The 5th International Symposium - Supercritical CO_2 Power Cycles
March 28-31, 2016, San Antonio, Texas

A Comparative Study of Heat Rejection Systems for sCO_2 Power Cycles

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ABSTRACT

All closed-loop power cycles must reject low-grade residual heat to the environment, and sCO_2 cycles are no exception. Heat rejection systems are a major cost element of a full sCO_2 power cycle installation, and their performance can have a significant impact on cycle performance. Water-cooled systems have traditionally been used preferentially over air-cooled heat rejection systems due to their perceived lower costs and improved overall cycle performance. However, since one of the primary advantages of sCO_2 power cycles over their steam-based counterparts is to operate in water-restricted environments, air-cooled systems extend that advantage even further. In this paper, designs of these two heat rejection options are created for a 10MWe notional power plant. Parameters studied include overall system performance for several different climates, parasitic power consumption, capital cost, operating and maintenance costs, reliability and physical footprint. For many installations, air cooling provides a superior heat rejection option, which provides a fully water-free power plant.

INTRODUCTION

All closed-loop thermodynamic cycles must reject low-grade residual heat to the environment. Historically, locally available water sources were the most common medium for heat rejection, as they represented a high capacity, low temperature heat sink that resulted in the most favorable thermodynamic performance. However, discharge of heated water back into waterways frequently resulted in negative ecological impacts to local marine life, and therefore this method is generally unavailable for new power plants.

Today, the most common cooling method for new power plants is a semi-closed loop of water, which is circulated to and heated by the thermodynamic cycle, and then cooled by evaporative means in an open cooling tower [1]. Water is required to replace losses due to evaporation and drift (non-evaporated water that escapes the cooling tower), and to limit the buildup of minerals in the cooling circuit by intentional drainage of a portion of the circulated water (commonly referred to as "blow-down").

Cooling of the thermodynamic cycle can also be performed by heat transfer from the working fluid to air, commonly called "dry cooling." Because air has a much lower heat capacity and density than water, the equipment required for an air-cooled solution is generally larger than that used for water cooling. Also, while an evaporatively-cooled system can approach the wet-bulb temperature of the environment, a dry-cooled system is limited to the dry-bulb temperature, which is always greater than or equal to the wet bulb temperature. Because the thermodynamic cycle's performance is improved with lower heat rejection temperature, an evaporatively-cooled system has a theoretical advantage over the dry-cooled system.

Supercritical carbon dioxide (sCO_2) power cycles are an interesting test case for comparisons of these two cooling technologies. With water becoming an increasingly scarce resource in many locations, sCO_2 cycles offer a water-free alternative to steam Rankine cycles. However, the advantage is largely eliminated if wet-cooling is utilized, as the water consumption for the cooling system is substantially larger than the water consumption rate of the basic steam Rankine cycle. Thus, dry cooling in combination with the sCO_2 cycle is particularly attractive as a truly water-free power solution.

However, due to the thermodynamic properties of carbon dioxide, the performance of sCO_2 cycles tends to be more strongly affected by heat rejection temperature than steam Rankine cycles. Thus, the higher heat rejection temperature that dry-cooling can achieve would appear to be a significant handicap.

In this paper, we examine two test cases for a 10MWe heat recovery sCO_2 cycle utilizing wet and dry cooling. The relative performance of the two alternatives will be compared under a range of ambient conditions and applied to several exemplar climates. Other considerations that will be discussed are the relative capital, operation and maintenance costs, and the physical footprint of the two cooling systems.

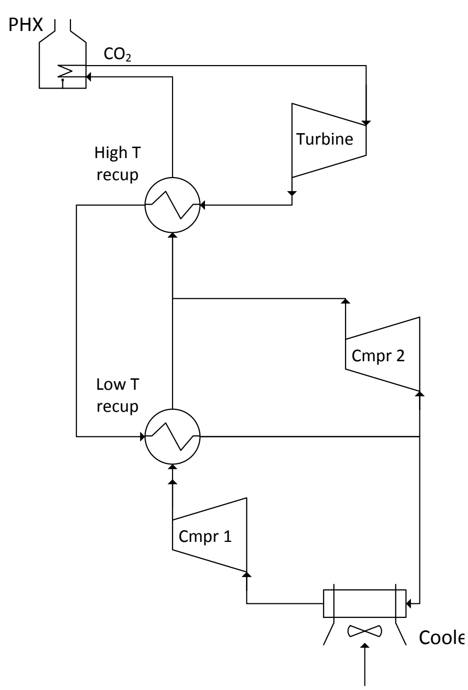

Figure 1: Process Flow Diagram of Recompression Brayton Cycle

SYSTEM COMPARISONS

The baseline system is a proposed cycle for the DOE Supercritical Transformational Electric Power (STEP) test facility (Figure 1), the "Recompression Brayton Cycle" (RCBC). The cycle features staged recuperators, two compression devices in parallel, a single power turbine, and heat addition and rejection heat exchangers [2]. For purposes of this study, the cycle is constrained to a maximum operating pressure of 25 MPa, and turbine inlet temperature of 700°C. For the test facility, a nominal net electrical output of 10MWe is targeted. This cycle is intended to be representative of an advanced power plant, whether using concentrated solar, nuclear or fossil-fuel thermal input, and targets a net efficiency of 50% at large scale.

Cycle performance is calculated as a function of the heat sink temperature, with the results shown in Figure 2. The heat exchanger, whether air- or water-cooled, is modeled as a counter-flow device, discretized into 20 zones. Fluid properties are calculated at the boundaries of each zone, and assumed constant within each zone. This approach provides a reasonable representation of the variation in CO_2 properties across the temperature range (Figure 3), and was successfully used to evaluate the performance of the water-cooled condenser/cooler (WCC) in a large-scale sCO_2 system test. Note that this approach is also used on the air-cooled condenser/cooler (ACC), even though it is a multi-pass cross-flow heat exchanger. However, the number of passes (6) is large enough that the performance closely approaches that of a counter-flow heat exchanger [3]. A more detailed description of the model assumptions and solution process is given in Reference [4].

Two cases are shown in Figure 2. In the first, the cycle pressure at the compressor inlet is forced to remain above the critical pressure (7.50 MPa minimum vs 7.38 MPa critical pressure). In the second, the

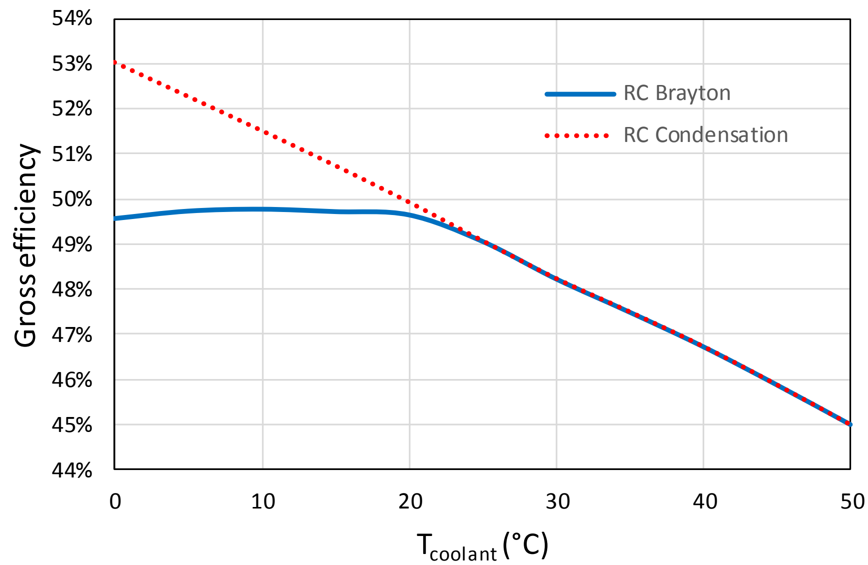

Figure 2: Cycle performance as a function of coolant temperature. The "RC Brayton" cycle forces supercritical compressor inlet pressure at low coolant temperatures. The RC Condensation cycle allows subcritical pressure at the compressor inlet.

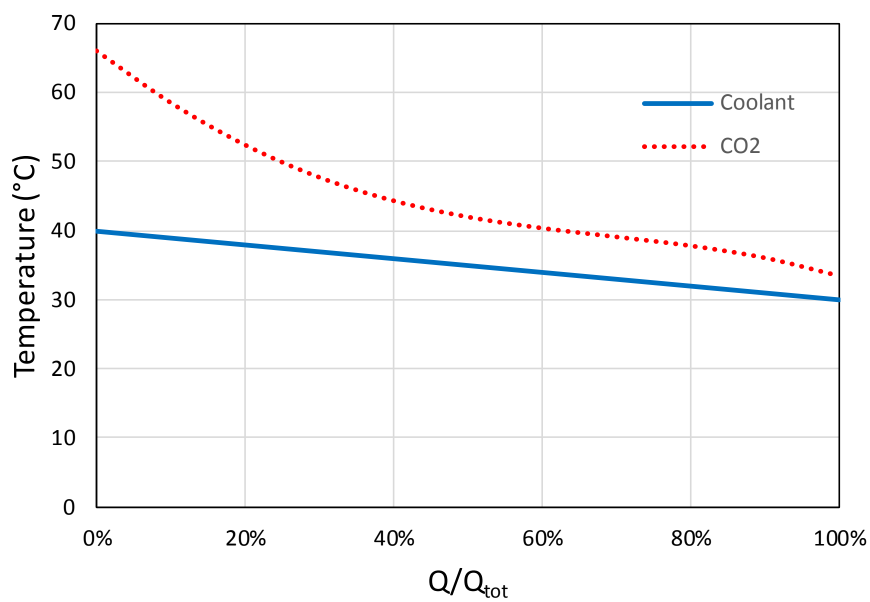

Figure 3: Temperature as a function of heat transferred within the condenser/cooler.

compressor inlet pressure is allowed to decrease below the critical pressure, and is set to a value that maximizes the cycle efficiency, while maintaining at least 1.0 MPa of margin above the CO_2 vapor pressure at the low temperature compressor inlet temperature. This second case is termed a “condensation cycle,” since the fluid state at the low temperature compressor can actually be a liquid. At coolant temperature above 20°C, both cycles switch to a compressor inlet map of pressure as a function of temperature that maximizes cycle efficiency.

The reference sizing case for the ACC, WCC and associated equipment is a dry bulb temperature of 30°C and a wet bulb temperature of 24°C, and a heat rejection load of 10.8MW.

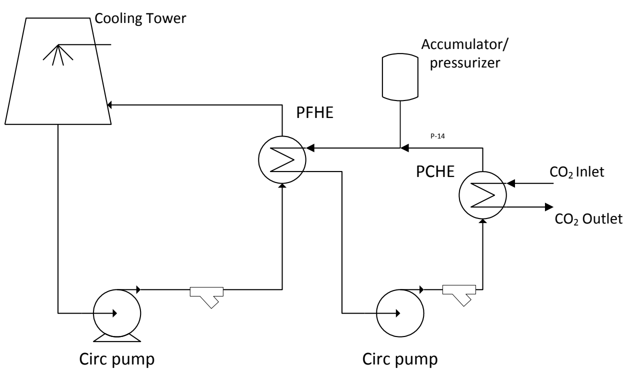

Figure 4: Process flow diagram for wet-cooled heat rejection.

Wet-cooled system

A process flow diagram of the wet-cooled system is shown in Figure 4. The cooling tower for this size of a system is typically a field-erected unit constructed of wood and fiberglass, with a concrete basin. The physical size and cost of the cooling tower are strong functions of the total heat rejection load and of the desired approach temperature – the difference between the water temperature leaving the cooling tower and the local wet-bulb temperature. The cooling tower performance model is based on the KaVL method, with an empirical model for KaVL as a function of water and air mass fluxes [5]. The cooling tower cost model is based on Zanker’s correlation [6], with an inflation adjustment to 2015 currency values. An example of cooling tower cost versus specified approach temperature is shown in Figure 5. Suppliers are generally reluctant to quote approach values below 3°C due to measurement uncertainty impacts on calculating performance [7], therefore this is the lower limit for used for this study.

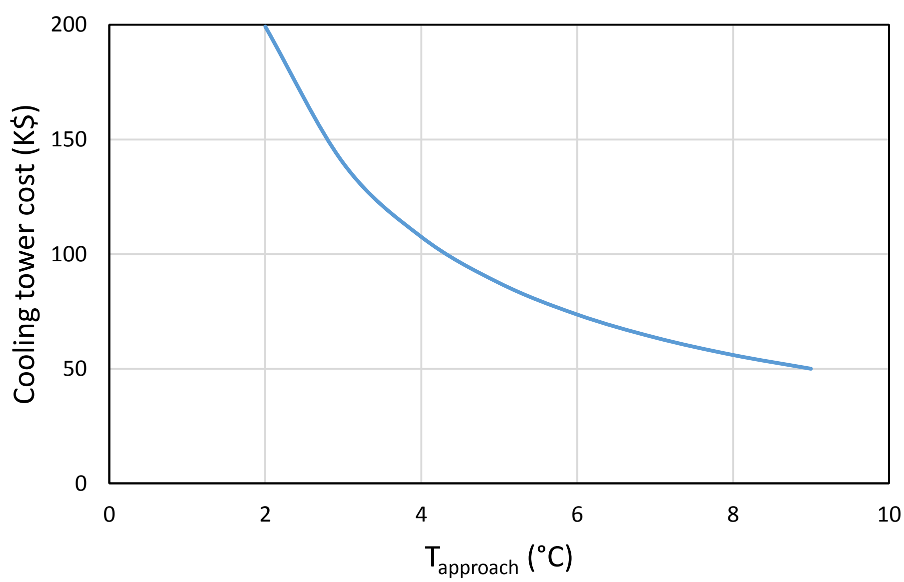

Figure 5: Cooling tower cost vs. approach temperature (T_{w_out} - $T_{wetbulb}$). Note that 3°C is generally considered to be a practical lower limit for this parameter

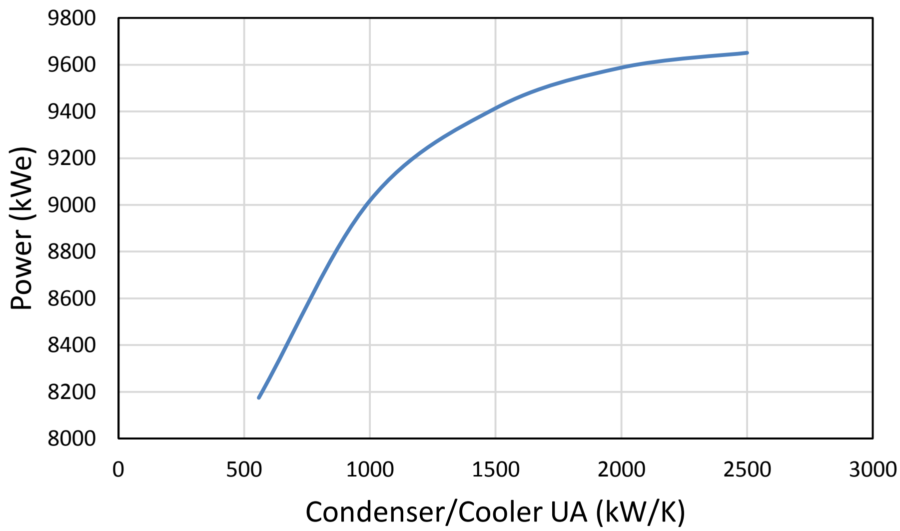

Figure 6: Cycle net power output as a function of the condenser/cooler conductance.

The water-cooled CO_2 cooler (WCC) heat exchanger is modeled as a Printed Circuit Heat Exchanger (PCHE) [8]. At present, the only commercially-available alternative configuration that can operate at the high pressures (up to 12MPa) of the low pressure side of the sCO_2 power cycle is the shell and tube (S&T) heat exchanger. However, the excessive size and cost of the S&T precludes it from serious consideration in this application.

System performance is strongly impacted by the sizing of the WCC (Figure 6). A UA value of 1500 kW/K (discretized basis) was selected as a reasonable balance between WCC size/cost and system performance. A temperature range (difference between the coolant temperature leaving and entering the WCC) of 10°C was selected, which establishes the required water flow rate and is consistent with standard practice for cooling tower water temperature. This value also provides a reasonable temperature distribution within the WCC (Figure 3), avoiding a pinch point at the inflection point of the CO_2 temperature curve. The estimated WCC cost is based on a calculated physical size and material cost factor [9].

The intermediate closed cooling loop shown between the open cooling tower water loop and the PCHE (Figure 4) adds cost and complexity. However, the small cooling passages within a PCHE are particularly susceptible to contamination – therefore the manufacturer strongly recommends against using an open loop cooling system with their heat exchangers [10]. Echogen's test experience with PCHEs directly coupled to cooling towers supports the use of the intermediate cooling loop as well. Although several months of such operation had no adverse impact on the measured PCHE performance, a brief period of slightly poor water quality as indicated by a 20% increase in water conductivity resulted in a large impact on PCHE performance.

The intermediate cooling loop consists of a plate and frame heat exchanger (PFHE), a circulation pump and filter. The closed loop is filled with demineralized water, or a glycol-water mixture if sub-freezing temperatures are likely to be encountered. For the purposes of this study, a 3°C approach temperature is assumed for the PFHE, and a balanced capacity factor (mass flow rate x heat capacity) for both sides of the PFHE, resulting in a required UA of 3600 kW/K. Cost estimates are based on supplier quotes for similar units.

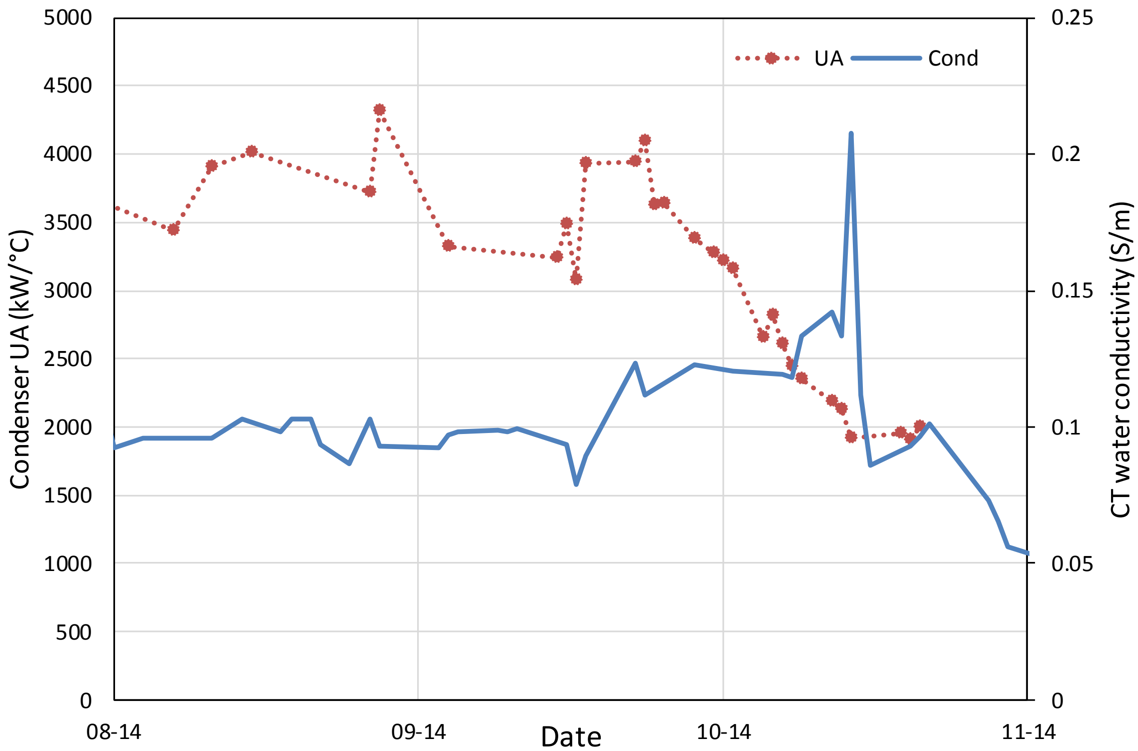

Figure 7: Measured PCHE performance and cooling tower water conductivity versus time during factory testing of the EPS100.

Air-cooled system

For an air-cooled configuration, CO_2 is circulated directly to several banks of fin-fan heat exchangers, which consist of multiple banks of finned tubes with air flowing across the tube banks driven by electrically-driven fans. The configuration of the tube banks can either be a flat bank with low velocity fans (Figure 8), or a V-shape with higher velocity fans. The V configuration has a smaller physical footprint, but is somewhat higher cost for the same capacity as the flat configuration. For this study, the flat configuration is assumed.

The same analysis performed for the WCC system is used to select the conductance of the ACC, with a UA value of 1500 kW/K (discretized basis) again selected to balance cost, size and performance, and a 10°C range selected consistent with supplier recommendations. The ACC cost and fan power values in this study are based upon several suppliers' estimates for similarly-sized systems.

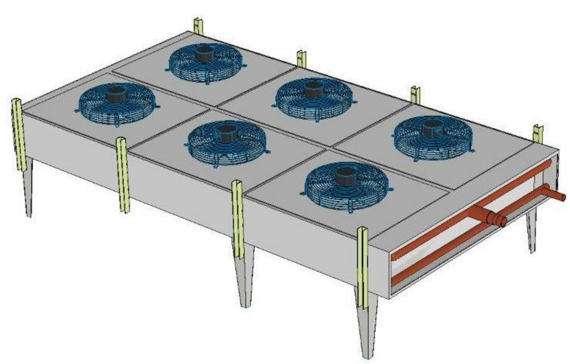

Figure 8: Fin-fan air cooler

PERFORMANCE COMPARISONS

As shown in Figure 2, cycle performance depends strongly upon the coolant temperature delivered to the ACC or WCC. In the case of the ACC, the coolant temperature is simply the local dry bulb temperature. For the WCC, the coolant temperature needs to be calculated starting from the local wet bulb temperature, taking into account the performance of the cooling tower, and the intermediate closed cooling loop.

Cooling tower off-design performance is modeled by varying the approach temperature until the calculated tower size equals the design sizing. For most cases, the fan power is assumed to be the limiting factor controlling air flow rate through the tower. For cases where the wet bulb temperature begins to approach 0°C, fan power is reduced until the water temperature leaving the tower basin is 15°C. This practice reduces the potential for ice buildup on the tower, which can result in significant damage if the accumulation exceeds the tower's mechanical capacity [7].

An example calculation of cooling water temperature as a function of wet bulb temperature is shown in Figure 9 for three different US cities, representing three different climates. The three curves were all calculated for the same cooling tower design; however, the tower performance depends secondarily upon dry bulb temperature, thus resulting in slightly different performance versus wet bulb temperature. The intermediate closed loop is assumed to operate at a constant temperature approach to the cooling tower water of 3°C.

Climate has a strong influence on the relative performance of air and water cooled systems. Wet bulb and dry bulb temperatures are not uniquely related to each other, with the difference arising from local climate variation, particularly in average humidity levels. The low humidity of a desert environment, represented here by Phoenix, AZ, results in a much lower wet bulb temperature for the same dry bulb temperature as a humid coastal environment (e.g., Houston, TX). In addition, the probability distribution functions of both wet and dry bulb temperature vary greatly by locality. To model these differences, hourly temperature and humidity observations were collected for a full year (2014) from these two cities plus Akron, OH as an exemplar temperate climate. The probability distribution function of wet bulb temperature and the mean coincident dry bulb temperature were used to calculate WCC system performance, while the probability distribution of dry bulb temperature was used to calculate ACC system performance.

Example calculations are shown for Houston, TX in Figure 11. For this case, the net power output of the air-cooled system would exceed that of the water-cooled system for all but the hottest 7% of the

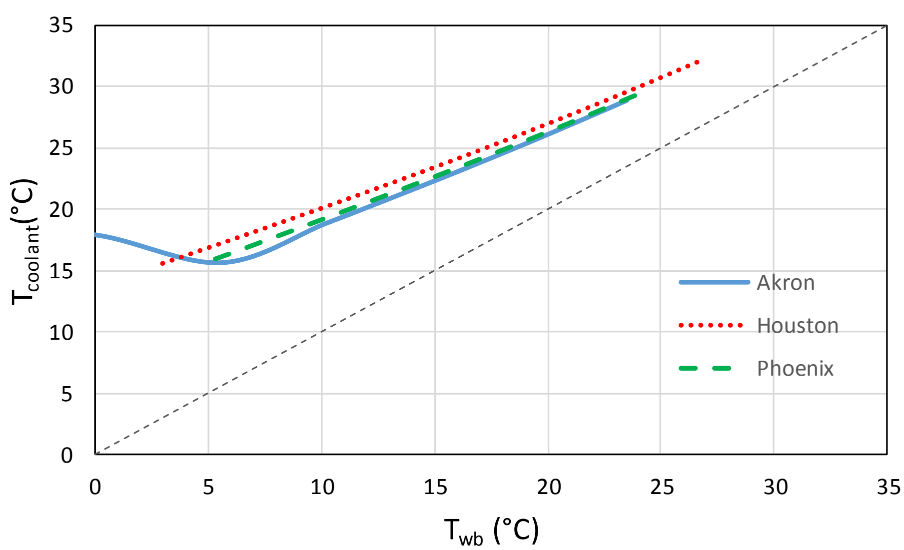

Figure 9: Coolant temperature as a function of wet bulb temperature for three US cities.

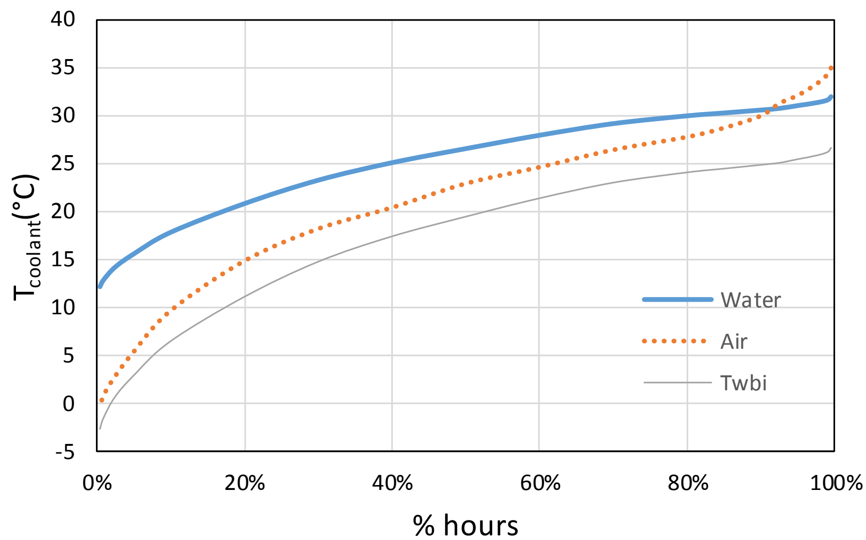

Figure 10: Coolant temperature as a function of the cumulative percentage of hours below that temperature. Wet bulb temperature is shown for reference. Climate data are from Houston, TX.

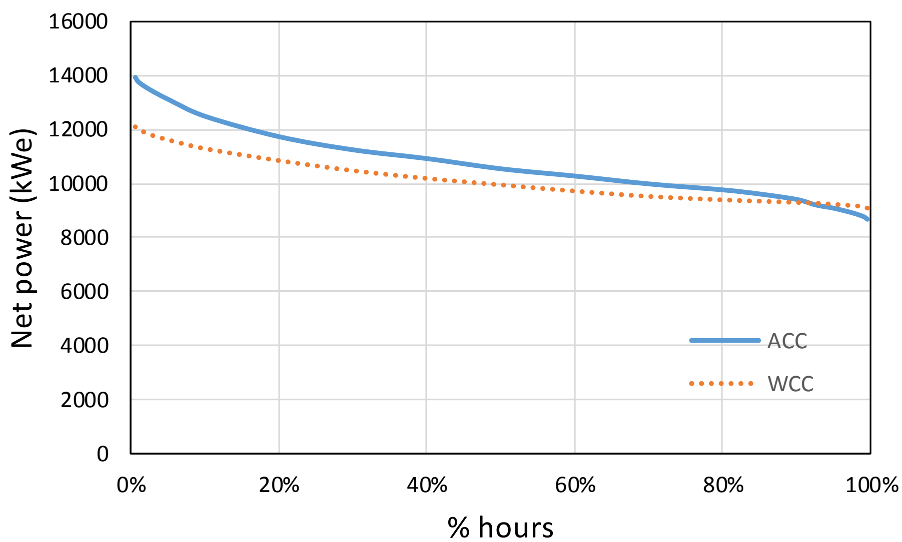

Figure 11: System net power output as a function of cumulative hours. Houston, TX, condensing cycle.

conditions encountered in Houston for the year 2014. For the hottest 40 hours of the year, the water-cooled system would generate approximately 377 kW, or 4.3% more power than the air cooled system.

The annual average power output for each system can be calculated by integrating the distribution functions in Figure 11 (Table 1). For Houston and Akron, significantly more power can be generated on an annualized basis by an air-cooled system, with both the condensing and supercritical cycles. For the desert climate, the water-cooled system outperforms the air-cooled version, although due to water supply limitations, the water-cooled system would not likely be selected regardless of its performance benefits.

Table 1: Annualized average performance

City	Supercritical		Condensing	
	WCC	ACC	WCC	ACC
Akron	10,136	10,647	10,215	11,922
Houston	9,999	10,434	10,024	10,719
Phoenix	10,571	9,916	10,647	10,068

CAPITAL COST

A summary of the projected equipment costs is shown in Table 2. The major equipment cost estimate sources have been described in the previous section. Pump and piping costs are based on actual prices for similar equipment. The cooling tower and ACC are assumed to be approximately 75 m from the main power cycle. To within the uncertainty of the cost estimates (roughly ±20%), the WCC and ACC solutions have essentially the same capital cost.

The estimated footprints of the cooling tower and ACC are also shown. For extremely space-constrained applications, WCC will have a slight advantage. However, the increased land cost for the ACC will be negligible in most cases, as the difference is a small fraction of a single acre.

Table 2: Equipment and installation cost estimates for air-cooled and water-cooled systems

	WCC	ACC
Cooling tower/ACC	$ 143,000	$ 695,000
Water treatment	$ 20,000	
CT Pumps	$ 44,000	
CT/ACC Piping	$ 41,000	$ 108,000
PFHE	$ 83,000	
PFHE Pumps	$ 46,000	
PFHE Piping	$ 16,000	
Filters	$ 5,000	
WCC PCHE	$ 300,000	
Total eqpt	$ 698,000	$ 803,000
Installation	$ 419,000	$ 482,000
Total	$ 1,117,000	$ 1,285,000
Footprint (m^2)	97	240

Table 3: Estimated auxiliary loads (kWe)

Load	WCC	ACC
Fans	46	310
CW pump	99	
PFHE pump	112	
Total	257	310

AUXILIARY LOADS

The estimated auxiliary loads for the two solutions are shown in Table 3. Even though the ACC has a significantly larger fan power requirement than does the cooling tower, the incremental water pump work largely offsets the fan power advantage.

OPERATION AND MAINTENANCE

A summary of projected operation and maintenance costs, other than the electrical power consumption accounted above, is listed in Table 4. Water consumption and disposal costs are based on recent industrial customer average rates [11]. When including treatment costs, water-related items represent 90% of the total O&M costs of a WCC system, while the air cooled system costs are confined to periodic motor and belt maintenance, and heat exchanger external cleaning.

Table 4: Projected annual operation and maintenance costs.

	WCC	ACC
Water consumption	$ 156,000	
Water disposal	$ 57,000	
Water treatment	$ 25,000	
Chemical replacement	$ 65,000	
Pump maintenance	$ 24,000	
Fan maintenance	$ 7,500	$ 30,000
Maintenance cost/yr	$ 334,500	$ 30,000
$/kWh	$ 0.0042	$ 0.0004

CONCLUSIONS

Given the water constraints in many locations, one of the key advantages of sCO_2 systems relative to competing steam Rankine cycles is the elimination of water usage from the power cycle. This advantage will only be realized in a dry-cooled system utilizing an ACC. In many climates, the performance of an ACC will be superior to a wet-cooled system for the majority of conditions. The climates which demonstrate the largest performance advantage of a wet-cooled system are those with high dry bulb temperatures and low wet bulb temperatures, precisely those which are most likely to be water-constrained. In addition, dry-cooled systems will realize significant operation and maintenance cost advantages, mainly due to the elimination of water treatment equipment and cost.

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