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SUPERCRITICAL CO_2 CYCLES FOR GAS TURBINE COMBINED CYCLE POWER PLANTS

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ABSTRACT

Supercritical carbon dioxide (sCO_2) used as the working fluid in closed loop power conversion cycles offers significant advantages over steam and organic fluid based Rankine cycles. Echogen Power Systems LLC has developed several variants of sCO_2 cycles that are optimized for bottoming and heat recovery applications. In contrast to cycles used in previous nuclear and CSP studies, these cycles are highly effective in extracting heat from a sensible thermal source such as gas turbine exhaust or industrial process waste heat, and then converting it to power. In this study, conceptual designs of sCO_2 heat recovery systems are developed for gas turbine combined cycle (GTCC) power generation over a broad range of system sizes, ranging from distributed generation (~5MW) to utility scale (> 500MW). Advanced cycle simulation tools employing non-linear multivariate constrained optimization processes are combined with system and plant cost models to generate families of designs with different cycle topologies. The recently introduced EPS100 [1], the first commercial-scale sCO_2 heat recovery engine, is used to validate the results of the cost and performance models.

The results of the simulation process are shown as system installed cost as a function of power, which allows objective comparisons between different cycle architectures, and to other power generation technologies. Comparable system cost and performance studies for conventional steam-based GTCC are presented on the basis of GT-Pro™ simulations [2]. Over the full range of systems studied, the sCO_2 cycles generated higher power output at a lower cost than the comparable steam systems. Projected operation and maintenance (O&M) costs are used to calculate projected levelized cost of electricity (LCOE) for the competing cycles, demonstrating that sCO_2 systems can provide a significant LCOE advantage across the full range of sizes studied.

INTRODUCTION

The combined cycle gas turbine (CCGT) power plant has established itself as the highest efficiency fuel-to-power conversion technology available today, with overall plant efficiency values running as high as 61% lower heating value (LHV) [3]. The combination of advanced gas turbine technology with the latest steam cycle innovations provides a reliable, low emissions power plant burning natural gas fuel. The performance and cost of the CCGT bottoming cycle have substantial effects on the overall plant economics, as up to 35% of the total output is generated by the bottoming cycle. In addition, the bottoming cycle represents a significant fraction of the total combined cycle plant cost.

Supercritical CO_2 heat recovery systems offer several advantages over existing steam-based systems. The compact size of sCO_2 turbomachinery, the elimination of water treatment systems, and the simplicity of the primary heat recovery heat exchanger enable lower capital and installation costs, while advanced cycle designs achieve cycle performance that can match or exceed the incumbent technology. In addition, the non-condensing nature and smaller physical size of sCO_2 turbine will reduce maintenance costs, and the elimination of water treatment systems will reduce operating costs. The low freezing point of CO_2 (-55°C) also eliminates the need for freeze-protection in cold climates. Finally, although the baseline for comparison in this study is for a water-cooled configuration, sCO_2 cycles can also be used in air-cooled configurations, thus allowing for a completely water-free installation.

In the present study, we consider the potential improvement of the CCGT system by replacing the steam bottoming cycle with an advanced system utilizing supercritical carbon dioxide (sCO_2) in a closed loop heat recovery cycle.

SUPERCRITICAL CO_2 CYCLE BACKGROUND

Supercritical fluid power cycles, and specifically those using carbon dioxide as the working fluid, have been considered as replacements for the steam Rankine cycle since at least the late 1960's [4,5]. The primary advantage identified by these early authors was that a supercritical fluid (a fluid at a higher pressure than the critical pressure) does not undergo a constant-temperature boiling process during heating. Rather, a continuous reduction in density occurs as the fluid is heated, eliminating the classical heat exchanger "pinch" problem that necessitates complex double or triple pressure heat exchanger arrangements to achieve high steam turbine inlet temperature and cycle efficiency.

Carbon dioxide was identified as an advantageous working fluid for these new supercritical cycles due to several factors. It has a relatively low critical pressure (7.38MPa, compared to 22.1MPa for water), allowing for cycle operation well above the critical pressure and vapor dome at working pressures for which process fluid equipment is readily commercially available. It is a relatively safe working fluid, having low toxicity and corrosivity, no flammability, and is thermally stable. Carbon dioxide is a low-cost, readily available fluid with an existing world-wide commercial distribution network. Finally, due to the high density of the fluid throughout the power cycle, the physical size of CO_2 equipment is compact.

The first documented consideration for sCO_2 cycles as gas turbine bottoming cycles dates from the 1970's, where the potential for heat recovery in a compact physical device garnered some attention for its use in shipboard applications [6]. However, further developments in sCO_2 cycles did not occur until the mid-2000's, at which point interest was renewed in context of their use in advanced nuclear cycles [7,8], and concentrated solar power (CSP) systems [9,10]. The advantages of sCO_2 thermodynamic power cycles have also been studied for waste and exhaust heat recovery [11,12] and oxyfuel combustion cycles for primary power [13]. Many of these early studies were focused on theoretical cycle development, although significant advances have been made in laboratory-scale experimental systems [14,15].

A sCO_2 cycle can take many forms, depending upon the application. The simplest practical form of a sCO_2 cycle, known as the “simple recuperated cycle,” is shown in Figure 1. The fluid is compressed from a state that is either a liquid, or in a high density supercritical state, into a state that is well above (typically 3-4 times) the critical pressure. The fluid then undergoes a sequence of both internal and external heat additions, until it has reached the highest temperature in the cycle at the turbine inlet(s). At this point, the fluid is expanded through one or more turbines, generating shaft work that can be converted to power, and/or used to drive additional equipment. As the overall cycle pressure ratio is low, significant enthalpy remains available in the turbine exhaust. To recover this heat, one or more internal heat exchangers (recuperators) are used to transfer the heat into the fluid at the high pressure state. Any residual fluid enthalpy is then rejected to the environment, allowing the fluid to return to the initial state in the cycle at the pump[1] inlet.

[1] Note that we use the term “pump” to refer to the device that increases the pressure at the lowest temperature point in the cycle, whether the fluid at the inlet is liquid or a supercritical fluid. See [1] for a more detailed description of the nomenclature challenges associated with this cycle.

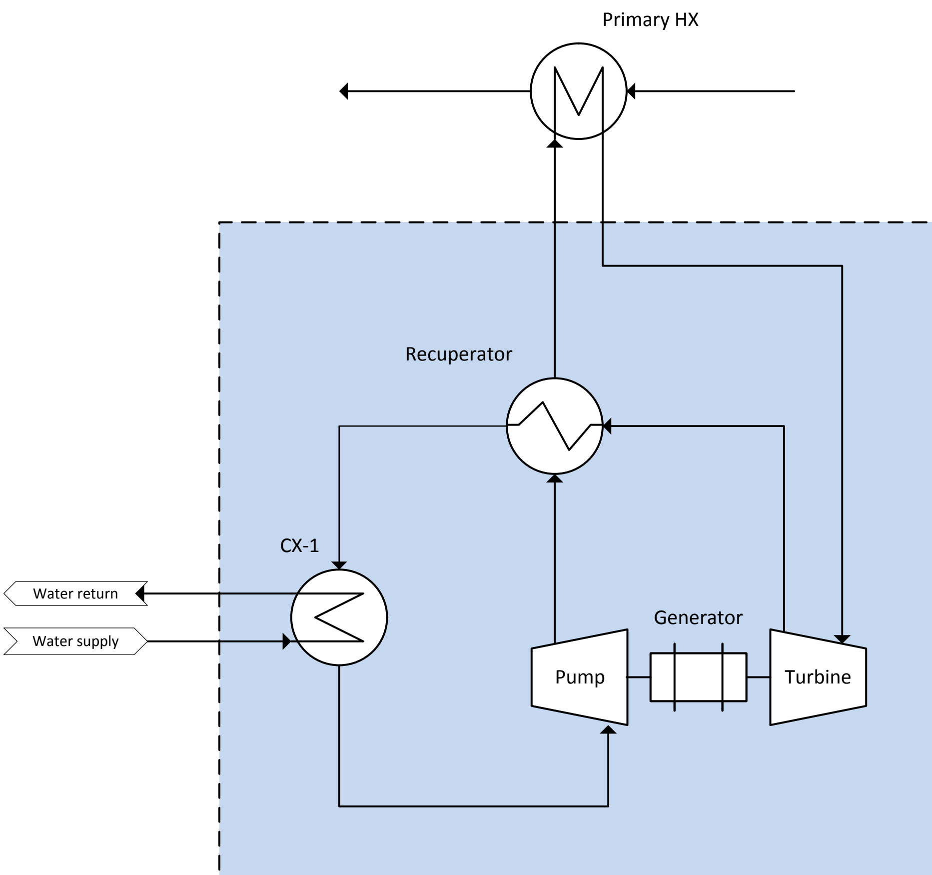

Figure 1: Simple recuperated sCO_2 cycle

More complex cycle architectures, such as the "recompression cycle" [7] enable very high thermodynamic efficiency (defined as net power output / heat input, or W_{out}/Q_{in}) for heat sources that are internally recirculated, such as nuclear and CSP. For these applications, maximum power output is achieved when the thermodynamic efficiency of the cycle is maximized. Because these cycles perform poorly for non-recirculated sources, such as bottoming applications, they are not considered here.

On the other hand, heat recovery cycles are designed to maximize power output by simultaneously achieving high thermodynamic efficiency and minimizing the unrecovered enthalpy to the greatest allowable extent. As a result, for the same heat source temperature, the direct conversion efficiency of heat recovery cycles is lower than that of the typically considered cycles (recompression, partial cooling, etc.). However, heat recovery cycles will deliver a significantly higher output power from a sensible enthalpy heat source because of their increased utilization of available enthalpy.

POWER CYCLE SYSTEM DESCRIPTION

Supercritical CO_2 power cycles have several distinguishing characteristics that influence their design and application. The foremost characteristic is that the pressure between the pump discharge and the turbine inlet is well above the critical pressure (7.38 MPa), typically 20-25MPa. At these conditions, the fluid no longer possesses distinct liquid and vapor states. Most importantly from the perspective of the power cycle, as heat is added to the fluid, its temperature increases continuously, in contrast to a subcritical fluid that undergoes a constant temperature boiling process as it transitions from liquid to vapor state. The absence of a boiling process greatly simplifies the exhaust heat exchanger (EHX) design, eliminating the need for multiple pressures, and separate economizer, boiler and superheater sections. At the same time, the sCO_2 EHX coils can use conventional finned tubes, leveraging many years of manufacturing experience.

As described above, the residual enthalpy in the expanded fluid is transferred back into the high pressure fluid by one or more recuperators. Few commercially available heat exchangers can support the high pressures and high effectiveness required by sCO_2 cycles. Currently, the only commercially available, practical solution is the Printed Circuit Heat Exchanger (PCHE), a diffusion-bonded stacked-plate design with chemically-etched passages [16]. A similar version of the heat exchanger can be used as a water cooled condenser/cooler, although the lower pressure of this heat exchanger does open possibilities for other configurations. For the present study, PCHE's are assumed for both types of heat exchanger.

Turbomachinery forms the "heart" of the power cycle, providing both the means to increase the pressure of the fluid and extract energy from the fluid and convert it to mechanical energy. One of the key features of sCO_2 power cycles is the small physical size of the turbomachinery, due in part to the high density of the working fluid, and also to the low pressure ratio of the cycle. This small size results in a lower cost, simpler turbomachinery, with lower installation costs. The low cycle pressure ratio also results in single-phase flow within the turbine, avoiding the droplet condensation erosion issues encountered in steam turbines.

SUPERCRITICAL CO_2 COMMERCIALIZATION

Within the last two years, Echogen Power Systems, LLC has developed and is continuing to refine commercial-scale sCO_2 cycles and systems specifically for moderate temperature thermal power conversion, including industrial waste heat recovery (WHR) and exhaust heat recovery (EHR) applications. These applications are characterized by heat source temperatures in the 300 to 600°C range, and heat that is in the form of sensible enthalpy (that is, $Q = w_{source}(h_{source} - h_{residual})$, where w_{source} is the mass flow rate of the thermal medium, h_{source} is the enthalpy of the heat source at the inlet of the main heat exchanger, and $h_{residual}$ is the unrecovered enthalpy from the source). The unrecovered enthalpy is that which cannot be recovered from the source, due to cycle limitations, technical limitations (e.g., a minimum allowable stack temperature to avoid condensation in the exhaust), or economic factors. The residual enthalpy is permanently lost to the energy conversion process, generally in the form of thermal energy in the exhaust.

As the first step in commercialization of sCO_2 EHR cycles, Echogen designed the EPS100 (Figure 2), a 7 to 8 MW class heat recovery engine, targeted at small-scale CCGT (~30MWe total output) applications, such as those used in distributed generation or oil and gas applications. The EPS100 recently completed factory validation testing at the Dresser-Rand/Siemens facility in Olean, NY [1]. The experience gained in cycle and system development during this process has provided added confidence in the ability of sCO_2 power cycles to form the basis for larger-scale CCGT plants. The performance and cost models used in the present work are largely based on the actual performance and costs developed in the EPS100 commercialization process.

Figure 2: EPS100 sCO_2 heat engine, process and power skids

The present work is a study of the potential for sCO_2 cycles to form the basis of combined cycle power plant bottoming cycles up to and including utility scale applications. This combined performance and economic assessment of sCO_2 cycles at large scale indicates that given sufficient time and effort, sCO_2 can deliver a lower cost, higher performance option for CCGT plants that does the current steam technology.

CYCLE DESIGN AND OPTIMIZATION

The management of the internal and external sources of enthalpy is a challenging aspect of sCO_2 power cycle design. The overall goal for the CCGT bottoming cycle is to generate the most power possible, while respecting appropriate economic and technical constraints.

The conceptual design process consists of two major activities – selecting the cycle architecture, and sizing the equipment. The term “cycle architecture” refers to the general arrangement of turbines, pumps, recuperators and external heat exchangers. Within a given cycle architecture, the sizing of the various heat exchangers has direct influence on both the system performance (power output) and capital cost. Selection of the cycle operating pressures and flow rates also affects performance, but has a relatively minor impact on system cost within reasonable ranges.

The process of cycle optimization requires a model that is a valid representation of both the thermodynamic performance of the system and its capital cost. The model utilized in this study contains an integrated set of component and system-based thermodynamic and cost submodels. Upon selection of the cycle architecture, and assignment of appropriate boundary conditions (generally heat source temperature, flow rate and constituents, and heat rejection sink temperature), the model uses a non-linear optimization process to define the lowest cost solution that will achieve a given target net output power. By specifying a range of target output powers, a curve of system cost as a function of power output can be generated. This process can be repeated over several different cycle architectures to develop a fair means of comparison between them. An example is shown in Figure 3, which compares several candidate architectures, including the original EPS100 architecture. Presuming that this is the appropriate basis of comparison, it is clear that the "dual rail" cycle architecture [17] has the superior overall performance, and is the selected architecture both for further commercial introduction of the EPS100, and the remainder of this study.

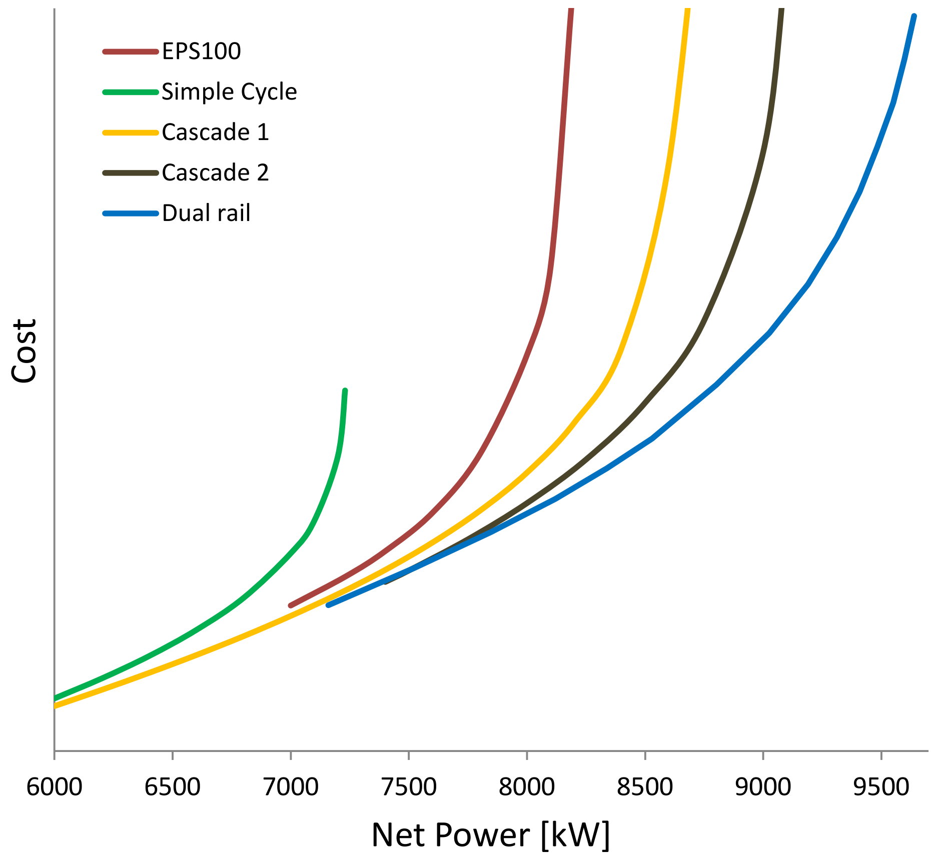

Figure 3: System cost versus power output, several candidate sCO_2 power cycle architectures

Dual rail cycle

The dual rail cycle (Figure 4) represents a balance between internal and external heat addition to the high pressure fluid stream. The fluid is separated into two primary paths, one through the external heat exchanger (EHX) and the second that recovers the residual heat from the recuperators. At points intermediate to these heat exchangers, fluid can be transferred in either direction through appropriate manipulation of flow split valves. The ability to tailor the flow rate sequentially through these two paths allows for ideal "temperature glide" matching between the two sides of each heat exchanger, thereby minimizing exergy (or "thermodynamic availability") destruction in the heat exchanger [18], and maximizing cycle performance.

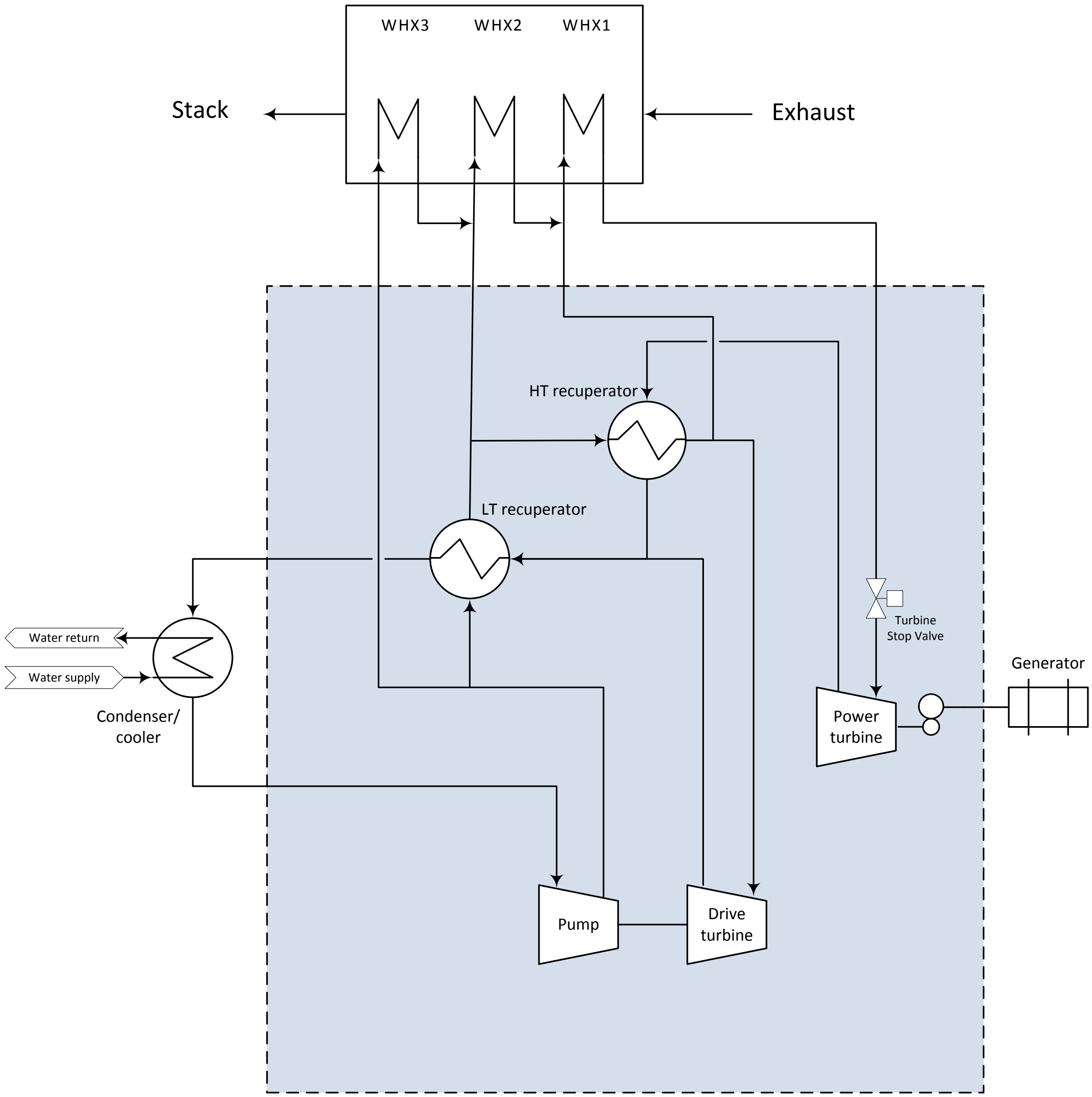

Figure 4: Dual rail cycle sCO_2 cycle.

The dual rail cycle offers a flexible method to accommodate the variation in fluid heat capacities, creating an excellent match of temperature glide in all the recuperator and external heat exchangers, and therefore providing high overall cycle efficiency.

Note that subcritical steam-based systems do not share this characteristic, since the constant temperature boiling process inevitably leads to a significant loss in thermodynamic availability. The design of modern heat recovery steam systems partially compensates for this loss by dividing the system into multiple pressures, which adds cost and complexity. At best, this can only approximate to a limited extent the continuous, well-matched thermodynamic process of the sCO_2 exhaust heat exchange.

Model assumptions and process

The cycle model is based on the dual rail cycle (Figure 4). The thermodynamic properties of all fluids are calculated by REFPROP 9.1 [19], using the Span & Wagner equation of state for CO_2 [20]. The exhaust properties (temperature, flow rate and composition) are taken from GT-Pro™ for standard ISO conditions (15°C (59°F) ambient temperature, 60% relative humidity, sea level). The reference bottoming cycle for comparison is a double-pressure steam Rankine system, using the default GT-Pro™ options.

The cycle model calculates a heat-and work-balanced condition that maximizes the net output power based upon a given cost target. To establish this design-point model, the solver varies overall heat exchanger sizing, turbine flow functions, internal flow splits, pump inlet and outlet pressures, and coolant flow rate. Off-design conditions can then be modeled by maintaining fixed geometry (turbomachinery flow function and efficiency, heat exchanger UA's and pressure drop characteristics), and solving for the heat- and work-balanced condition, this time only varying certain flow splits and pump inlet pressure, while respecting specified constraints on maximum pressure, minimum stack exit temperature, etc. The various component-level submodels are described below.

Turbine

Turbine performance is modeled by two parameters, isentropic efficiency and turbine flow function ($FF = w\sqrt{T}/P$), with additional models for mechanical (e.g. gearbox and bearing) losses. For design-point calculations, these parameters are input as constant values. When available, detailed performance maps in which these parameters are functions of normalized operating conditions (such as corrected speed and corrected enthalpy drop) may be used to simulate off-design conditions.

The largest currently-operating sCO_2 turbine is the EPS100 power turbine, a single-stage radial design, with a wheel diameter of approximately 0.24m (9.5 in), and a design-point efficiency of 85% [1]. The maximum practical size of a radial sCO_2 turbine is uncertain. Based on physical diameter and tip speed, much larger radial turbines, albeit at a lower shaft power rating, are in commercial service (e.g., [21]). For the purposes of the present study, we assume that the largest practical radial turbine will be approximately 20MW. Using a mean-line radial turbine design code [22], estimated efficiency at the upper end of the radial turbine size range is expected to be in the 86% range.

At larger scales, axial turbomachinery will likely be the selected configuration. In Figure 5, axial turbine efficiency vs shaft output power is shown for a variety of steam turbines. The majority of the values are taken from GT-Pro™ performance calculations of steam turbines in combined cycle applications. At utility-scale (~ 900MW), steam turbine efficiency can reach as high as 94% [23].

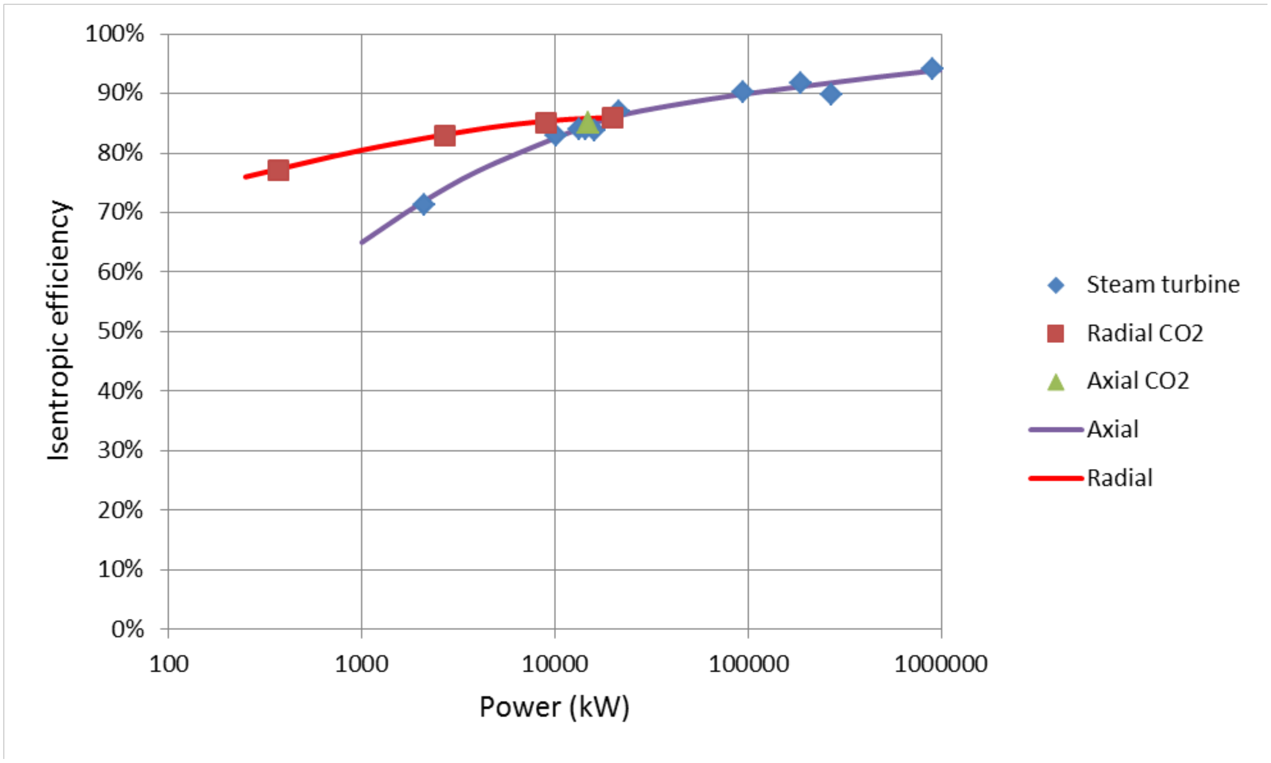

Figure 5: Turbine efficiency vs shaft power

No data currently exist for axial sCO_2 turbomachinery. A recent study of a 10MW class turbine for the DOE SunShot program projects turbine efficiency in excess of 85% [24], similar to steam and radial CO_2 turbines of comparable size. For the present study, we assume that sCO_2 axial turbine efficiency will follow a trend similar to steam turbines.

Pump

For design-point calculations, pump performance is managed similarly, where the only input parameter is the pump isentropic efficiency, and the optimal pump flow rate is determined by the solver. For off-design conditions, detailed pump performance is modeled with head-rise coefficient and efficiency maps as a function of normalized flow coefficient.

There are few operating CO_2 pumps for which published data are available. However, due to the high density and relatively low compressibility of sCO_2 at the pump, we assume that the general trends found in traditional pump literature will hold (Figure 6). The measured efficiency of the 2.7MW EPS100 pump is consistent with the correlation, as is the published pump performance map of a 1.6MW barrel casing CO_2 pump [25].

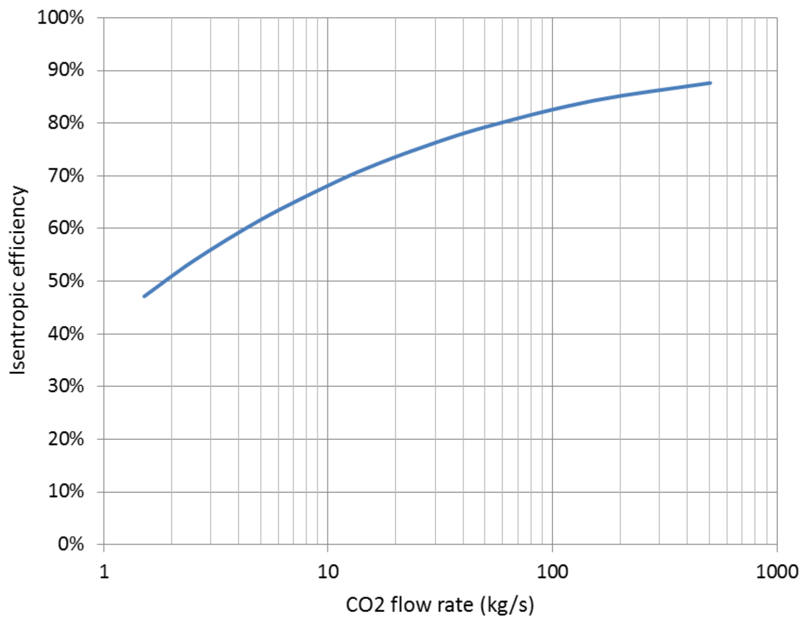

Figure 6: Pump efficiency as a function of capacity, assuming high specific speed.

Other mechanical and electrical losses are accounted internally. These include gearbox (below 50MW – above that power level, axial turbine speed is consistent with direct generator drive), bearing, and generator losses. Models for these losses are based on general sizing rules and established performance from existing systems.

Heat exchangers

The overall heat exchanger size is specified on the basis of a UA value, normally defined as the design-point heat transferred divided by a log-mean temperature difference (UA=Q/LMTD). However, to accommodate the variation in the heat capacity of CO_2, a 25 sub-element discretized model of the heat exchanger is used to calculate the total heat transferred. The performance of this model has been shown to accurately reproduce the measured heat transfer performance of PCHE's in the EPS100 test program [1]. Design-point pressure drops for the PCHE's were set to values consistent with the EPS100, 0.1-0.3 MPa depending on the heat exchanger.

Although the EHX configuration is a cross-flow finned-tube heat exchanger (exhaust side mixed, CO_2 side unmixed), the number of cross-flow passes is typically at least eight. With this large number of passes, the performance of the cross-flow heat exchanger very closely approaches the performance of counter-flow geometry [26]. Therefore for the current study, the EHX is modeled as a counter-flow heat exchanger. Exhaust-side pressure drop was set to 1 kPa for the exhaust heat exchanger (EHX) coils consistent with typical gas turbine experience, and CO_2-side pressure drop is set to 0.15 MPa per coil.

Cost models and inputs

Balancing power and cost, such as shown in Figure 3, requires not only a thermodynamic performance model for the cycle, but also models for the individual components and installation costs. The only existing sCO_2 heat engine of appreciable size is the EPS100. The cost models used herein are based on the experience gained in the design and fabrication of this system, and to a lesser extent additional experience developed during pilot-scale system design and fabrication, and the ongoing design of the 1.5 MWe EPS30M system [27].

The system hardware costs are divided into two main categories. The first category constitutes items that represent an essentially fixed cost, such as instrumentation and controls, for which the costs determined in the EPS100 program are used directly. The second category includes items that scale with power output, such as generators, gearboxes, piping and valves. For these items, individual cost models have been created based on literature references, supplier quotes, and catalog pricing as available. In general, the cost models follow power-law expressions:

$$Cost = A + b \cdot P^n,$$

where P is the rated output power of the system. The exponent n ranges from 0.6 to 1.0, depending on the component. Installation costs are assumed to follow a similar trend as steam system components, with an "installation cost factor" defined as the total installed cost of the component or subsystem divided by the hardware cost of the item. These cost factors are derived from system design studies using the Thermoflow GT-Pro/PEACE software [2].

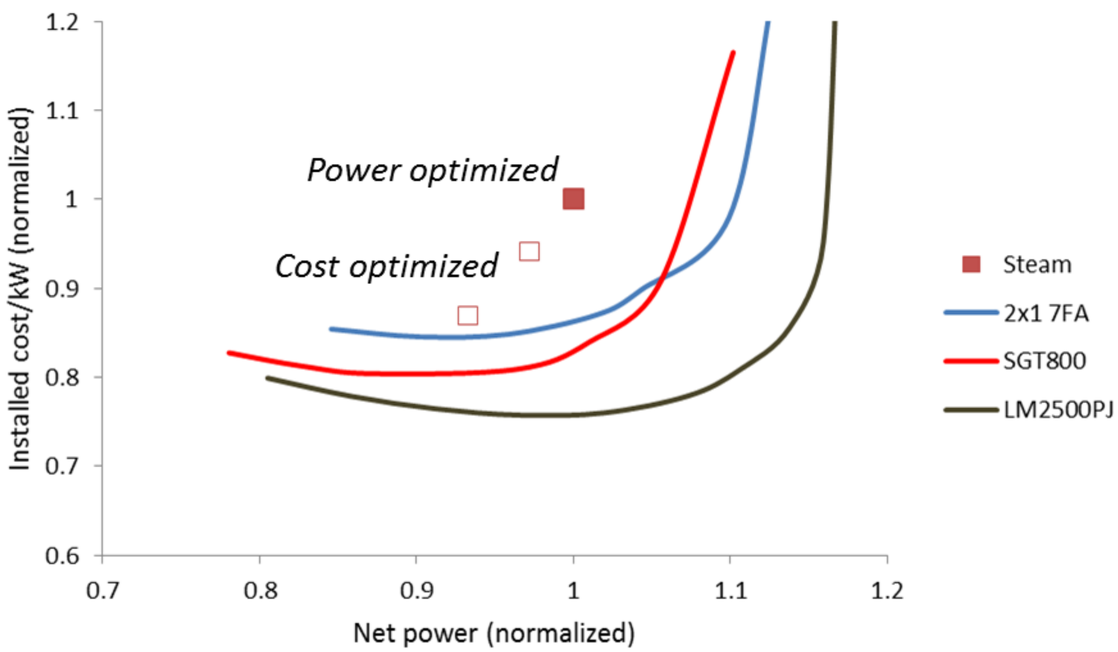

Figure 7: Cost/kW vs net power for three sizes of combined cycle plants, normalized to GT-Pro "power-optimized" cases. The open symbols are two example GT-Pro cases using the "cost-optimized" option.

RESULTS

Comparisons between system performance and cost are shown in Figure 7 for three different general types and sizes of combined cycle plants. The cost/kW and net power are normalized to the GT-Pro "power-optimized" output and cost values. For the same net power output, the installed cost is projected to be 10-20% lower than the comparable steam system. Alternatively, for the same installed cost, the sCO_2 systems are projected to have a 7-14% higher output power. Additional details of each case study are described below.

LM2500PJ. The first study covers the relatively small-scale application LM2500PJ aeroderivative gas turbine, which is the design target size of the EPS100. This gas turbine is frequently used in mechanical drive applications, such as gas pipeline compressor stations. The excess power derived from the EPS100 can be exported to the grid if a utility tie-in point is near enough to be economically feasible, or can be used for additional mechanical drive to either supplement or replace some of the gas turbine output.

A comparison of the sCO_2-based greenfield installation to the comparable steam system installation is shown in Figure 8. The main exhaust heat exchangers are similar in size, since the limiting heat transfer coefficient is the exhaust side, and the two heat exchangers therefore have similar heat transfer surface area. The exchangers also have similar frontal area, as they are designed to a similar gas-side pressure drop. Heat rejection systems also are of similar scale, due to the comparable heat rejection duties of the two systems.

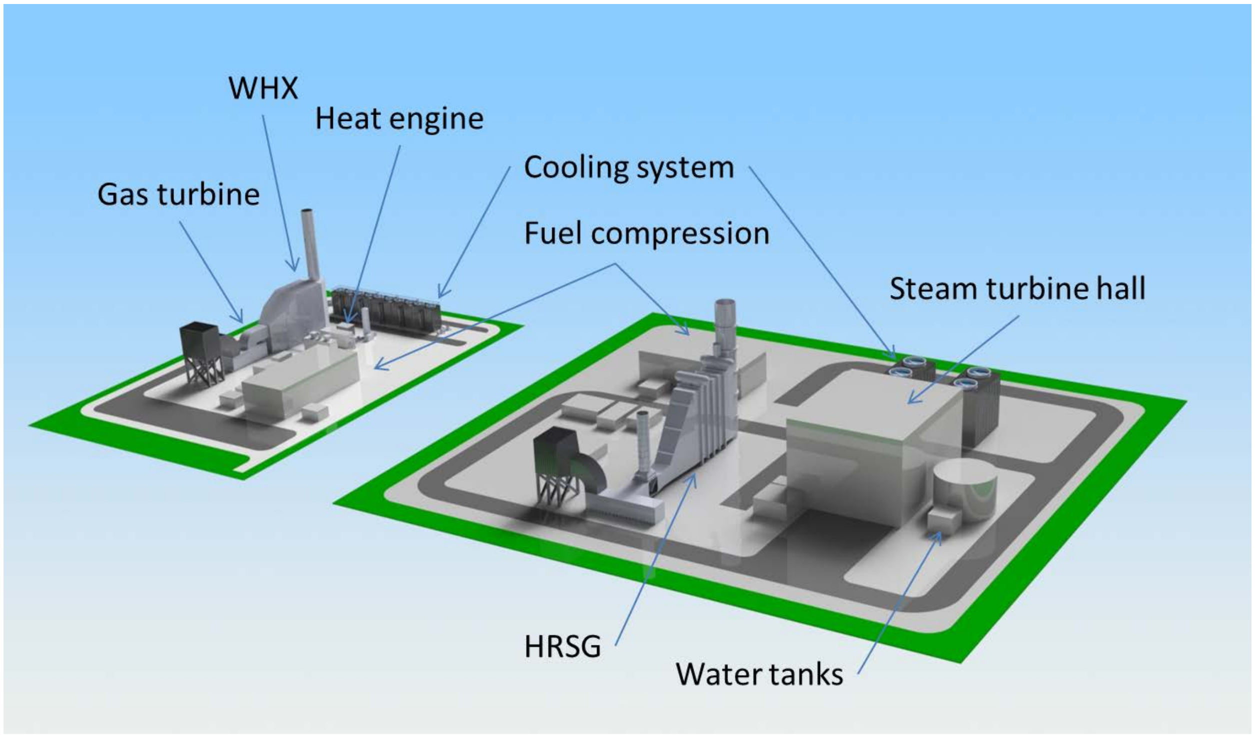

Figure 8: Comparison of LM2500/sCO_2 combined cycle installation with LM2500/HRSG

The major footprint difference between the two installations is in the primary power generation equipment. The small size of the power turbine and condenser compared to their steam equivalents permit a much reduced footprint. The recuperators represent additional hardware compared to steam systems, but their total volume (approximately 2 m^3) represents a small increment.

Turbine architecture selection at this size range strongly favors single-stage radial designs. As a result, the physical size of the turbines is quite compact (0.1-0.2m wheel diameter), and the speeds are high (30-35kRPM). The power turbine thus requires a compound epicyclic gearbox to match to a 2-pole generator. However, the smaller single-stage drive turbine matches speed well to a single-stage pump design, allowing for a compact, low-cost design.

For sites located in areas in which sub-freezing temperatures are common, the steam turbine, condenser, water treatment and auxiliary systems are typically housed within a turbine house as shown in Figure 8. Due to the skid-mounted arrangement of the EPS100, and its tolerance to extremely low ambient temperatures without risk of freezing of the working fluid, the power generation equipment occupies a much smaller footprint. The elimination of the building and other infrastructure required for the water treatment systems also represents a significant installation cost advantage for sCO_2 systems.

SGT800: The second example represents a natural scale-up of the EPS100. The Siemens SGT800 is a small industrial gas turbine, which bridges the application space between larger oil and gas applications and small-scale power generation. With a lower pressure ratio than typical aeroderivative gas turbines, and therefore a higher exhaust temperature, the SGT800 is well suited to combined cycle applications.

The same sCO_2 and steam cycle configurations are used for this larger-scale application. For the sCO_2 system, the turbine and pump characteristic curves were used to estimate the projected isentropic efficiency values. The turbines for this configuration are nearing the point where multi-stage axial turbines are able to approach the isentropic efficiency of single-stage radial turbines. The final selection of turbine architecture will require a more detailed study of performance and cost. Using classical turbine scaling guidelines [28], a single stage radial turbine would operate at a speed of approximately 20000RPM, in the range where a single-stage parallel shaft gearbox could drive a 2-pole generator. A multi-stage axial turbine would permit lower shaft speeds, but may result in slightly lower efficiency. For the present, we assume radial designs at this size.

The heat exchanger UA's were allowed to vary to establish the optimal balance between cost and performance as shown in Figure 7, using the cost scaling rules described above. The power-optimized steam point is shown for comparison. Again, the sCO_2 system delivers a lower cost solution at a comparable output, or a higher output at a similar cost to the conventional steam system.

2x2x1 GT-7F.04: The third case study is a utility-scale power plant, using two GE GT-7F.04 gas turbines, two exhaust heat recovery heat exchangers, and a single sCO_2 power cycle (a 2x2x1 configuration) to generate approximately 550MW net electrical power. This arrangement is commonly used in steam-based combined cycle gas turbine power plants. This configuration is a significant scale-up (roughly a factor of 20) in sCO_2 power output from the current EPS100 system. The challenges and opportunities involved in this scaling are discussed below.

At the 180MW scale of this application, a multi-stage axial turbine is the preferred configuration. A preliminary scaling design for the power and drive turbines resulted in a design with 5 stages, and a diameter of 0.8 meters, very similar to the first few stages of a high pressure turbine in a similar size range. At this size, shaft speeds can be reduced to the 3000-3600RPM range, permitting direct generator drive. Total turbine weight is projected to be approximately 50 tons, with a rotor mass of 15 tons. In comparison, a 160MWe steam turbine has a total weight of 200 tons, and a rotor mass of 85 tons. The reduced rotating mass in particular can be expected to generate substantial savings in foundation weight and cost.

Scaling the recuperators to these larger sizes requires dividing the total heat transfer duty into multiple devices, as transportation and mechanical support issues can limit the maximum weight of a single heat exchanger – for the purposes of this study, a maximum single-item weight of 75 tons is assumed. For the design case selected, each recuperator would consist of 2 or 3 individual units, connected by external manifolding. While this adds some piping complexity, it would not represent a significant mechanical or site footprint disadvantage.

Similar to the preceding cases, the projected performance and cost of the sCO_2 system is compared to the power-optimized steam case, now with a triple-pressure HRSG as typically employed at these larger scales. The cost extrapolation to this size is considerable, but the general trend still holds – sCO_2 systems can exceed the power output of a typical steam system at a lower projected total cost.

COST REDUCTION POTENTIAL

Steam-based power plants have over 100 years of commercialization and cost optimization history, while sCO_2 systems are only now entering commercial service. It is to be expected that over time, significant cost reduction potential exists for sCO_2 bottoming cycles. Some potential areas for further optimization are outlined below.

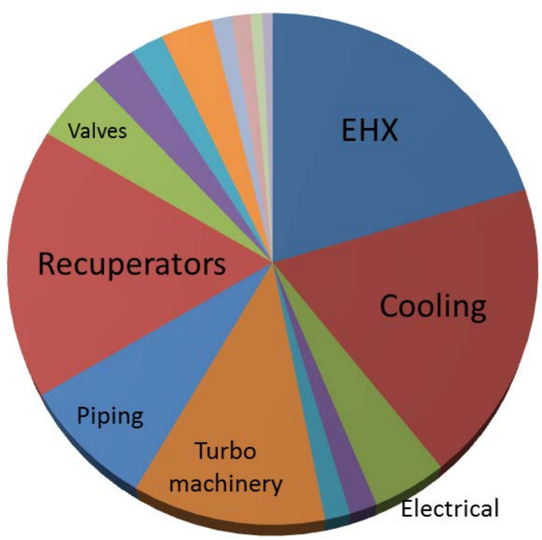

Figure 9: Selected approximate equipment cost breakdown for sCO_2 system

An approximate breakdown of the component equipment costs for an example sCO_2 system is shown in Figure 9. As the technology and commercial market matures, we expect that opportunities for significant cost reduction will be realized in the following areas.

Recuperators: At present, the supplier base for PCHE's is only beginning to expand beyond the pioneering company in this field, and new technologies for lower cost compact heat exchangers specifically designed for sCO_2 power cycles are being developed under private and government funding [29–31]. It is reasonable to expect that significant improvements in the cost basis of sCO_2 cycles will be achieved as these technology and commercial development plans develop into maturity.

Exhaust heat exchangers: The physical size of the sCO_2 exhaust heat exchanger (EHX) is similar to that of the HRSG. In both sCO_2 and steam-based systems, finned tube heat exchangers are used, and the limiting heat transfer coefficient is the exhaust heat transfer by convection to the fin and tube surfaces, representing 85-95% of the overall thermal resistance. Therefore, the amount of physical surface area needed to extract the same amount of heat from both cycles is similar. The sCO_2 exhaust heat exchanger operates at higher pressure than does the HRSG, and thus thicker wall tubing is required, making the tube bundles themselves heavier. However, the higher tube weight of the EHX is offset by the much lower weight of the simple pipe headers compared to the thick-walled steam drums of a conventional HRSG. The smaller wall thickness of the EHX header also simplifies manufacturing, as the ratio of header thickness to tube thickness is much smaller, making the welding process less challenging.

In addition, the EHX consists of three coils of similar construction, while a double-pressure HRSG typically can require 6-11 different coils (generally at least a separate economizer, boiler and evaporator for each pressure level), and triple-pressure HRSGs up to 13 coils. In general, we have found that quoted prices for a similar capacity EHX are 30-40% lower than a comparable HRSG, largely due to the simplicity of construction.

sCO_2 technology offers potential advantages for more compact, and theoretically lower-cost EHX technologies. Combining the high pressure capability of diffusion-bonded heat exchangers with the large surface area and low pressure drop of formed fin geometry, a hybrid heat exchanger [16] could provide a significant advantage over conventional finned tube heat exchangers for EHX service. In a preliminary study of a small-scale (~1.5 MW) system, heat exchanger mass could be reduced by a factor of two, and footprint by a factor of three relative to a finned tube geometry with 50mm tubes. Although significant development is required to implement advanced heat exchanger geometries, the potential for major gains in performance, footprint and cost is clear.

Condenser: A steam condenser operates at low vacuum, and therefor large volumetric flow rate of steam. For the 2x2x1 7FA case, the turbine exhaust flow is $3900 m^3/s$, requiring a flow area of at least $80 m^2$, while the sCO_2 system has a condenser inlet volumetric flow rate of only $11 m^3/s$ due to the much higher density of CO_2 at those conditions. This results in much smaller interconnecting piping, which is particularly important for air cooled systems. However, the higher density is due in part to the much higher pressure of the sCO_2 system, which therefore results in heavier (although smaller volume) components. For instance, the EPS100 condenser takes up 15% of the footprint of the equivalent steam condenser, but weighs approximately the same.

Air cooled condensers are of interest in many regions where water is of limited availability. Here the lower volumetric flow rate of sCO_2 systems can lead to much lower cost ACCs, since pipe and tubing diameters can be drastically reduced compared to steam. We are finding that the cost of ACC systems for sCO_2 is comparable to that of a WCC system, when including the condenser, cooling tower, pumps and water treatment systems. This stands in contrast to the situation for steam ACC's which are considerably more expensive than their water cooled counterparts.

OPERATION AND MAINTENANCE

Several factors are expected to lead to reduced O&M costs for sCO_2 systems relative to conventional steam-based systems:

- Elimination of water treatment and blowdown disposal systems.
- Reduced turbine wear due to reduced stage count and non-condensing operation.
- Reduced pump impeller wear due to reduced stage count and elimination of cavitation damage.
- Reduced EHX maintenance compared to HRSG due to single-phase operation (no condensate flooding or desuperheater overspray), elimination of boiler drums, a major source of fatigue cracking failures, and reduction in instrumentation and controls complexity.

The reduced capital cost of sCO_2 systems and lower O&M costs will also lead to substantial improvement in levelized cost of electricity (LCOE). A set of example calculated values are shown in Figure 10.

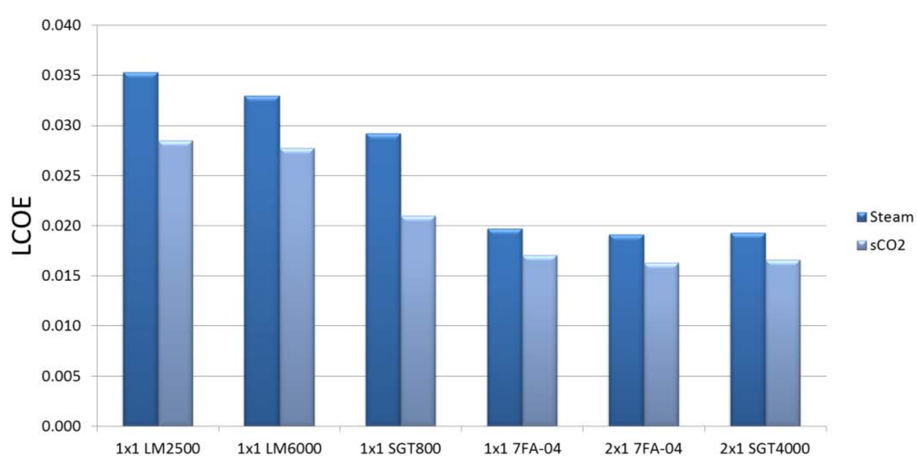

Figure 10: Projected LCOE for bottoming cycle power generation, assuming 5% discount rate, 30 year plant life and 85% utilization factor for several sample applications.

SUMMARY AND CONCLUSIONS

With appropriate cycle design and economic optimization, supercritical CO_2 power cycles can generator more power than existing steam cycles in combined cycle gas turbine applications. The small size of sCO_2 turbomachinery, the simplicity of the exhaust heat exchanger, and the potential to operate in entirely water-free environments result in reduced plant footprint, simplified and flexible installation and operation. With continued development in advanced heat exchangers, and further developments that occur with product maturity, the advantages of using sCO_2 cycles as high efficiency, low cost and low environmental impact power generation systems will become even clearer.

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