbine engine.[21] Combustion turbines using steam injection have

[16] ETN Global. SGT-A65. Accessed at https://etn.global/gas-turbine-products/sgt-a65/.
[17] In general, the additional of liquid water or steam will not increase emissions of carbon monoxide or unburned hydrocarbons. However, at higher injection rates emissions of carbon monoxide and unburned hydrocarbons can increase.
[18] EPA (2002). *CAM Technical Guidance Document.* B.17 Water or Steam Injection. Accessed at https://www3.epa.gov/ttnchie1/mkb/documents/B_17a.pdf.
[19] Water injection is not used for NO_X control in combustion turbines using lean premixed combustion (i.e., dry low NO_X) systems.
[20] Kotob, M. R., Lu, T., Wahid, S. S. (2021). *Experimental comparison between steam and water tilt-angle injection effects on NO_x reduction from the gaseous flame.* Royal Society of Chemistry. https://doi.org/10.1039/D1RA03541J.
[21] Innovative Steam Technologies. *GTI*. Accessed at https://otsg.com/industries/powergen/gti/.

characteristics of both simple cycle and combined cycle units. For example, when compared to standard simple cycle turbines, they are more efficient but more complex with higher capital costs. Conversely, compared to combined cycle combustion turbines, they are simpler and have shorter construction times, lower capital costs, but have lower efficiencies.[22, 23] Combustion turbines using steam injection can start quickly. Have good part load performance and can respond to rapid changes in demand—making the technology a potentially a good solution to reduce GHG emissions from low and intermediate load combustion turbines. A potential drawback of steam injection is that the additional pressure drop across the HRSG can reduce the efficiency of the combustion turbine when the facility is running without the steam injection operating.

A steam injection gas turbine cycle (STIG) is the steam injection process used in GE combustion turbines. For STIG cycles, the steam source is specifically provided by the HRSG to increase both cycle efficiency and power output.[24] One study modeled the performance of a GE Frame 6b simple cycle unit that was retrofitted with a STIG cycle, and results suggest that efficiency can be increased from 30 percent to 40 percent and that power output can be increased from 38 to 50 MW.[25] The relative improvements suggested by this study are similar to estimates from GE. GE advertises that its LM2500 aeroderivative combustion turbine can improve power output by 25 percent when outfitted with a STIG cycle.[26] STIG uses a constant pressure HRSG and operation is limited to near full load when thermal load is relatively constant. The exhaust temperature drops at partial loads and the HRSG cannot maintain a balanced heat transfer.

Mitsubishi Power's Smart-AHAT (Advanced Humid Air Turbine) is a steam injection system that achieves near-zero water makeup using an integrated water recovery system. The system is potentially less complex and more flexible than combined cycle systems, with efficiencies and outputs significantly higher than comparable conventional simple cycle plants. The HRSG involved in the system is a conventional single-pressure unit that produces the steam required for the combustion turbine's steam injection. An important benefit of the Smart-AHAT system is water preservation. Without a water recovery system (WRS), large amounts of water in the form of steam would be lost to the atmosphere through the HRSG stack. Smart-AHAT uses a direct, spray-type heat exchanger to reduce the HRSG exhaust gas temperature below the water dew point of the flue gas and causes condensation of the water vapor. Some condensate is recirculated to the spray nozzles of the heat exchanger while the rest is treated and returned as feed water for the HRSG steam production.

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[22] Bahrami, S., et al (2015). *Performance Comparison between Steam Injected Gas Turbine and Combined Cycle during Frequency Drops*. Energies 2015, Volume 8. https://doi.org/10.3390/en8087582.
[23] Mitsubishi Power. *Smart-AHAT (Advanced Humid Air Turbine*. Accessed at https://power.mhi.com/products/gasturbines/technology/smart-ahat.
[24] Bouam, A., Aissani, S., Kadi, R. (2019). *Gas Turbine Performances Improvement using Steam Injection in the Combustion Chamber under Sahara Conditions.* Oil and Gas Science and Technology. Institut Français du Pétrole (IFP), 2008, 63 (2), pp.251-261. 10.2516/ogst:2007076.
[25] Wang, F. J., Chiou, J. S. (2002). *Performance improvement for a simple cycle gas turbine GENSET – a retrofitting example*. Applied Thermal Engineering 22 (2002) 1105-1115. Accessed at https://citeseerx.ist.psu.edu/viewdoc/download?doi=10.1.1.583.9680&rep=rep1&type=pdf.
[26] GE Power & Water. Accessed at https://www.ge.com/gas-power/products/gas-turbines/.product-spec-table.

Innovative Steam Technologies (IST) offer a once through HRSG steam injection for new combustion turbines or retrofits to existing simple cycle turbines. Combustion turbines are generally designed to accept up 5 percent of the compressor air flow as superheated steam and some designs are capable of accept up to 10 percent. At an injection rate of 3.5, the heat rate is reduced by 3.7 percent and output is increased by 8.4 percent.[27] Due to differences in materials and design, the equipment needed for steam injection is around 60 percent smaller than a similar drum type HRSG, which can make retrofitting steam injection easier.[28] IST reports turnkey costs inclusive of auxiliary equipment is approximately $250/kW. Retrofits to existing simple cycle turbines requires the addition of the once through HRSG, a demineralized water system, new and updated controls, valves, nozzles, and piping.

The Cheng Cycle uses a variable-pressure HRSG and the operating range is from idle to full load.[29] Performance measurements indicate that implementing a Cheng Cycle system on a combustion turbine can provide up to a 26 percent efficiency improvement compared to the base turbine engine.[30]

2.2 Hybrid Combustion Turbine-Battery Systems

Co-location of energy storage with new combustion turbine EGUs offers multiple potential environmental benefits and additional value streams. Hybrid combustion turbine-battery systems can provide power within 1 second and increase the spinning reserve capacity of the EGU. They also can allow the EGU to balance the grid and absorb excess grid power. Specifically, energy storage allows for operational changes so that combustion turbines can minimize starts and stops, reduce fuel consumption, optimize power output, and operate more continuously at optimal efficiency, all of which reduce GHG emissions and O&M costs. In contrast to standalone energy storage systems, the co-location with power generation can reduce transmission constraints by locating close to end users and charge during periods of low demand by the transmission grid. Like co-located renewables, co-located energy storage shares development costs and capital expenditures for permitting, siting, infrastructure, and grid interconnections and associated transmission and distribution capabilities.[31] Other benefits include shorter project schedules, power for ancillary services, and increased black start capabilities as an alternative to diesel.

An example of the successful integration of short-term storage with simple cycle turbines can be seen at two 50-MW peaking plants operated by Southern California Edison (SCE). In 2017, the utility's stations in Norwalk and Rancho Cucamonga began operating the world's first Hybrid Electric Gas Turbine

[27] McArthur, J., Brady, M. (2002). *Gas Turbine Performance Enhancement with Once Through Heat Recovery Steam Generators.* Innovative Steam Technologies. https://otsg.com/ist-uploads/2019/09/WP0203.pdf.
[28] Ibid
[29] Ganapathy, V., Heil, B., Rentz, J. (1988). *Heat Recovery Steam Generator for Cheng Cycle Application*. Industrial Power Conference, PWR, Vol. 4. Accessed at http://v_ganapathy.tripod.com/cheng.pdf.
[30] Digumarthi, R., Chang, C. (1984). *Cheng-Cycle Implementation on a Small Gas Turbine Engine.* Journal of Engineering for Gas Turbines and Power. Volume 106, Issue 3. https://doi.org/10.1115/1.3239626.
[31] Gorman, W., Mills, A., Bolinger, M., Wiser, R., Singhal, N., Ela, E., & O'Shaughnessy, E. (2020). Motivations and options for deploying hybrid generator-plus-battery projects within the bulk power system. *The Electricity Journal, 33*(5). https://doi.org/10.1016/j.tej.2020.106739.

systems, or Hybrid EGT.[32, 33] The plants' energy storage comes from co-located 10-MW/4.3-MWh lithium-ion batteries that pull excess renewable energy from the grid. The stored energy serves as spinning reserves, giving the turbines time to ramp up, if necessary. According to project developers, this Hybrid EGT system alleviates operational stress and reduces maintenance costs by reducing the number of starts and reduces onsite GHG emissions because the turbines no longer need to operate as often. Systemwide fuel savings and other emission reductions can also be achieved due to the system operator not needing other thermal units to operate at less efficient part load conditions to provide spinning reserve headroom.

2.3 Pressure Gain Combustion

Pressure gain combustion (PGC) has the potential to increase combustion turbine EGU efficiency and reduce emissions. Estimates for higher efficiencies could reach 4 to 6 percent for simple cycle systems and 2 to 4 percent in combined cycle systems. In conventional combustion turbines, engines undergo steady, subsonic combustion that results in a total pressure loss. In PGC, multiple physical phenomena, such as resonant pulsed combustion, constant volume combustion, and detonation can be used to create a rise in effective pressure across the combustor while consuming an equal quantity of fuel.[34] The U.S. Department of Energy (DOE) assessed the inclusion of PGC in combined cycle power plants. The study found that a PGC-integrated system produced 3.09 percent more power at the same fuel flow rate and reduced the cost of electricity (COE) by 0.58 percent.[35] One key advantage of PGC technology is that it can be compounded with other combustion turbine technology improvements such as compressor efficiency. Applications of PGC hold promise toward the Advanced Turbine Program's efficiency goals.[36] The DOE's integrated PGC system achieved a net lower heating value (LHV) efficiency of 64.56 percent, while a PGC system that included other combustion turbine technology improvements achieved a LHV efficiency of 66.68 percent.

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[32] Aoyagi-Stom, C. (18 April 2017). SCE unveils world's first low-emission hybrid battery storage, gas turbine peaker system. *Energized*. Edison International. Accessed at https://energized.edison.com/stories/sce-unveils-worlds-first-low-emission-hybrid-battery-storage-gas-turbine-peaker-system.
[33] Patel, S. (1 September 2017). Two SCE gas-battery hybrid projects revolutionize peaker performance. *Power*. Retrieved August 26, 2021, https://www.powermag.com/two-sce-gas-battery-hybrid-projects-revolutionize-peaker-performance/.
[34] DOE NETL. *Pressure Gain Combustion*. Accessed at https://netl.doe.gov/node/7553.
[35] DOE (2016). *Combined Cycle Power Generation Employing Pressure Gain Combustion*. Accessed at https://www.osti.gov/servlets/purl/1356814.
[36] Neumann, Nicolai, & Peitsch, Deiter (2019). *Potentials for Pressure Gain Combustion in Advanced Gas Turbine Cycles*. Accessed at https://www.mdpi.com/2076-3417/9/16/3211.

3 Potential Efficiency Gains in Combined Cycle EGUs

3.1 Advances in Combined Cycle Operation

While many configurations of HRSGs are available to improve the efficiency of the bottoming steam cycle, several improvements have been made to other parts of the combined cycle. These include improvements to the combustion turbine engine, turbine cooling, compressors, condensers, and more. Improved performance in industry standard combined cycles has resulted from years of iterative industry innovation.

GE provides an example of the evolution of more efficient combustion turbine technology with its 7HA and 9HA designs, which operate at 60 hertz (Hz) and 50 Hz, respectively. These combustion turbines represent current "H-class" technology and feature firing temperatures greater than 1,430 °C. The H-class is an evolution of GE combustion turbines that began with the E-class and F-class combustion turbine. In general, the firing temperature has increased from the E-class (earliest iteration) to the H-class and the resulting combined cycle efficiency has increased as well. In addition, the H-class combustion turbine includes completely air-cooled hot gas paths due to advanced turbine cooling, sealing, materials, and coating. Within the 7/9HA combustion turbines, the 7/9HA.02 has increased power output compared to the 7/9HA.01 because of increased compressor inlet and turbine exit annulus areas, with an increased pressure ratio to maintain flow. It should be noted that the HA products can ramp to full plant load in less than 30 minutes, ensure ramping capability in emissions compliance of greater than 15 percent load per minute, and include fuel flexibility to operate on both gaseous and liquid fuels.[37] It should also be noted that a third generation 7HA combustion turbine, the 7HA.03, has been designed to be even more efficient, and the first two GE 7HA.03 combustion turbines have recently begun operating at the Dania Beach Clean Energy Center (DBEC) in Broward County, Florida.[38] Combustion turbine combined cycle design specifications are outlined for the 7/9HA family in Figure 3.

FIGURE 3: DESIGN SPECIFICATIONS AT ISO CONDITIONS FOR THE GE 7/9 HA COMBUSTION TURBINE FAMILY[39]

Model	No. & Type Combustion turbine	Net Plant Output (kW)	Net Heat Rate (Btu/kWh)	Net Plant Efficiency (LHV)	Net Plant Efficiency (HHV)
9HA.01 (50 Hz)	1 x 9HA.01	680,000	5,356	63.7%	57.4%
9HA.01 (50 Hz)	2 x 9HA.01	1,363,000	5,345	63.8%	57.5%

[37] Vandervort, C., Leach, D., Scholz, M. (2016). *Advancements in H Class Gas Turbines for Combined Cycle Power Plants for High Efficiency, Enhanced Operational Capability, and Broad Fuel Flexibility.* 8th International Gas Turbine Conference. 12-13 Oct. 2016. Brussels, Belgium. https://etn.global/wp-content/uploads/2018/09/ADVANCEMENTS-IN-H-CLASS-GAS-TURBINES-FOR-COMBINED-CYCLE-POWER-PLANTS-FOR-HIGH-EFFICIENCY-ENHANCED-OPERATIONAL-CAPABILITY-AND-BROAD-FUEL-FLEXIBILITY.pdf.

[38] Patel, S. (2022). *GE Debuts First 7HA.03 Gas Turbines at 1.3-GW Plant in Florida*. Power Magazine. Accessed at https://www.powermag.com/ge-debuts-first-7ha-03-gas-turbines-at-1-3-gw-plant-in-florida/

[39] GTW (2021). *2021 GTW Handbook.* Volume 36. Page 82-90. Pequot.

9HA.02 (50 Hz)	1 x 9HA.02	838,000	5,320	64.1%	57.7%
9HA.02 (50 Hz)	2 x 9HA.02	1,680,000	5,306	64.3%	57.9%
7HA.01 (60 Hz)	1 x 7HA.01	438,000	5,481	62.3%	56.1%
7HA.01 (60 Hz)	2 x 7HA.01	880,000	5,453	62.6%	56.4%
7HA.02 (60 Hz)	1 x 7HA.02	573,000	5,381	63.4%	57.1%
7HA.02 (60 Hz)	2 x 7HA.02	1,148,000	5,365	63.6%	57.3%
7HA.03 (60 Hz)	1 x 7HA.03	640,000	5,342	63.9%	57.6%
7HA.03 (60 Hz)	2 x 7HA.03	1,282,000	5,331	>64.0%	>57.6%

Notice that small increases in net plant efficiency occur by collocating two combustion turbines at one combined cycle plant.

Another advanced combustion turbine operating within the combined cycle class is the Siemens HL combustion turbine. The "HL" terminology indicates that the current technology is an intermediate between the H-class technology and the L-class technology of the future that will be capable of 65 percent efficiency (LHV) when employed in a combined cycle plant. The HL combustion turbine evolved from the H-class turbine with some notable improvements. Namely, the turbine inlet temperature of the HL is about 100 °C higher than that of the H-class. This has a large impact on the increase in efficiency. Additionally, a new combustion system called "Advanced Combustion system for high Efficiency" (ACE), is employed to reduce the increase in NO_X emissions resulting from the increase in inlet temperature. Moreover, the number of compressor stages is reduced from 13 to 12 while simultaneously increasing the pressure ratio for increased performance and reduced complexity. Turbine blade internal cooling features were added to accommodate the higher temperatures, which also reduces dependency on cooling air consumption. Lastly, internally cooled free-standing blades are employed in stage 4 of the turbine, as opposed to uncooled blades in stage 4 for the H-class turbine, resulting in higher power output and exhaust temperatures. Exhaust temperatures of the HL-class combustion turbine are designed to be approximately 680 °C compared to 630 °C for Siemen's H-class combustion turbine.[40]

Additionally, the DOE's Advanced Turbines Program is supporting the development of advanced turbine technologies, which includes combined cycle. The program's goal is to reach 65 percent efficiency (LHV) for combined cycle technology by conducting research on hot section components and technology, including but not limited to materials, advanced cooling, leakage control, advanced

[40] Modern Power Systems (2018). *Siemens HL: the bridge to 65%+ efficiency*. Accessed at https://www.modernpowersystems.com/features/featuresiemens-hl-the-bridge-to-65-efficiency-6045386/.

aerodynamics, and altogether new turbine design concepts. Most notable, the program hopes to develop combustors that operate at higher temperatures with lower NO_X emissions.[41] Specifically, the goal is to increase the firing temperature of combustion turbines in combined cycle plants to 3,100 °F.[42] Furthermore, it should also be noted that the DOE cited combined cycle[43] efficiency goals of 67 percent (LHV) and "long-term" goals of 70 percent efficiency (LHV) in its 2022 fiscal year congressional budget request.[44] Combined cycle power plants employing Siemens HL-class technology are currently rated at greater than 63 percent combined cycle efficiency compared to 61 percent for those plants employing H-class technology.[45]

3.2 HRSG Configurations

The design of a HRSG can impact how long it takes to start producing steam and generating power. Currently, the most efficient combined cycle EGUs utilize HRSGs with a steam reheat cycle and multi-pressure steam. A steam reheat cycle extracts and reheats steam that has been partially expanded in the steam turbine prior to expansion in the lower pressure portion of the turbine. A reheat module allows more efficient operation of the steam turbine and prevents formation of water droplets that can damage the steam turbine's lower pressure stages. The use of three discrete steam pressures (high pressure (HP), intermediate pressure (IP), and low pressure (LP)) maximizes efficiency. Each of these three sections contains separate superheater, evaporator, steam drum, and economizer modules. The HP steam section is located on the high-temperature end of the HRSG, closest to the combustion turbine exhaust duct. The LP steam section is located on the low-temperature end of the HRSG, just before the stack. This arrangement maximizes the degree of superheat (*i.e.*, the quantity of energy per pound of steam) delivered to the steam turbine. Simpler, low-cost, less-efficient HRSGs are also available in single-, double-, and 3 pressure designs and without a reheat cycle. After the energy has been extracted for steam production, the flue gas enters an economizer, which preheats the condensed feedwater recycled back to the HRSG. The final heat recovery section, which is not used on all combined cycle EGUs, is the fuel preheater. The fuel preheater preheats the fuel used for the combustion turbine engine.

While a HRSG has no moving parts, thermal inertia and rapid heating can stress the components and shorten the operating life of the unit.[46] The high-pressure drum is the most vulnerable component when subjected to rapid heating; therefore, the drum is typically heated slowly with designated hold points during startup.[47] While relatively inefficient, a dual-pressure HRSG without a reheat cycle has a simpler

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[41] U.S. Department of Energy (DOE) (2021). *Advanced Turbines*. Accessed at https://netl.doe.gov/sites/default/files/2021-10/Program-108.pdf.
[42] DOE National Energy Technology Laboratory (NETL). *Advanced Combustion Turbines*. Accessed at https://netl.doe.gov/carbon-management/turbines/act.
[43] Efficiency using LHV for combined cycles using natural gas.
[44] DOE (2022). *Department of Energy FY 2022 Congressional Budget Request*. DOE/CF-0174, Volume 3 Part 2, Page 199. Accessed at https://www.energy.gov/sites/default/files/2021-06/doe-fy2022-budget-volume-3.2_0.pdf.
[45] Gas Turbine World (2021). *2021 GTW Handbook.* Volume 36. Page 82-90. Pequot.
[46] Pasha, A. (1992). *Combined Cycle Power Plant Start-up Effects and Constraints of the HRSG*. Proceedings of ASME Turbo Expo, 1992. Power of Land, Sea, and Air. https://doi.org/10.1115/92-GT-376.
[47] Pasha, A. (1992). *Combined Cycle Power Plant Start-up Effects and Constraints of the HRSG*. Proceedings of ASME Turbo Expo, 1992. Power of Land, Sea, and Air. https://doi.org/10.1115/92-GT-376.

startup procedure and can start quicker than a more efficient 3 pressure HRSG with a steam reheat cycle. Also, an auxiliary boiler can maintain the HRSG temperature, reducing the time required for an HRSG to begin producing steam. However, the use of an auxiliary boiler decreases the overall efficiency of the combined cycle EGU.

For HRSGs, there are currently three main configurations found in industry: two-pressure non-reheat (2PNR), three-pressure non-reheat (3PNR), and three-pressure reheat (3PRH). The two-pressure (2P) versus three-pressure (3P) designations refer to the number of steam pressures in the steam cycle. A 2P steam cycle employs two steam turbines—a low pressure (LP) steam turbine and a high pressure (HP) steam turbine. Similarly, a 3P steam cycle employs three steam turbines where one is an LP turbine, one is a HP turbine, and the third, located between the LP and HP turbines, is an intermediate (IP) pressure turbine. A 2P or 3P cycle can also employ reheating as a method to increase steam turbine efficiency. With reheating, steam is routed back to the HRSG to be reheated prior to further expansion through subsequent lower pressure turbines.

A HRSG can also include duct burners, sometimes called supplemental firing. Supplemental firing is the mixing of additional fuel to turbine exhaust—which still contains available oxygen to support additional combustion. The combustion of this supplemental fuel increases the useful thermal output of the HRSG and is typically only done during periods of high electric demand. While the use of duct burners can increase output during critical periods, they reduce the overall efficiency of the combined cycle EGU. Since the additional fuel is only using the bottoming Rankine cycle, incremental efficiencies are on the order of a simple cycle turbine. Typically, duct burners are categorized as either small or large based on duct size, spacing, and design constraints. Small duct burners are intended to make up capacity that is lost during periods of high ambient temperatures. Small duct burners only impact efficiency while operating. In contrast, combined cycle designs with large duct burners oversize the steam turbine relative to the output that can be provided by the combustion turbine engine. The use of large duct burners provides significant additional capacity. However, since the steam turbine is more often operating at partial load and is less efficient, the combined cycle efficiency is impacted even when the duct burners are not operating.

An alternative to the use of duct burners is complementary firing. Complementary firing combines a relatively small combustion turbine(s)[48] with a larger combined cycle facility. The small turbine is generally used during periods when the steam turbine is not operating at capacity (*e.g.*, during periods of high ambient temperatures that often correspond to periods of peak electric demand). The exhaust from the smaller turbine is sent to the HRSG of the combined cycle EGU. In essence, the smaller combustion turbine is a combined cycle EGU that is used for peaking applications. The benefits of complementary firing are that the incremental electricity is generated more efficiently than by using duct burners or from a standalone simple cycle turbine and the exhaust from the small combustion turbine is routed through the post-combustion control technology of the larger combined cycle EGU. An additional advantage of complimentary firing compared to the use of duct burners is that because most of the incremental

[48] The complimentary fired combustion turbine engines would be sized such that the turbine exhaust could be accommodated by the HRSG. This generally limits the size of the complimentary turbine engine(s) to less than 10% of the output of the primary turbine engine(s).

electricity is generated by the turbine engine, there is potentially less demand placed on the Rankine cycle portion of the larger combined cycle EGU. Drawbacks of complimentary firing compared to the use of duct burners are higher capital costs, less fuel flexibility (duct burners can burn a variety of fuels), and more limited part-load performance.[49]

4 Potential Efficiency Gains in the Bottoming (Rankine) Cycle

The primary differences between a 2PNR, 3PNR, and 3PRH HRSGs are efficiencies and construction costs.[50] The complexity and costs increase with the number of steam pressures. However, increasing the number of steam pressures allows more energy to be extracted from the exhaust gas, improving overall efficiency. A reheat cycle adds additional complexity and capital costs but increases the efficiency of the Rankine cycle by increasing the average temperature of the heat addition within the process.[51] These capital costs can at least be partially offset by reductions in fuel costs. For 2P and 3P HRSGs without reheat cycles, the efficiencies are approximately 20 and 26 percent, respectively. A 3P HRSG with a reheat cycle improves the efficiency of thermal energy to electrical output to approximately 30 percent.

According to *Gas Turbine World*, all aeroderivative and frame combined cycles with base load ratings of less than 500 MMBtu/h use 2P HRSGs. Meanwhile, 3P HRSGs without a reheat cycle are used for frame combined cycle EGUs up to 2,000 MMBtu/h, and 3P HRSGs with a reheat cycle are used for frame combined cycle EGUs with base load ratings of greater than 2,000 MMBtu/h. From a practical standpoint, the use of a reheat cycle is limited to combustion turbine engines with exhaust temperatures greater than 593 °C and for steam turbines greater than 60 MW.[52] However, 3P HRSGs have been applied to aeroderivative combined cycle EGUs and could be adopted on smaller frame combined cycle EGUs as well.[53]

Several studies have compared various HRSG configurations for combined cycle EGUs. One study directly compared 2PNR, 3PNR, and 3PRH steam cycles. It concluded that increasing the number of pressure cycles leads to an increase in efficiency of the entire cycle. Additionally, the study concluded that although increasing steam generation pressure levels requires a larger upfront investment, it ultimately yields a higher return, and the net present value (NPV) of the higher-pressure level plants (*i.e.*, 3PRH) increases. The study concluded that the estimated NPV of a 3PNR or 3PRH plant increases by 0.03 and 7 percent, respectively, when compared to the NPV of a 2PNR plant.[54] Figure 4 shows the costs and efficiencies with more complex HRSG configurations compared to one with a 2PNR HRSG.

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[49] In order to achieve part-load capabilities with complimentary firing multiple smaller turbines would be required.
[50] GTW (2021). *2021 GTW Handbook.* Volume 36. Pages 27-28. Pequot.
[51] Rashidi, M. M., Aghagoli, A., Ali, M., *Thermodynamic Analysis of a Steam Power Plant with Double Reheat and Feed Water Heaters*. Advances in Mechanical Engineering. Volume 2014, Article ID 940818, 11 pages. https://doi.org/10.1155%2F2014%2F940818
[52] Chase, D.L. and P.T. Kehoe, *GE Combined-Cycle Product Line and Performance*. GE Power Systems. GER-3574G. Accessed at: https://hi.dcsmodule.com/js/htmledit/kindeditor/attached/20220402/20220402143103_85047.pdf
[53] https://www.ijert.org/off-design-performance-analysis-of-a-triple-pressure-reheat-heat-recovery-steam-generator
[54] Mansouri, M. T., Ahmadi, P., Kaviri, A. G., Jaafar, M. N. M. (June 2012). *Exergetic and economic evaluation of the effect of HRSG configurations on the performance of combined cycle power plants.* Energy Conversion and Management. Volume 58. Pages 47-58. https://doi.org/10.1016/j.enconman.2011.12.020.

Figure 4: Relative Efficiencies and Costs of Combined Cycle with Various HRSG Configurations

HRSG Configuration	Combined Cycle Net Efficiency	Increase in Combined Cycle Efficiency Relative to 2PNR	Combined Cycle Cost ($/kW)	Increase in Combined Cycle Cost Relative to 2PNR
2PNR	56.06%	-	520.1	-
3PNR	56.22%	0.29%	530.5	2.0%
3PRH	57.15%	1.9%	540.6	3.9%

It should also be noted that single-pressure HRSG technology is available as well. While they are the lowest-cost and simplest HRSG design, they are also the least efficient and infrequently used in new combined cycle EGUs. The thermal efficiency of a single-pressure, no-reheat HRSG system is estimated to be 3.7 percent less than that of a comparable 2PNR system.[55] One study estimated the efficiencies and electricity costs of single-, dual-, and 3 pressure HRSGs. It found that when compared to single-pressure HRSGs, dual-pressure and 3 pressure HRSGs resulted in combined cycle efficiencies increasing by 4.5 and 7.2 percent, respectively. Moreover, the study estimated the cost of electricity from combined cycles utilizing single-pressure, dual-pressure, and 3 pressure HRSGs to be $48.13/MWh, $46.39/MWh, and $45.79/MWh, respectively. According to this study, utilizing a dual-pressure HRSG may result in a 3.6 percent electricity cost reduction compared to single-pressure HRSG utilization, and utilizing a 3 pressure HRSG may result in a 4.9 percent electricity cost reduction compared to single-pressure HRSG utilization.[56] Figure 5 shows the costs and efficiencies with more complex configurations compared to one with a single-pressure HRSG.

Figure 5: Relative Efficiencies and COE with Various HRSG Configurations

HRSG Configuration	CC Net Efficiency	Increase in Combined Cycle Efficiency Relative to 1-pressure	Total Capital Requirement (TCR) (million $)	Increase in TCR relative to 1-pressure	COE ($/MWh)	Decrease in COE Relative to 1-pressure
1-pressure	50%	-	116.1	-	48.13	-
2-pressure	52.25%	4.5%	119.3	2.76%	46.39	3.62%
3-pressure	53.6%	7.2%	129.9	11.89%	45.79	4.86%

4.1 Heat Recovery Steam Generation Design Optimization

For a given HRSG design, parameters can be thermodynamically optimized to achieve the maximum overall efficiency. Optimization of HRSG performance can identify a best-case scenario for which

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[55] Chase, D.L. and P.T. Kehoe, *GE Combined-Cycle Product Line and Performance*. GE Power Systems. GER-3574G

[56] Zhao, Y., Chen, H., Waters, M., Mavris, D. N. (2003). *Modeling and Cost Optimization of Combined Cycle Heat Recovery Generator Systems*. Proceedings of ASME Turbo Expo, 2003. Power of Land, Sea, and Air. https://doi.org/10.1115/GT2003-38568.

similar-designed HRSGs could be operated. Examples of design parameters include, but are not limited to, the pinch point temperature difference,[57] inlet gas temperature, exit gas temperature, pressure within the turbine(s), mass flow rate, heat transfer area, pipe/tube/steam materials, condenser and cooling tower heat transfer surface area, steam turbine exhaust annulus area, external insulation to extract additional useful thermal output while maintaining the flue gas above the flue gas temperature, etc. Studies have both thermodynamically and economically optimized HRSG performance and development.

One study thermodynamically optimized the parameters within HRSGs for single-, double-, and 3 pressure turbine use. The results indicate that single-, double-, and 3 pressure HRSGs can increase combined cycle power output by 0.05, 0.28, and 0.29 percent, respectively, for every 10-bar inlet pressure increase. Furthermore, it found that the net combined cycle power output will decrease by 0.54, 0.21, and 0.17 percent for every 10 °C evaporator pinch point temperature difference.[58] The results suggest that a significant performance increase can result from choosing optimum operating conditions for a given HRSG. Additionally, the findings suggest that single-pressure HRSGs are most susceptible to efficiency decrement for suboptimum operation, and 3 pressure HRSGs have the most potential for improvement through optimization.

In addition, integrated fuel gas heating results in higher turbine efficiency due to the reduced fuel flow required to raise the total gas temperature to firing temperature. Fuel heating occurs before the fuel is fed into the combustion chamber of the combustion turbine and can be carried out by using the heat of the exhaust gases of the combustion turbine. Heating fuel gas from a base temperature of 0 °C to a temperature of 450°C increases combustion turbine efficiency from 35.05 to 35.39 percent.[59]

4.2 Intercooled Combined Cycle

Intercooling is a concept that is being used in the latest combustion turbine systems. In simple cycle systems, intercooling is used to improve the overall efficiency and reduce the compression work by cooling the hot gases to atmospheric temperature. The energy of the hot water at the intercooler outlet is lost to the atmosphere. In a combined cycle combustion turbine this energy could be used to heat the feed water to the HRSG. In a combined cycle plant, the feed water entering the HRSG must have a higher temperature than the dew of the acid vapor of sulfur. The application of the intercooler as the feed water heater of the HRSG increases the overall efficiency of the combined cycle as it reduces the compression work in the upper cycle. An increase of feed-water temperature from 20 °C to 60°C could increase the overall efficiency by approximately 2 percent.[60]

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[57] The pinch point temperature difference is the difference between the gas temperature leaving the evaporator section and the temperature of the fluid entering the evaporator section.
[58] Rahim, M. A. (September 2012). *Combined Cycle Power Plant Performance Analyses Based on the Single-Pressure and Multipressure Heat Recovery Steam Generator*. Journal Of Energy Engineering. Volume 138, Issue 3. https://doi.org/10.1061/(ASCE)EY.1943-7897.0000063.
[59] Marin, G. et al. (2020). *Study of the effect of fuel temperature on gas turbine performance*. Accessed at https://www.e3s-conferences.org/articles/e3sconf/abs/2020/38/e3sconf_hsted2020_01033/e3sconf_hsted2020_01033.html.
[60] Shukla, P., et al (2010). *A Heat Recovery Study: Application of Intercooler As A Feed-Water Heater of Heat Recovery Steam Generator*. Accessed at https://asmedigitalcollection.asme.org/IMECE/proceedings-abstract/IMECE2010/44298/611/357134.

4.3 Blowdown Heat Recovery

In combined cycle combustion turbines, the concentration of impurities in the steam flow must be controlled to prevent corrosion of the steam turbine blades.[61] A portion of saturated water is continuously drained through boiler blowdown where it is discharged to the outside environment through a steam vent or drain flow. This process wastes energy and decreases the efficiency and net generated power of the cycle. Waste heat from the boiler blowdown stream can be recovered with a heat exchanger, a flash tank, or a combination of both.[62] In a flash tank, the pressure can be lowered to allow a portion of the blowdown to be converted into low-pressure steam, which can be used in the cycle again as a heat source to preheat the feed water. The recovery of the wasted heat contributes to an increase in net power and energy efficiency of the Rankine cycle, as well as a reduction in annual water usage.[63] The usage of a flash tank could increase the net power and the energy efficiency of the Rankine cycle by 0.23 percent, respectively. Since about one-third of the output from a combined cycle is from the Rankine cycle, blowdown heat recovery could increase the output of the combined cycle EGU by 0.24 percent and the absolute efficiency by 0.077 percent.

4.4 Design and Operating and Maintenance Practices

While several state-of-the-art turbines and design alterations exist for new combined cycle turbines to maximize efficiency, efficiency can also be gained for existing combined cycle turbines by proper maintenance and reparations/reinstallation of various working components. All major manufacturers offer packages for plants to uprate, and these include improvements to seals, vanes, blades, and other materials within a plant. GE offers improved wire brush seals which can act as an alternative to both labyrinth seals for compressor shafts and high-pressure packing seals. Replacing the labyrinth and/or high-pressure seals can result in output increases of 1 and 0.3 percent, respectively, and heat rate increases of 0.5 and 0.2 percent, respectively. Moreover, advanced materials can reduce the need to cool turbine blades, or steam cooling of turbine blades can be used to recover the steam in a closed loop. Other options resulting in improvements for the power generation process include advanced coatings of turbine blades and combustor components, replacement of combustion liners, replacement of turbine vanes/blades, and inlet-air fogging.[64] **Figure 6** outlines the capacity and heat rate impacts and the corresponding capital costs for various turbine upgrades.

FIGURE 6: COMPARISON OF VARIOUS TURBINE UPGRADE OPTIONS[65]

[61] Saedi, Ali, et al (2022). *Feasibility study and 3E analysis of blowdown heat recovery in a combined cycle power plant for utilization in Organic Rankine Cycle and greenhouse heating*. Accessed at https://www.sciencedirect.com/science/article/pii/S0360544222019600.

[62] DOE (2012). *Recover Heat from Boiler Blowdown*. https://www.energy.gov/eere/amo/articles/recover-heat-boiler-blowdown#:~:text=Heat%20can%20be%20recovered%20from,occur%20with%20high%2Dpressure%20boilers.

[63] Vandani, Amin, et al (2015). *Exergy analysis and evolutionary optimization of boiler blowdown heat recovery in steam power plants.* Accessed at https://www.sciencedirect.com/science/article/pii/S0196890415008535.

[64] Andover Technology Partners (2018). *Improving Heat Rate on Combined Cycle Power Plants*. Accessed at https://www.andovertechnology.com/wp-content/uploads/2021/03/C_18_EDF_FINAL.pdf.

[65] Andover Technology Partners (2018). *Improving Heat Rate on Combined Cycle Power Plants*. Accessed at https://www.andovertechnology.com/wp-content/uploads/2021/03/C_18_EDF_FINAL.pdf.

Combustion Turbine Upgrade Option	MW Increase (%)	Heat Rate Impact (%)	Capital Cost ($/kW)[66]
Comprehensive Upgrade[67]	10-20	1-5	150-250
High-Flow Inlet Guide Vanes	4.5	1	<100
Hot Section Coatings	5-15	0.5-1	50-100
Compressor Coatings	0.5-3	0.5-3	50
Inlet-Air Fogging	5-15	1-5	50-100
Supercharging Plus Fogging	15-20	4	200

Proper cleaning of HRSG components can also have worthwhile impacts on turbine performance as it can maintain low pressure drop across the HRSG. Various contaminants, most notably ammonium bisulfate, can accumulate in the HRSG and can produce pressure losses. In one case study on HRSG cleaning, GE removed 14 tons of debris, resulting in a reduced turbine back pressure of 8 inches water column. The combined annual fuel savings and additional power output are believed to have netted the facility $500,000/year in avoided costs/additional revenue.[68] Similarly, plant condensers should be regularly cleaned. Airborne dust and debris can accumulate and degrade condenser performance.[69] Note that turbine overhauls can range from $2 to $12 million for 200-MW turbines but could provide heat rate improvements of 100 to 300 Btu/kWh, which represents approximately 1 to 3 percent of the steam cycle. Additionally, proper O&M practices can reduce heat rates by approximately 30 to 70 Btu/kWh (~0.3 to 0.7 percent of the steam cycle) for a cost of $30,000 annually, and feed pump rebuilds can improve the steam cycle heat rates by 0.25 to 0.5 percent for costs of $250,000 to $350,000.[70,71]

As it relates to the steam system, there are several operational practices that can reduce the heat losses within the system. Some of these methods are outlined as follows[72]:

- Minimize airin-leakage
- Clean HRSG heat transfer surfaces
- Improve water treatment to minimize HRSG blowdown
- Recover energy from HRSG blowdown

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[66] Costs shown in 2002 dollars.
[67] May include “replacement of combustion liners, transition pieces, 1st stage turbine vanes, and 2nd stage vanes and blades with [GE] Frame 7EA parts.”
[68] GE (2017). *When is 28,000 pound pile of rust a good thing?*. Accessed at https://www.ge.com/content/dam/gepower-new/global/en_US/downloads/gas-new-site/services/hrsg-services/pressurewave-case-study.pdf.
[69] Andover Technology Partners (2018). *Improving Heat Rate on Combined Cycle Power Plants*. Accessed at https://www.andovertechnology.com/wp-content/uploads/2021/03/C_18_EDF_FINAL.pdf.
[70] Costs given in 2008 dollars.
[71] Sargent & Lundy (2009). *Coal-Fired Power Plant Heat Rate Reductions*. SL-009597. Final Report. Accessed at https://www.epa.gov/sites/default/files/2015-08/documents/coalfired.pdf.
[72] Andover Technology Partners (2018). *Improving Heat Rate on Combined Cycle Power Plants*. Accessed at https://www.andovertechnology.com/wp-content/uploads/2021/03/C_18_EDF_FINAL.pdf.

- Add/restore HRSG and steam plant insulation
- Optimize deaerator vent rate
- Repair steam leaks
- Minimize vented steam
- Ensure that steam system piping, valves, fittings, and vessels are well insulated
- Implement an effective team-trap maintenance program
- Isolate steam from unused lines
- Optimize condensate recovery
- Clean combustion turbine flow path components

4.5 Once-Through (Benson®) HRSG Technology

The use of a once-through (*i.e.*, Benson®) HRSG can also improve the ability of a combined cycle EGU to start quickly and maintain efficiency at part load. A once-through HRSG does not have a steam drum like a traditional HRSG. Instead, the feedwater is converted to steam in the HRSG furnace waterwalls and goes directly into the steam turbine. This allows for the use of higher pressure steam, which improves design efficiencies, provides higher part-load efficiencies, allows reduced startup times, and results in more flexible operation.

The Benson Technology is touted as "a proven process for large-scale steam generation in power plants with the heart of this process being the once-through principle. Combined with sliding pressure operation, this allows for highly efficient, flexible, and reliable power plant operation."[73]

Advantages of the Benson Once-Through HRSG:[74]73

- It retains all the virtues of the proven natural circulation principle of drum-type boilers (*i.e.*, flow stability and uniform temperature distribution), yet at the same time replaces the high-pressure drum with thin-walled components to improve operating flexibility.
- Significant shortening of plant startup time by allowing unrestricted combustion turbine startup.
- Increase of efficiency during startup by minimizing combustion turbine operation in part loads.
- Reduction of gaseous and liquid emissions through shorter startup process and elimination of drum blowdown.
- Reduced consumption of chemicals through advanced feedwater treatment.
- Improved efficiency at high ambient temperatures due to adjustable evaporating point.

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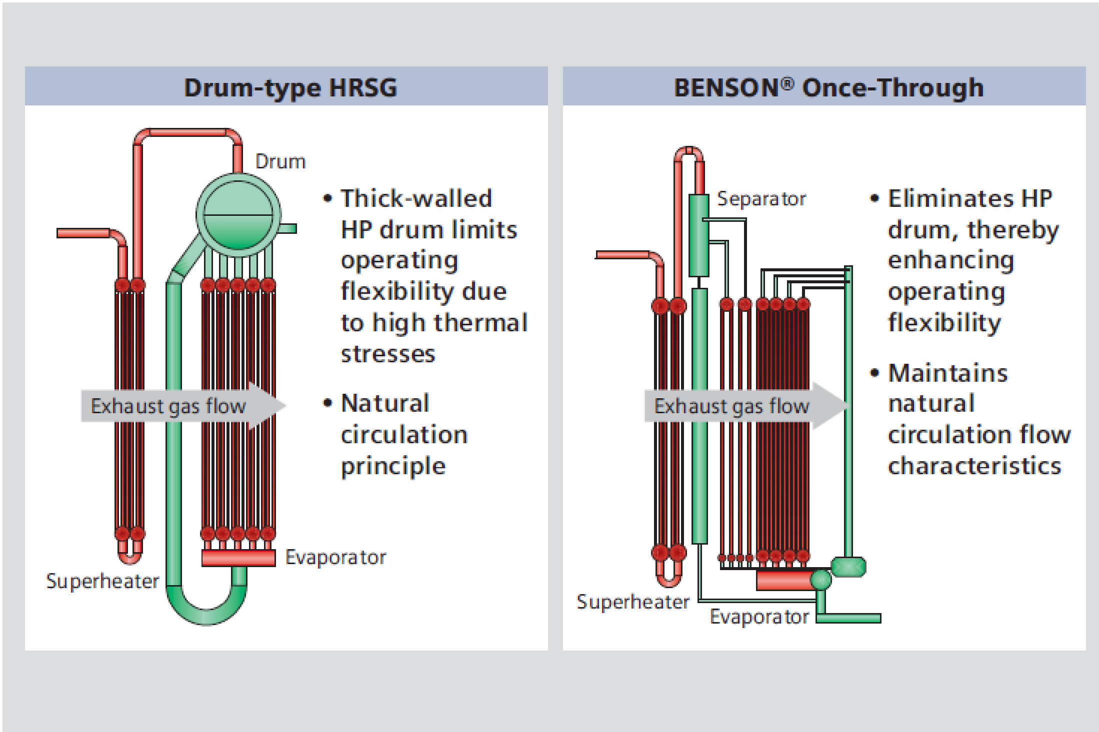
[73] https://www.siemens-energy.com/global/en/offerings/power-generation/power-plants/benson-technology.html ; https://assets.siemens-energy.com/siemens/assets/api/uuid:b5c2c3b8-eb59-430b-8065-ba09d31eb37b/flyer-benson-hrsg-210920.pdf ; and https://assets.siemens-energy.com/siemens/assets/api/uuid:ef5fb27a-d2e0-4222-a0d0-2cd24146a937/new-benson-evaporator.pdf

[74] https://www.siemens-energy.com/global/en/offerings/power-generation/power-plants/benson-technology.html ; https://assets.siemens-energy.com/siemens/assets/api/uuid:b5c2c3b8-eb59-430b-8065-ba09d31eb37b/flyer-benson-hrsg-210920.pdf ; and https://assets.siemens-energy.com/siemens/assets/api/uuid:ef5fb27a-d2e0-4222-a0d0-2cd24146a937/new-benson-evaporator.pdf

- Capability for higher steam parameters (pressure and temperature) because there are no limitations through natural circulation.



The BENSON® system – Elimination of thick-walled components

FIGURE 7: FROM SIEMENS "BENSON® ONCE-THROUGH HEAT RECOVERY STEAM GENERATOR" BROCHURE, PG. 3 (2006)

4.6 <u>The Use of Supercritical Steam Conditions</u>

Combined cycle EGUs typically have HRSGs that operate at subcritical steam conditions. However, once-through HRSGs can be designed to operate using supercritical steam conditions. "Supercritical" is a thermodynamic term describing the state of a substance in which there is no clear distinction between the liquid and the gaseous phase (*i.e.*, they are a homogenous fluid). In contrast to a subcritical steam generator, a supercritical steam generator operates at pressures above the critical pressure—3,200 psi (22 MPa). Combustion turbine engines larger than approximately 200 MW typically have exhaust temperatures high enough to support the use of supercritical steam conditions. However, the steam turbine in combined cycle configurations where one turbine engine is paired with a steam turbine (1-1 configuration) is smaller than typical EGUs using supercritical steam conditions. Steam turbine sizes in combined cycle configurations where two or three large turbines are paired with a single steam turbine (2-1 or 3-1 configuration) are as large as typical EGUs using supercritical steam conditions.

Thermodynamic modeling has been applied to assess the potential of using supercritical steam as the working fluid in the HRSG. One study suggests that using a supercritical steam once-through HRSG will increase steam power by 5 percent when compared to using a subcritical steam HRSG. Since the steam turbine typically makes up approximately one-third of the overall output of a combined cycle EGU, if a combined cycle EGU were designed to use supercritical steam conditions in the high-pressure portion of the steam turbine, it would reduce overall fuel use by 2 percent.[75] Another study analyzed the improvement of a 3PRH combined cycle EGU when using supercritical steam as opposed to subcritical steam. It indicates that if using supercritical steam as the working fluid for the HP turbine, it is possible to obtain a plant efficiency of 64.45 percent and capacity of 1.214 GW power output, compared to 63.08 percent and 1.19 GW when using subcritical steam. Additionally, the economic analysis predicts that plants can return up to an additional $14 million per year when considering the difference between the annual revenue from electricity sales and annual fuel costs when using supercritical steam.[76]

Another study compared the use of 2P and 3P cycles using subcritical and supercritical steam conditions with and without steam reheat. The results followed the patterns such that efficiency increased from 2P to 3P, from non-reheat to reheat, and from subcritical to supercritical. The analyses were conducted on a Siemens V94.3 combined cycle gas combustion turbine, and the findings are outlined in Figure 8.[77]

FIGURE 8: RESULT OF THERMODYNAMIC ANALYSIS OF SIEMENS V94.3 COMBINED CYCLE GAS COMBUSTION TURBINE EFFICIENCIES[77]

HRSG Cycle	HP-Pressure (bar)[a]	Net LHV Combined Cycle Efficiency (%)	Relative Combined Cycle Efficiency Increase (%)
2PNR	80	53.6	-
2PRH	140	54.0	0.75
3PNR	100	54.1	0.93
3PRH	140	54.6	1.87
2PRH - Supercritical	250	54.6	1.87
3PRH - Supercritical	260	55.1	2.80

[a] Pressures are provided as “reasonable” choices for each HRSG cycle type

While combined cycle efficiencies are routinely above 55 percent on a higher heating value (HHV) basis, the result of the Bolland (1990) study still carries important implications for the comparisons of 2PNR, 3PNR, and 3PRH HRSGs. Additionally, it’s useful to compare the three HRSG types with subcritical and supercritical steam as the working fluid. As shown in Figure 3, the use of supercritical steam appears to be an important option for increasing efficiency, with the efficiency of a dual-pressure supercritical reheat HRSG being equal to that of a 3pressure reheat.

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[75] Alobaid, Falah & Ströhle, Jochen & Epple, Bernd & Kim, Hyun-Gee (2009). *Dynamic simulation of a supercritical once-through heat recovery steam generator during load changes and start-up procedures.* Applied Energy, Elsevier, vol. 86(7-8), pages 1274-1282, July. https://ideas.repec.org/a/eee/appene/v86y2009i7-8p1274-1282.html.
[76] Marcin Jamróz, Marian Piwowarski, Paweł Ziemia´nski, and Gabriel Pawlak (2021). *Technical and Economic Analysis of the Supercritical Combined Gas-Steam Cycle*. Energies 2021, 14, 2985. https://www.mdpi.com/1996-1073/14/11/2985
[77] Bolland, Olav (1990). *A Comparative Evaluation of Advanced Combined Cycle Alternatives*. American Society of Mechanical Engineers (ASME). https://doi.org/10.1115/90-GT-335.

4.7 The Use of Alternate Working Fluid

In addition, alternate working fluids—such as organic fluids, supercritical CO_2 (sCO_2), or ammonia/water mixtures rather than steam—also have the potential to increase the efficiency of combined cycle EGUs. Organic Rankine cycles are primarily applicable to temperatures lower than combustion turbine engine exhaust temperatures.[78] While the use of sCO_2 as the working fluid in a Rankine cycle is of most interest for nuclear and coal-fired EGUs, it also has the potential to improve the overall efficiency of combined cycle EGUs.[79] The primary efficiency benefit would be for combined cycle EGUs using smaller frame or aeroderivative combustion turbine engines that typically use a double-pressure HRSG without a reheat cycle.[80] However, a HRSG using sCO_2 has the potential to improve the efficiency of combined cycle EGUs compared to 3 pressure steam with a reheat cycle as well.[81]

The potential of sCO_2 has been assessed in multiple studies. One study found that sCO_2 potentially has the advantages of being more compact and higher efficiency compared to steam-based combined cycle technology. Additionally, when including O&M costs, calculations demonstrated that sCO_2 can provide levelized cost of electricity (LCOE)[82] advantages as well. When comparing different steam turbine models, the LCOE decreased by an average of 15 percent when using sCO_2 versus subcritical steam.[83] Another study modeled the performance of sCO_2 use versus 2PNR and 3PRH alternative HRSG use. It found that when compared to steam for bottoming cycles for 2PNR, sCO_2 as a working fluid has better performance at pressures above 200 bar. However, when compared to 3PRH, assuming high sCO_2

[78] The Kalina Cycle® is another cycle that has the potential for efficiency gains compared to a water-based Rankine cycle. See http://www.kalinapower.com/technology/.

[79] Patel, S. (2021b, October 27). The POWER interview: Pioneering STEP supercritical carbon dioxide demonstration ready for 2022 commissioning. *Power*. https://www.powermag.com/the-power-interview-pioneering-step-supercritical-carbon-dioxide-demonstration-readying-for-2022-commissioning/?oly_enc_id=3025B2625790F2W.

[80] Using the design HRSG efficiencies listed in Gas Turbine World and the efficiency of the design efficiency of the Echogen supercritical EPS100 heat recovery system (24 percent net, https://www.echogen.com/our-solution/product-series/eps100/), the median decrease in design heat rates for replacing dual pressure HRSG with supercritical CO_2 HRSG is 7 percent.

[81] Thanganadar, D., Asfand, F., & Patchigolla, K. (2019). Thermal performance and economic analysis of supercritical carbon dioxide cycles in combined cycle power plant. *Applied Energy, 255*(1), 113836. https://doi.org/10.1016/j.apenergy.2019.113836.

[82] Levelized cost of electricity (LCOE) is defined as the price at which the generated electricity should be sold for the system to break even at the end of its lifetime. LCOE is a good indicator of cost-effectiveness, because it can be calculated without requiring for assumptions about the price at which the electricity can be sold to the grid or to an end user, as is the case when calculating the payback period or the net present value. LCOE is an indicator that can be used to compare different technologies, without any framework conditions affecting the assessment. With the use of LCOE, the financial viability in specific conditions can be indicated by just comparing directly the LCOE with the price at which electricity could be sold. Papapetrou M., Kosmadakis G. (2022). *Salinity Gradient Heat Engines*, https://www.sciencedirect.com/topics/engineering/levelized-cost-of-electricity.

[83] Held, T (2015). *Supercritical CO_2 for Gas Turbine Combined Cycle Power Plants*. Echogen Power Systems. Power Gen International, December 8-10, Las Vegas, Nevada. https://www.echogen.com/_CE/pagecontent/Documents/Papers/Supercritical%20CO2%20Cycles%20for%20Gas%20Turbine%20Combined%20Cycle%20Power%20Plants.pdf.

expander and pump isentropic efficiencies at 95 percent, the maximum pressure of the sCO_2 cycle needs to exceed 300 bar to outperform 3PRH steam cycles.[84]

The DOE's National Energy Technology Laboratory (NETL) is working on improvements to a sCO_2 power cycle.[85] One pilot power plant was recently completed that uses sCO_2 technology.[86] In 2018, Southwest Research started building a Supercritical Transformational Electric Power (STEP) pilot plant, which will use sCO_2 technology with a design capacity of 10 MWe. It is estimated that replacing water with sCO_2 increases the efficiency by up to 10 percent. Additionally, STEP turbomachinery can be one-tenth the size of a conventional power plant's components, providing potential to lower environmental footprint and construction costs of new facilities.[87] NETL conducted a study on the use of sCO_2 in coal-fired power plants that indicates a sCO_2 power cycle can achieve higher efficiencies than a pulverized coal (PC)/Rankine systems using supercritical steam conditions with no increase in cost of electricity.[88]

Another report by *Echogen* compared the use of sCO_2 as the Rankine Cycle working fluid to that of a steam-based Rankine Cycle system. *Echogen* claims its EPS100 has up to 40 percent lower install cost per kilowatt than that of a comparable dual-pressure steam system utilizing GT-PRO/PEACE. The install cost largely results from the smaller installation footprint and simplicity of the sCO_2 system. The lower install costs contribute to a 10 to 20 percent lower LCOE of the EPS100 system compared to that of traditional dual-pressure HRSGs.[89] In Canada, Siemens Energy and TC Energy agreed to build a waste-heat-to-power facility using the EPS100 technology. Commissioned in 2022, the facility captures waste heat from a combustion turbine and converts it into power using a sCO_2 power cycle.[90]

Ammonia/water mixtures can be utilized as a working fluid through the Kalina cycle. Depending on the application, the Kalina cycle can improve power plant efficiency by 10 to 50 percent over the Rankine cycle.[91] As plant operating temperatures are lowered, the Kalina cycle experiences a higher increase in relative gain in comparison to the Rankine cycle. Advantages for the Kalina cycle include lower upfront capital costs, lower demand for cooling water and cooling infrastructure, minimal maintenance downtime, and minimal required supervision. A study found that the use of ammonia-water instead of a

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[84] Huck, Pierre, Freund, Sebastian, Lehar, Matthew, & Peter, Maxwell (2016, March 28-31). *Performance comparison of supercritical CO2 versus steam bottoming cycles for gas turbine combined cycle applications*. GE Global Research. The 5th International Symposium - Supercritical CO2 Power Cycles, http://sco2symposium.com/papers2016/SystemConcepts/092paper.pdf.

[85] https://netl.doe.gov/project-information?p=FE0028979

[86] https://www.swri.org/press-release/step-10-megawatt-supercritical-carbon-dioxide-pilot-plant-building

[87] Southwest Research Institute (SwRI) (2022). *Supercritical Transformational Electric Power Pilot Plant.* https://www.swri.org/industry/advanced-power-systems/supercritical-transformational-electric-power-pilot-plant.

[88] NETL (2019). *Supercritical Carbon Dioxide (Sco2) Cycle As An Efficiency Improvement Opportunity For Air-Fired Coal Combustion*. Accessed at https://www.osti.gov/servlets/purl/1511695.

[89] Persichilli, M., Kacludis, A., Zdankiewicz, E., Held, T. (April 2012). *Supercritical CO_2 Power Cycle Developments and Commercialization: Why sCO_2 can Displace Steam.* Echogen Power Systems LLC. Accessed at https://www.echogen.com/_CE/pagecontent/Documents/Papers/why-sco2-can-displace-steam.pdf.

[90] Power Magazine (2021). *First Commercial Deployment of Supercritical CO2 Power Cycle Taking Shape in Alberta*. Accessed at https://www.powermag.com/first-commercial-deployment-of-supercritical-co2-power-cycle-taking-shape-in-alberta/.

[91] Kalina Power (2015). *Technology: Kalina Cycle*. Accessed at http://www.kalinapower.com/technology/.

steam only cycle increased the efficiency of the system from 57.5 percent to 62.5 percent, while the cost of electricity marginally increased from $0.06718/kWh to $0.06723/kWh. A study on a system with intercooling, a 3 pressure reheat HRSG, and ammonia/water cycle at each pressure level produced a minimum cost of electricity production of 0.06723 $/kWh. The same system obtained a 62.5 percent maximum value of efficiency, a second law efficiency of 60.7 percent, and a maximum work output of 1,789.39 kJ/kg of air.[92]

4.8 The Use of Thermoelectric Materials

Combined cycle EGUs generate significant quantities of relatively low-temperature heat (*i.e.*, waste or byproduct heat) that cannot be used by the traditional Rankine cycle and is sent to the power plant cooling system (*i.e.*, cooling tower). If this energy could be recovered to produce additional electricity, it could reduce the environmental impact of power generation. Thermoelectric materials (*e.g.*, bismuth telluride (Bi_2Te_3), lead telluride (PbTe), silicon-germanium (SiGe), magnesium antimonide (Mg_3Sb_2), and magnesium bismuthide (Mg_3Bi_2)) can be used to generate electricity due to temperature differences across the material.[93,94] While still in development, this technology has the potential to recover useful energy from the waste heat from power plants. However, if a thermoelectric generator were able to convert 5 percent of combustion turbine waste heat to electric output, the CO_2 emissions rate for simple cycle EGUs would be reduced by approximately 10 percent and combined cycle EGUs by approximately 5 percent.

Currently, optimizing thermoelectric generation (TEG) power output and efficiency is very dependent on thermoelectric (TE) material properties and dimensions. Currently, TE materials are based on the use of tellurium and germanium, which are expensive elements. Consequently, development of polymer, silicide, oxide, and tetrahedrite TE materials are being explored. Challenges of commercial TEG are mainly the materials development and systems engineering.[95]

However, the potential of TEG utility has been studied and shown promising results. On a study of a ship's waste heat recovery, it was concluded that a TEG-organic Rankine cycle (ORC) method increased the waste heat utilization rate while reducing power generation costs. Results show that for a TEG/ORC bottoming cycle ratio of 0.615, the output power, thermal efficiency, and generation costs of the TEG-ORC combined cycle experimental system were estimated to be 134.50 W, 6.93 percent, and 0.461 $/kWh, respectively.[96] Another study showed the promise of bismuth-telluride-based thermoelectric

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[92] Maheshwari, M., Singh, O. (2020). *Thermo-economic analysis of combined cycle configurations with intercooling and reheating.* Accessed at https://www.sciencedirect.com/science/article/pii/S0360544220311567.

[93] Electricity can also be generated from electrochemical reactions at different temperatures and pressures, See https://jtecenergy.com/technology/. In addition, thermogalvanic cells use temperature differences to generate an electric current. (See e.g., Yuan)

[94] Yuan Yang, *et al. (2014). Charging-free electrochemical system for harvesting low grade thermal energy.* https://www.pnas.org/content/111/48/17011.

[95] LeBlanc, S. (2014). *Thermoelectric generators: Linking material properties and systems engineering for waste heat recover applications*. Sustainable Materials and Technologies. Volumes 1-2, Pages 25-35. https://doi.org/10.1016/j.susmat.2014.11.002.

[96] Kiu, C., Ye, W., Li, H., Liu, J., Zhao, C., Mao, Z., & Pan, X. (2020). *Experimental study on cascade utilization of ship's waste heat based on TEG-ORC combined cycle.* International Journal of Energy Research. https://doi.org/10.1002/er.6083.

micro-generators (μ-TEGs) when it found that a power output of 5.5 μW per thermocouple can be generated under a temperature difference of only 5 K.[97] The findings of these studies are indicative of TEGs potential to increase energy efficiency of combustion turbines.

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[97] Oualid, S. E., Kosior, F., Dauscher, A., Candolfi, C., Span, G., Mehmedovic, E., Paris, J., & Lenoir, B. (2020). *Innovative design of bismuth-telluride-based thermoelectric micro-generators with high output power.* Energy & Environmental Science. Issue 10. https://pubs.rsc.org/en/content/articlelanding/2020/EE/D0EE02579H .

5 Combined Cycle Startup Times

Improving startup time of combined cycle EGUs makes combined cycle EGUs a more dependable power source for load-following supply, and research/practice suggests several ways to improve combined cycle startup times. Combustion turbines operating as EGUs in a combined cycle system have historically been designed to operate for extended periods of time at steady loads. Since these combined cycle EGUs were not intended to start and stop on a regular basis, they had relatively long startup times depending on unit-specific factors and whether startup was initiated from a cold, warm, or hot state. During the past decade, the demands placed on this conventional mode of steady, base load operation have changed. The latest combined cycle EGUs are designed with advanced technology and features to be more flexible and respond faster to increased demand for reliable electricity, support increased generation from intermittent sources (*i.e.*, renewables), capitalize on financial incentives to improve dispatch or supply non-spinning reserves, operate at higher efficiencies, and emit less pollution. As a result, advanced fast-start, combined cycle EGUs incorporate multiple techniques that allow the EGU to start and stop faster, cycle output faster, and maintain higher part-load efficiencies than previous designs.

Several combustion turbine manufacturers market complete combined cycle systems that can ramp up to full load from a cold start in less than an hour, depending on unit-specific factors. Advanced combustion turbines, when isolated from the HRSG and steam turbine, can reach full load at full speed as a simple cycle (*i.e.*, Brayton) unit in less than 20 minutes.[98] When adhering to some of the following fast-start techniques, the HRSG, steam turbine, and balance of plant equipment can reach safe operating temperatures and pressures and begin generating additional electricity within 30 to 45 minutes of ignition of the combustion turbine. Techniques that can be used to reduce startup times for combined cycle systems are discussed below.

Slower startup times of combined cycle EGUs are largely attributed to HRSGs needing a slower and more gradual startup to reduce thermal stress in the HRSG thick-walled components, such as steam drums. During startup, a temperature gradient will exist between the inside and outside of a steam drum, leading to damage of the steam drum if not properly managed.[99] However, because the slow startup of the full combined cycle is limited by the HRSG, the combustion turbine can startup and begin producing power if the combustion turbine exhaust gas is properly managed.

One option is to employ a bypass damper to reduce the amount of exhaust gas passing through the HRSG as it warms up. The damper blocks the natural draft of cooler, ambient air back through the HRSG stack. Another practice to maintain temperature is to insulate the HRSG stack.[100] Keeping critical elements of the HRSG in a warm or hot state following shutdown is an important technique for reducing

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[98] Gulen, S.C. (2013). *Gas Turbine Combined Cycle Fast Start: The Physics Behind the Concept*. Accessed at http://www.mcilvainecompany.com/Decision_Tree/subscriber/Tree/DescriptionTextLinks/Physics.pdf.
[99] Power Magazine (2013). *Fast-Start HRSG Life-Cycle Optimization*. June 1, 2013. Accessed at https://www.powermag.com/fast-start-hrsg-life-cycle
[100] Eddington, et al. (2017). *Fast start combined cycles: how fast is fast?*. Accessed at https://www.power-eng.com/emissions/fast-start-combined-cycles-how-fast-is-fast/#gref.

startup times. Reducing the exhaust gas passing through the HRSG allows for the steam turbine to ramp up to full power without jeopardizing the thick-walled components within the HRSG.[101]

Additionally, a bypass stack allows the exhaust energy from the combustion turbine to be decoupled from the heat recovery unit and steam turbine generator. The bypass allows the combustion turbine engine—the fastest-starting component of a combined cycle system—to operate independent of the HRSG and come to partial or full load as a simple cycle EGU at a faster ramp rate. Documented start times range from approximately 10 minutes[102] for a hot start to approximately 15 to 20 minutes for a warm start and to approximately 20 to 25 minutes for a cold start.[103] The HRSG, steam turbine generator, and balance of plant piping and equipment can then be slowly brought to temperature while the combustion turbine engine operates at high load.[104] The use of preheaters to gradually warm major steam lines can add significant time to startup procedures.[105]

During a conventional startup, combined cycle turbines hold at low load for an extended time to gradually warm the HRSG and steam turbine generator components and prevent thermal stresses that can reduce the lifespan of the equipment. The elimination of this long hold is key to a fast start and may be possible with a bypass stack and a modulated damper that can control the amount of exhaust heat and flow that control the steam production rate and temperatures that reach the HRSG.[106] Fast-start, advanced class combined cycle designs may include a HRSG capable of tolerating rapid changes in temperature and flow of high-temperature exhaust generated by rapidly ramping the turbine.

Additionally, the startup time of a HRSG is largely dependent on how warm the system is already (*i.e.*, warm start vs. cold start). Maintaining warm conditions for the HRSG after shut down can result in faster startup times when ramping back up. One option to do this is with cascaded latent heat storage (CLHS), which can deploy stored thermal energy to keep the HRSG warm.[107] Note that startup times to reach full load can be significantly faster for hot startups compared to cold startups. One estimate indicates that the duration of startup for cold, warm, and hot combined cycle plants averages around

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[101] Kim, T. S., Lee, D. K., Ro, S. T. (2000). *Analysis of thermal stress evolution in the steam drum during start-up of a heat recovery steam generator*. Applied Thermal Engineering. https://doi.org/10.1016/S1359-4311(99)00081-2.

[102] Pasha, A. (1992). *Combined Cycle Power Plant Start-up Effects and Constraints of the HRSG*. Proceedings of ASME Turbo Expo, 1992. Power of Land, Sea, and Air. https://doi.org/10.1115/92-GT-376.

[103] GE (2016). *Startup time reduction for Combined Cycle Power Plants*. Accessed at https://etn.global/wp-content/uploads/2018/09/Startup_time_reduction_for_Combined_Cycle_Power_Plants.pdf.

[104] Previous combined cycle designs had to operate the combustion turbine engine at low loads to slowly increase the HRSG temperature. Configurations with a stack bypass can slowly increase the percentage of the combustion turbine engine exhaust into the HRSG to increase the HRSG temperature without damage.

[105] Eddington, et al. (2017). *Fast start combined cycles: how fast is fast?*. Accessed at https://www.power-eng.com/emissions/fast-start-combined-cycles-how-fast-is-fast/#gref.

[106] Gulen, S.C. (2013). *Gas Turbine Combined Cycle Fast Start: The Physics Behind the Concept*. Accessed at http://www.mcilvainecompany.com/Decision_Tree/subscriber/Tree/DescriptionTextLinks/Physics.pdf.

[107] Li, D., Hu, Y., Li, D., Wang, J. (2019). *Combined-cycle gas turbine power plant integration with cascaded latent heat thermal storage for fast dynamic responses.* Energy Conversion and Management. https://doi.org/10.1016/j.enconman.2018.12.082.

147.5, 117.5, and 50 minutes, respectively.[108] Thus, there is incentive to keep the HRSG warm when feasible.

5.1 Purge Credit

This technique involves an EGU receiving credit for a mandatory purging of the fuel systems during shutdown and adding isolation valves in the fuel supply system. This purge of residual fuel from the combustion system with fresh, ambient air is necessary to remove excess combustible fuels in the unit and lower the risk of fire. During a conventional combined cycle startup, this purge takes place prior to ignition, which increases start times, reduces efficiency by decreasing the temperature of the HRSG, and increases thermal fatigue on the units. Generating purge credits during shutdown allows fast-start EGUs to start up without a purge.

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[108] Decoussemaeker, P., Nagasayanam, A., Bauver, W. P., Rigoni, L., Cinquegrani, L., Epis, G., Donghi, M. (2016). *Startup Time Reduction for Combined Cycle Power Plants*. The Future of Gas Turbine Technology. 8th International Gas Turbine Conference. Accessed at https://etn.global/wp-content/uploads/2018/09/STARTUP-TIME-REDUCTION-FOR-COMBINED-CYCLE-POWER-PLANTS.pdf.

6 Levelized Cost of Electricity for Simple and Combined Cycle Turbines

6.1 Simple Cycle Turbines

A challenge with estimating the costs of higher efficiency simple cycle turbines is that the available cost and performance information is for different size combustion turbines. While larger combustion turbines tend to have higher efficiencies, due to economies of scale larger simple cycle turbines can also have lower costs on a $/kW basis. Therefore, direct comparison between combustion turbines is not a meaningful measure of the cost of efficient generation. To estimate the cost of increased generation of simple cycle turbines the EPA first estimated the costs of simple cycle turbines based on the combined cycle model plants in the NETL Baseline Flexible Operation Report.[109] Specifically, the EPA:

- Used the detailed costs of the frame-based combined cycle turbines in the NETL Flexible Generation Report and subtracted the costs of equipment not included in the aeroderivative simple cycle turbine model plants. The EPA used this approach to estimate the capital costs for both the EPA derived F-class and H-class simple cycle turbines.
- To estimate the fixed and variable operating costs, the EPA scaled the combined cycle fixed and variable operating costs based on the ratio of the F-class simple cycle and F-class combined cycle costs reported in the Annual Energy Outlook 2025.
- The EPA used the details of the NETL report for the efficiency of the simple cycle turbines.

The EPA then compared the derived costs against what would be estimated using generic scaling factors to approximate the economies of scale. Specifically, the EPA:

- Used the "rule of six-tenths" to estimate the cost of the larger H-class turbine assuming the same performance characteristics as the F-class turbine.[110] The costs were scaled based on the heat input ratings of the combustion turbines.
- The difference between the estimated costs and the NETL derived costs was determined to be the price of the improved performance of the H-class combustion turbine relative to the F-class combustion turbine. Specifically, the cost premium of a combustion turbine with an 8% lower heat rate (the difference between the H-class and F-class combustion turbines) is estimated as 10%.
- The EPA applied the 10% cost factor to both the capital and fixed cost of an EPA derived F-class combustion turbine with an 8% lower heat rate. The variable costs were not adjusted.

The EPA derived simple cycle turbined were used to estimate the compliance costs of a BSER based on the use of higher efficiency simple cycle turbines. Holding everything constant and assuming a 30 year facility life, the levelized costs of electricity (LCOE) from the combustion turbines is the same at a 31%

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[109] "Cost and Performance Baseline for Fossil Energy Plants, Volume 5: Natural Gas Electricity Generating Units for Flexible Operation." DOE/NETL-2023/3855. May 5, 2023.

[110] The rule of six-tenths is a generic approach to estimating economies of scale.

$$Estimated\ Cost\ of\ the\ Facility = Cost\ of\ Known\ Facility * (Size\ of\ Estimated\ Facility/Size\ of\ Known\ Facility)^{0.6}$$

capacity factor. At this capacity factor, compliance costs would be zero. At a 20% and 15% capacity factor, the estimated compliance costs would be $1.5/MWh and $35/tonne and $3.0/MWh and $69/tonne, respectively. While the estimated compliance costs have a relatively high degree of uncertainty and are likely high given the common use of high efficiency simple cycle turbines without a regulatory driver. The EPA has determined that even at the incremental costs the use of high efficiency simple cycle turbines as the BSER for intermediate load combustion turbines is reasonable.

6.2 Steady State Conditions - Combined Cycle

To determine the compliance costs of switching from a simple cycle to a combined cycle turbine, the EPA first determined the LCOE of various simple cycle and combined cycle turbines under 30-year steady state operating conditions.

To estimate the relative costs of simple cycle to combined cycle turbines, the EPA compared the LCOE of model simple cycle and combined cycle turbines at different capacity factors. The net output of combined cycle turbine model plants is often much larger than simple cycle turbine model plants and those costs cannot be compared directly for multiple reasons. First, a large, combined cycle turbine serves a different purpose than a smaller, or multiple smaller, simple cycle turbines. For example, a single 400 MW combined cycle turbine has different characteristics than four separate 100 MW simple cycle turbines. The minimum run time and minimum downtime are significantly shorter for simple cycle turbines compared to combined cycle turbines.[111] This flexibility allows simple cycle turbines to cycle more frequently than combined cycle turbines. In additon, simple cycle turbines can start up and shut down in 10 minutes or less while fast-start combined cycle turbines can take between 30 minutes and 2 hours to reach full load and shutdown takes 45 to 70 minutes.[112] This flexibility allows simple cycle turbines to respond more quickly to changes in electricity demand than combined cycle turbines. Finally, simple cycle turbines can operate at as low as 15 percent of the rated full load while the minimum load for combined cycle turbines is 30 percent of the rated full load.[113] This flexibility provides a block of simple cycle turbines the ability to provide small increments of electricity to the grid, relative to a similar-sized combined cycle turbine. In addition, the costs of combustion turbines on a $/kW basis decline with size, and the calculated LCOE of the model combined cycle turbines benefits from these economies of scale.

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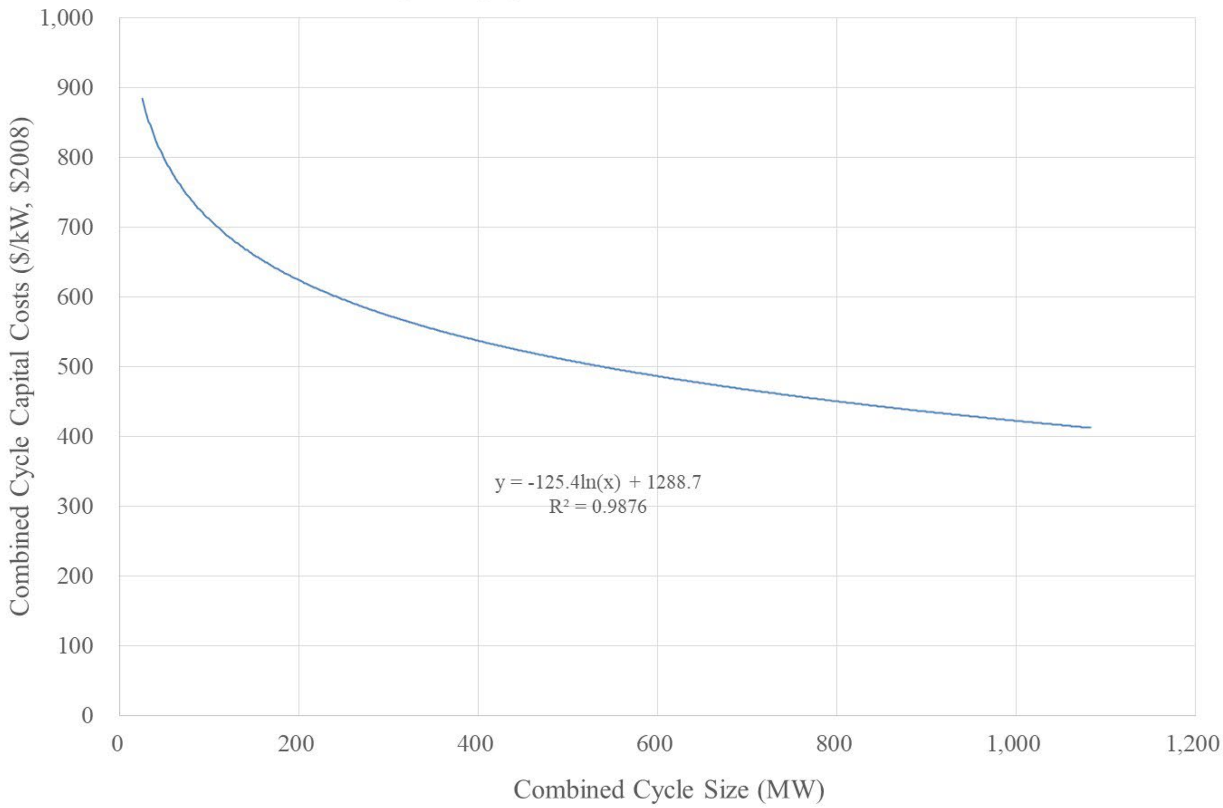
[111] The NETL Flexible Operation Report lists the minimum run times of simple cycle and combined cycle turbines as 15 minutes and 120 minutes, respectively. The NETL flexible Generation Report lists the minimum shutdown times of simple cycle and combined cycle turbines as 30 minutes and 60 minutes, respectively.

[112] The NETL flexible generation report lists the warm startup times of simple cycle turbines as 10 minutes and the hot startup times as between 5 to 8 minutes (depending on the specific model). The NETL flexible generation report lists the cold, warm, and hot startup times of fast start combined cycle turbines as between 10 minutes and the hot startup times as between 120 to 130 minutes (depending on the specific model), 45 to 85 minutes (depending on the specific model), and 30 to 35 minutes (depending on the specific model), respectively. The NETL baseline report (Exhibit 5-2. F- versus H-class combustion turbines) lists the startup times of F-Class and H-Class simple cycle turbines as 25 minutes and < 30 minutes, respectively.

[113] The NETL flexible generation reports list the minimum emissions-compliant load of simple cycle turbines as between 155 to 50% (depending on the model) and between 30% to 42% (depending on the model) for combined cycle turbines.

To compare the costs of simple and combined cycle turbines, the EPA used the EPA dervied model F-class and H-class simple cycle turbines. The capital and fixed costs were scaled using the factors derived from Study of Equipment Prices in the Power Sector. Figure 9 shows the relationship between the size of a combined cycle turbine and the capital costs.[114] The EPA used the relative costs of the combined cycle and simple cycle turbine to scale the capital and fixed operating costs to compare costs for the same size combustion turbines.

Figure 9: Economies of Scale for Combined Cycle Turbines
Study of Equipment Prices in the Power Sector



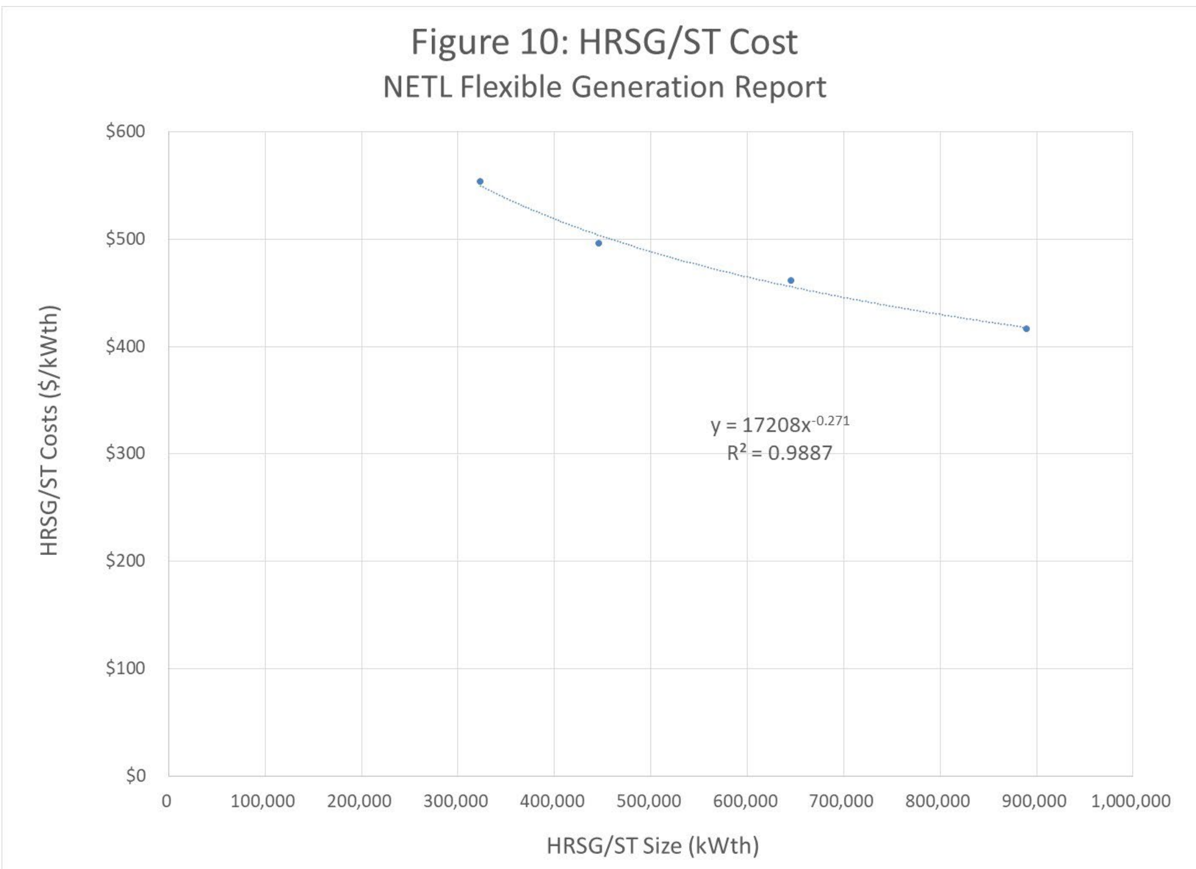
Using these costs, the NETL F-class and H-class combined cycle turbines have lower LCOEs than comparable simple cycle turbines at capacity factors of 37 percent and 40 percent, respectively. In addition, using the AEO 2025 costs and scaling the combined cycle costs estimates that the LCOE of an H-class combined cycle is lower than a comparable simple cycle turbine at capacity factors of 38 percent and higher. Based on this analysis, there are no compliance costs for using a combined cycle technology, compared to a simple cycle technology, capacity factors of approximately 40% or higher. The EPA

[114] Pauschert, D. (2009). *Study of equipment prices in the power sector (English).* World Bank Group. Accessed at http://documents.worldbank.org/curated/en/952421468330897396/Study-of-equipment-prices-in-the-power-sector

notes that these costs are for combustion turbines with similar output ratings. However, in practice combined cycle turbines tend to be much large than simple cycle turbines and benefit from economies of scale.

The EPA also estimated the costs of combined cycle turbines using aeroderivative combustion turbines. First, the EPA used publicly available design specifications to estimate the heat rate of an aeroderivative-based combined cycle turbine. To estimate the total overnight costs of the HRSG and steam turbine of the combined cycle facility, the EPA:

- Used the costs of the HRSG/ST costs for the NETL F-class and H-class 1x1 and 2x1 combined cycle turbine model plants to develop a $/kW-thermal cost for a HRSG/ST against the HRSG/ST size in kW-thermal. This curve was then used to estimate the $/kW-thermal costs for the smaller HRSG/ST used on the model aeroderivative combined cycle turbines. The kW-thermal heat input to the HRSG was estimated by subtracting the gross output of the turbine from the heat input.



- The projected costs were discounted by 2.2 percent to account for not requiring an auxiliary boiler and a further 3.9 percent assuming the HRSG is a 2-pressure HRSG without reheat (the NETL model HRSGs use 3-pressure steam and a reheat cycle).

To estimate the increase in fixed costs, the EPA:

- Determined the estimated increase in fixed costs for the H-class combined cycle facility from the HRSG/ST on a MMBtu/h heat input to the HRSG basis.
- Increased the fixed costs of the simple cycle facility by multiplying this factor times the estimated heat input to the aeroderivative HRSG.
- The EPA then multiplied the increase by the ratio of fixed capital costs of the HRSG/ST to get the total increase in fixed costs.
- The projected costs were discounted by 2.2 percent to account for not requiring an auxiliary boiler and a further 3.9 percent assuming the HRSG is a 2-pressure HRSG without reheat (the NETL model HRSGs use 3-pressure steam and a reheat cycle). The cost of the 100 MW aeroderivative HRSG was reduced by another 40 percent to account the intercooler removing heat from the system prior to the exhaust and less thermal energy entering the HRSG. The EPA based this value on the relative efficiency gains of the 50 MW and 100 MW combined cycle turbines compared to the simple cycle turbines.

To estimate the increase in variable operating costs, the EPA:

- Added the EPA estimated increase in operating costs for an H-class simple cycle compared to an H-class combined cycle ($0.9/MWh) to the variable costs of the simple cycle turbine.
- The EPA scaled this value based on the relative output of the steam turbine to the total output of the combined cycle facility.

Using the EPA develop model aeroderivative combined cycle turbines, the EPA estimated that the LCOE of an aeroderivative combined cycle turbine is lower than a corresponding aeroderivative simple cycle turbine at capacity factors of 41 percent and 56 percent for the 50 MW and 100 MW aeroderivative simple cycle turbines, respectively. Figure 11 shows the estimated steady state LCOE of simple cycle turbines and the corresponding similar sized combined cycle turbines.

Figure 11: Simple Cycle and Combined Cycle Turbine LCOE (steady state conditions)

	Steady State LCOE ($/MWh)							
Capacity Factor (%)	F-Class Combined Cycle	F-Class Simple Cycle	H-Class Combined Cycle	H-Class Simple Cycle	100 MW Aeroderivative Combined Cycle	100 MW Aeroderivative Simple Cycle	50 MW Aeroderivative Combined Cycle	50 MW Aeroderivative Simple Cycle
5%	308	237	268	205	428	380	506	448
10%	166	136	146	119	229	207	267	242
20%	96	86	85	76	130	121	147	139
30%	72	69	65	62	96	92	107	104
40%	60	61	54	55	80	78	87	87
50%	53	56	48	50	70	69	75	77
60%	48	53	44	47	63	64	67	70
70%	45	50	41	45	58	59	62	65
80%	43	49	39	44	55	56	57	61

6.3 Variable Operation—Combined Cycle

The above LCOE calculations do not account for startup and shutdown costs and potential increase in emission rates due to non-steady state operation. The EPA used the NETL Flexible Generation Report to account for synchronization costs, startup and shutdown costs, and the use of start-based O&M costs instead of hours-based O&M costs.

The EPA used the synchronization costs directly without adjustment of the NETL model plants.

- For the EPA-derived model frame simple cycle turbines, the synchronization costs were estimated by scaling the costs of the large aeroderivative simple cycle turbine based on the heat input rating.
- For the EPA-derived aeroderivative combined cycle turbines, the EPA scaled the hot synchronization costs of the 1x1 F-class combined cycle turbine based on the heat input rating. The EPA assumed the aeroderivative combined cycle turbines would use a less complex HRSG and would not have an auxiliary boiler and applied the single derived cost to hot, warm, and cold starts.

When the average run time per start drops below 25 hours, the NETL flexible generation report states that operation and maintenance costs switch from an hours of operation approach to a starts-based approach. To account for the increase in costs, the EPA applied the cost-per-start costs listed in the appendix of the report. For the EPA-derived aeroderivative combined cycle turbines, the EPA scaled the 1x1 F-class combined cycle turbine costs ($14,000/start) by the relative output from the steam turbine. The EPA did not adjust the variable or fixed O&M costs for the starts-based costing approach.

The EPA used these costs to estimate the LCOE of flexible fast-start combined cycle turbines that could replace high capacity factor simple cycle turbines.