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Techno-Economic Analysis of a Geothermal sCO_2 Thermosiphon Power Plant

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ABSTRACT

Geothermal energy provides clean, baseload electricity generated from the heat of the Earth. It has huge market potential but comprises less than 1% of the U.S. utility electricity generation market today as it is economically not scalable. Recent research and demonstration projects have investigated using CO_2 as a working fluid as it has a near-ambient critical temperature and exhibits large changes in density with temperature change, and thus can circulate without mechanical pumping through a thermosiphon. In addition, use of sCO_2 can significantly increase utilization efficiency of the heat conversion cycle while reducing upfront capital cost mainly due to the small size of the turbomachinery equipment. The current project used a small-scale flow loop to calibrate and validate the thermodynamic model of a sCO_2 thermosiphon. The reduced order physical modeling tool (GeoTwin™) then used these results to simulate and optimize a closed loop geothermal system. The model provided expected pressures, temperatures, and flow rates for a surface sCO_2 turbine. This data was then used to investigate the surface capital costs

for such a geothermal system, including the turbine and required cooling equipment. Results show that a radial turbine is well suited for the mid-enthalpy sCO_2 inlet conditions (80-150°C). The analysis also shows that the cooling of the process fluid before reinjection is the largest challenge for the techno-economics of the power cycle and represent much larger costs than the sCO_2 turbine. However, the cost of cooling is comparable to present data geothermal ORC cycles while greatly reducing the cost of the power generation equipment.

INTRODUCTION

Current geothermal power production occurs with one of two different cycles. The first is a flash steam cycle, where hot water from within the earth is converted, or flashed, into steam and run directly through a steam turbine [1]. The second is a binary plant, where the heat from the produced water is transferred to a separate fluid, which has its own closed loop power generation cycle [2]. Both operating systems have technical benefits and challenges based on the downhole conditions.

The current study investigates the techno-economics of a newly proposed power plant based on a sCO_2 thermosiphon, similar to that proposed by Atrens et al. [3]. The proposed study looks at a formation producing a brine/methane mixture, as shown in Figure 1. Cold dense CO_2 is injected into the well through an outer annulus. The large density in the injection leg results in a high hydrostatic pressure at the bottom of the well. The fluid is insulated from the formation on the injection leg to maintain the high fluid density. The CO_2 then flows through an inner annulus that is insulated from the outer annulus but transfers heat from the produced brine mixture. As the CO_2 heats up, density is reduced, thus maintaining flow through a thermosiphon. The CO_2 then exits the annulus and flows through a turbine, converting thermodynamic work to electrical work. The expanded CO_2 is then cooled to the dense phase for reinjection into the well. The brine exiting the formation is reinjected to a neighboring injection well to maintain the reservoir pressure or merely for disposal purposes. The thermosiphon has the benefit of eliminating the pump or compressor (needed for binary cycles) except during initial flow commencement by using gravitational potential to generate the working fluid's mass flow, which improves thermal efficiency. Additionally, the sCO_2 density in the turbine is high relative to other fluids allowing for reduced capital cost for the turbine itself.

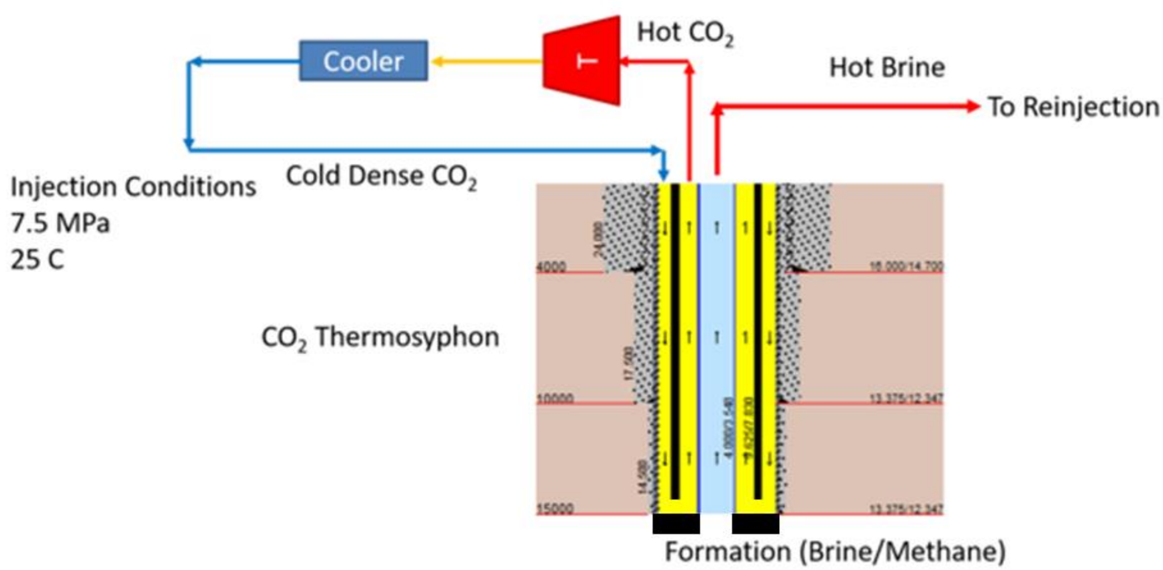

Figure 1. Schematic of the Geothermal Thermosiphon Configuration Indicating the CO_2 Injection Leg, CO_2 Production Leg, Surface Equipment, and Brine Production Leg

The team used a reduced order physics-based model for simulating the CO_2 thermosiphon and all heat exchange processes. The model follows a similar approach used by Atrens et al. [4]. Specifically, the well is discretized into short lengths, and the change in pressure and change in enthalpy is calculated for each discretized step, utilizing a constant flow rate (conservation of mass). The pressure changes from the change in gravity potential and friction losses are calculated using equations 1 and 2 as follows.

$$\Delta P = \rho g \Delta z - \Delta P_f \qquad \text{Eq. 1}$$

$$\Delta P_f = f \frac{\Delta z}{D} \rho \frac{V^2}{2} \qquad \text{Eq. 2}$$

Here, ρ is the density of the fluid at the inlet, Δz is the change in height, V is the velocity, D is the hydraulic diameter, and f is the friction factor calculated using the Colebrook equations. The change in enthalpy is calculated as follows.

$$\Delta H = g\Delta z + Q \qquad \text{Eq. 3}$$

where Q is the heat transfer from the rock formation, or between two concentric tubes. The model uses a concurrent/counter flow heat exchanger equation to calculate the heat transfer between the different fluid streams (LMDT) method. The heat transfer is calculated using an energy balance for each stream which includes convective heat transfer at the wall (Nusselt number calculation) and the conductive heat transfer based on the medium. An iterative algorithm is utilized to determine the outlet temperatures until the solution matches the discretized length for the simulation. Heat transfer occurs between the formation and the outer casing, with the cement providing some insulation. Heat transfer occurs between the outer and inner annuli (CO2 streams) with vacuum insulated tubing providing some insulation. Heat transfer occurs between the inner annulus and the production stream through the steal tubing. The work of the turbine and cooling required for the system are calculated using the CO_2 fluid properties coming out of the well (equations 4 and 5) as follows:

$$\dot{W}_{Turbine} = \dot{m}\,(h_{turbine_{in}} - h_{turbine_{out}})\eta_{turb} \qquad \text{Eq. 4}$$

$$\dot{Q}_{Cooler} = \dot{m}(h_{turbine_{out}} - h_{injetion}) \qquad \text{Eq. 5}$$

Here, the turbine efficiency is assumed to be 85%. The parasitic load of the coolers depends on the cooling system selected. The model is used to predict the technical potential of the sCO_2 cycle though the net power produced. Additionally, the team investigated the cost of installing this system. The economic analysis was completed by performing the following steps:

- Predicting sCO_2 turbine cost based on previous designs
- Receiving vender quotes for
  - Dry Coolers
  - Adiabatic Coolers
  - Chillers
  - Water Towers
  - Gearboxes
  - Generators
  - Bearings
  - Machining Costs
- Comparing cost to predictions made from Weiland et al. [5]

Drilling costs were estimated using a cost per foot for a given well depth and hole diameter. Currently, a cost for a reinjection well is not included, but, a thermosiphon can also be utilized on the reinjection well to capture the remainder of the heat in the brine at a similar cost and power production as the injection well. The simulated net power output and estimated capital costs are combined to estimate the Levelized Cost of Energy (LCOE) as:

$$LCOE = \frac{(Capital\ Cost * CRF) + fixed\ O\&M\ cost}{8760 * Capacity\ Factor} + Variable\ O\&M\ cost \qquad \text{Eq. 6}$$

$$CRF = \frac{i(1+i)^n}{(1+i)^n - 1} \quad \text{Eq. 7}$$

where i is assumed to be 8%, n is the lifetime of the plant (assumed to be 30 years), 8760 is the total number of hours in a year, the capacity factor is the net power multiplied by the yearly capacity (assumed to be 90%), variable O&M cost is assumed zero, the fixed O&M cost rate is assumed to be $0.014/kWh, and the capital cost is the estimated capital cost of the system. The simulations were performed assuming a 14'' OD production casing, an injection depth of 4,500 m, and a formation temperature of 200°C.

RESULTS AND DISCUSSION

We assumed ambient air conditions of a dry bulb temperature of 35°C and a wet bulb temperature of 21.1°C for sizing an adiabatic cooler that cools the CO_2 to 25°C (used for costing and parasitic loads) and an injection pressure of 7.5 MPa (near the critical pressure). The thermosiphon is heavily depended on ambient temperatures. For purposes of simplicity, an average dry bulb and wet bulb temperature was utilized, but future simulations will include year long simulations based on changing ambient conditions. Simulations are performed from mass flow rates ranging from 25-75 kg/s of CO_2 flow where the turbine pressure ratio would set the mass flow. It is assumed that the turbine and drilling costs are fixed regardless of the flow rate, because the turbine is designed for the maximum power (3MW) and because the well geometry does not change. The cost of the coolers is dependent on the amount of cooling required, because one can install the required number of cooler bays (at this scale, modular scale up is required). Figure 2 shows the simulation outputs for cooling duty, turbine power, the parasitic load of the coolers and the net power. The power output of the turbine follows a second-order polynomial, which increases with flow rate until it hits a peak near 70 kg/s, and then decreases. The mass flow increases the power output until the friction forces of the closed loop restrict the pressure ratio across the turbine. The cooling duty increases continually with mass flow and follows a power law. The cooling parasitic load is proportional to the number of coolers and therefore, is proportional to the cooling duty. The cooling parasitic is removed from the turbine power to estimate net power output.

Figure 3 shows the turbine inlet conditions based on flow rate. The turbine inlet pressure continually decreases with increased flow rate due to the induced friction losses. The shape is a second-order polynomial because the friction is proportional to velocity squared, V^2. The temperature has a small difference dependent on the mass flow rate. The peak occurs near 53 kg/s.

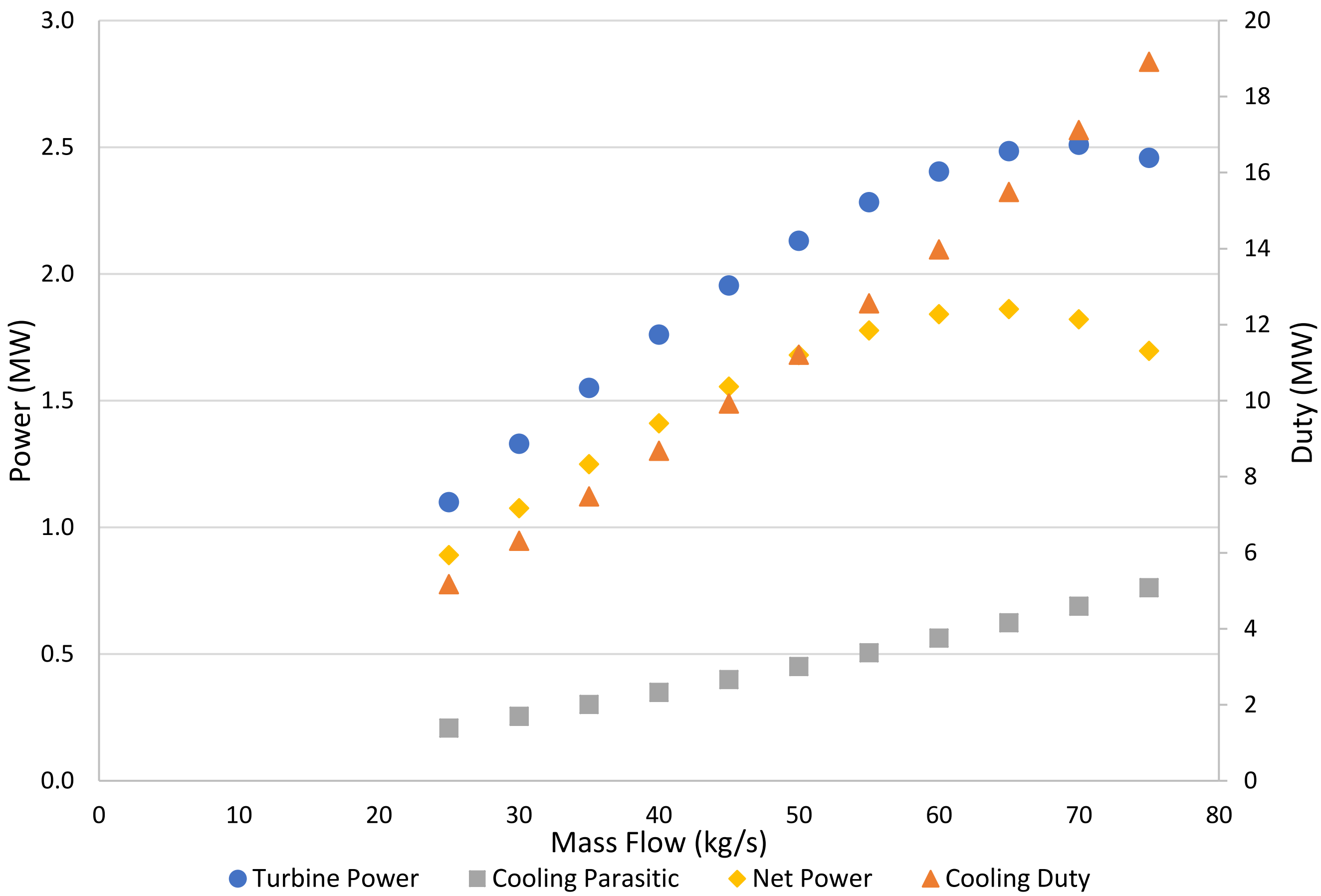

Figure 2. Simulation Output for Power and Cooling Duty

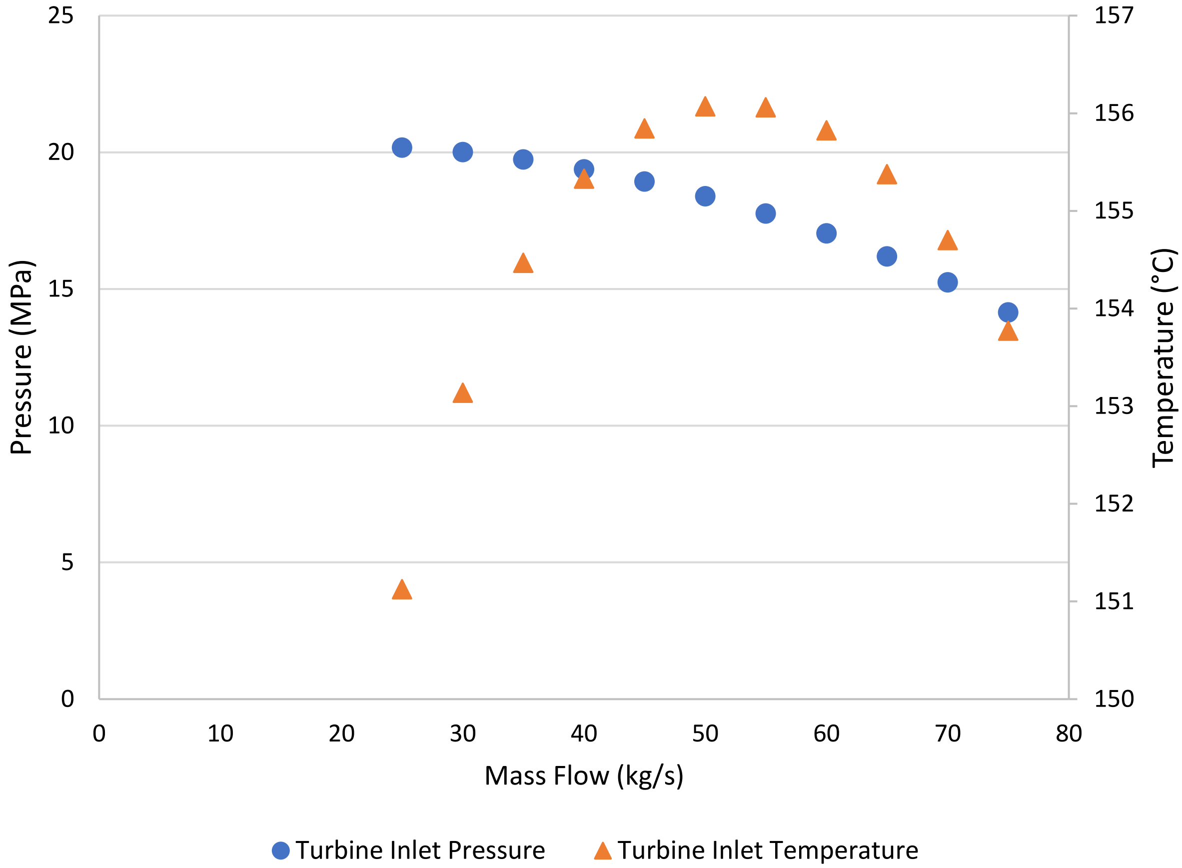

Figure 3. Simulated Turbine Inlet Conditions

The geothermal sCO_2 thermosiphon model outputs were used to estimate capital cost and LCOE. The capital cost is made up of the turbine (including gearbox and generator), cooling costs and well construction costs, including drilling and casing. Figure 4 shows the estimated capital cost, cooler cost, and LCOE for the simulations. The increase in capital cost and cooling cost is proportional to the duty, and therefore follows a power law correlation with mass flow rate. The LCOE reaches a minimum of $0.062/kWh at 60 kg/s.

One of the unexpected findings of the study is that the peak maximum or minimum of the turbine power, net power, and LCOE all occur at different flow rates. The peak turbine power occurs at 70 kg/s, while the peak net power occurs at 65 kg/s, and the minimum LCOE occurs at 60 kg/s. This exemplifies the importance of clearly defining the goal of an optimization process when designing a sCO_2 thermosiphon for geothermal applications. In the end, the total system LCOE should be used to determine the best operating flow rate for maximum LCOE (assuming normal market conditions), which creates the best return on investment.

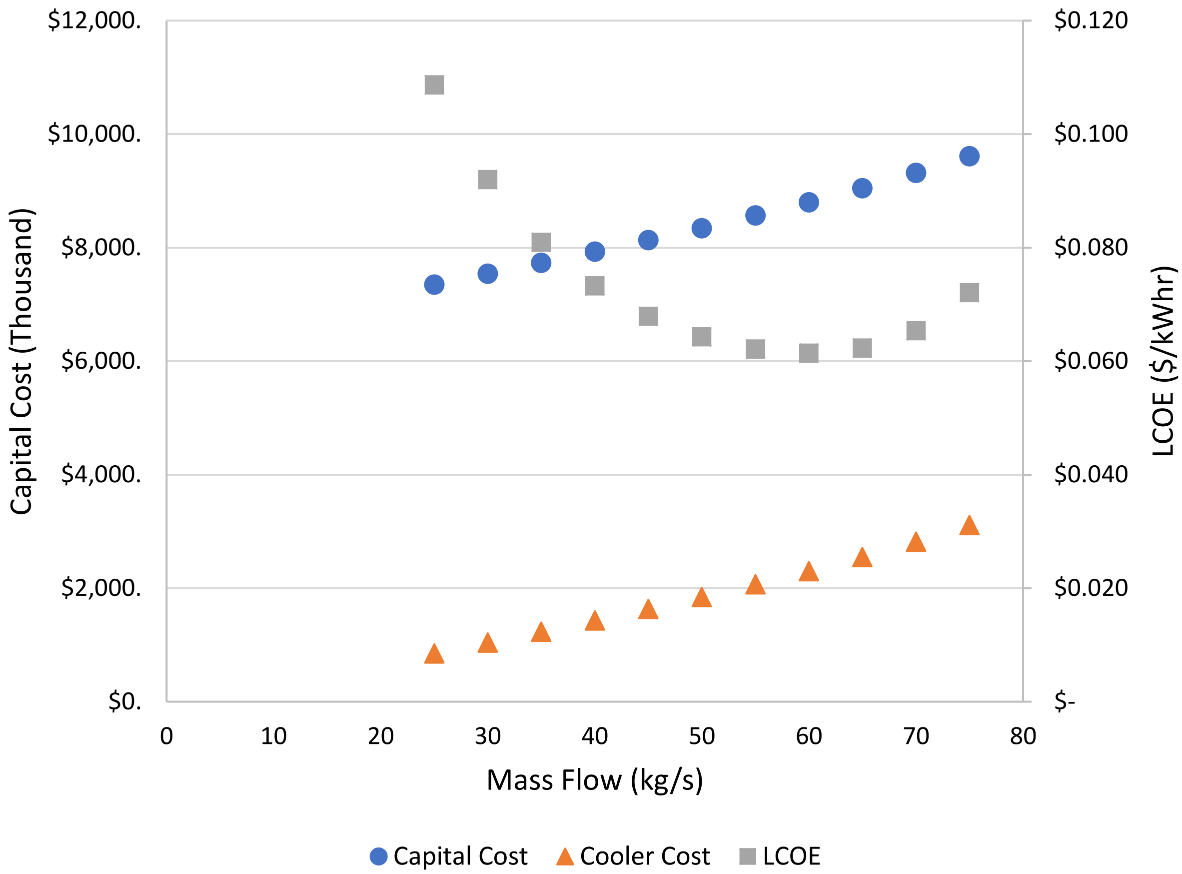

Figure 4. Estimated Capital Cost and LCOE

The LCOE was estimated for costs of a one-off system, and not a system that is manufactured at production quantities during scale-up. The turbine cost could be reduced through simple bulk manufacturing of a standardized design. The cost of drilling is expected to reduce when drilling a pad with multiple wells of standardized design, rather than a single well (this has been shown in the oil and gas industry, especially in the unconventional shales). The cost of a cooling system might be reduced through a bulk order and price negotiations. Therefore, a sensitivity study was performed for each of the main capital cost components, including drilling and casing, the turbine, and the coolers. A 30% reduction in turbine costs reduced the LCOE from $0.062/kWh to $0.060/kWh. The compact nature of the turbine makes it the smallest capital cost contributor. A 30% reduction in drilling and casing costs would reduce the LCOE from $0.062/kWh to $0.052/kWh. A 30% reduction in cooling costs would reduce the LCOE from $0.062/kWh to $0.057/kWh. If a 30% reduction was realized for all three categories, the LCOE would be $0.045/kWh. The LCOE of the geothermal thermosiphon is competitive with many other forms of both renewable energy and fossil fuels, even at the current one-off design prices.

The sCO_2 thermosiphon was also compared to a binary cycle utilizing sCO_2, with the results shown in Table 1. The binary plant reduces the drilling cost by eliminating two annuli from the downhole casing but adds to the surface cost through: (a) the purchase of a pump/compressor and (b) increasing the required cooling duty. Running the configuration with a pump (such that the pump inlet is 7.0 MPa), requires a larger turbine than the 3 MW design, adding some additional capital cost. The extra power produced is taken up through a parasitic load of a compressor or pump. The LCOE for the binary plant utilizing a sCO_2 pump is $0.077/kWh. Running the configuration with a compressor (inlet pressure at 8.5 MPa and inlet temperature at 37°C) reduces the power output of the turbine compared to the pump arrangement, but has a similar compression power parasitic. The LCOE for the compressor configuration is $0.128/kWh. Both binary configurations require more cooling than the thermosiphon, because of the nature of the counter flow heat exchanger (i.e., the brine can be reduced to below the CO_2 inlet to the turbine). The LCOE for the thermosiphon is the lowest, despite the additional costs required for well construction (drilling larger hole and casing costs). The cost of the compressor/pump and heat exchanger are similar to the extra cost for drilling and installing the annuli. Additionally, the binary plants have additional parasitic loads for the additional cooling required.

Table 1. Comparison of Binary sCO_2 Brayton Cycle and Thermosiphon

	Binary Pump	Binary Compressor	Thermosiphon
Turbine Power (kW)	5133	4199	2404
Pump Power (kW)	2503	2678	0
Cooling Duty (kW)	19857	15353	13978
Cooling Parasitic (kW)	839	649	591
Net Power (kW)	1791	872	1813
Compressor/Pump	$2,000,000.00	$650,000.00	$ -
Heat Exchanger	$ 930,008.00	$ 678,734.00	$ -
Turbine	$ 1,300,000.00	$ 1,300,000.00	$ 1,000,000.00
Cooler	$ 3,266,519.48	$ 2,525,601.73	$ 2,299,411.26
Well	$ 3,500,000.00	$ 3,500,000.00	$ 5,500,000.00
Total	$ 10,996,527.48	$ 8,654,335.73	$ 8,799,411.26
LCOE $/kW-hr	$ 0.077	$ 0.128	$ 0.062

Concerning the surface costs, the coolers are approximately 2-3x the capital cost of the turbine and about 10-20x the physical footprint. Significant gains can be realized through reducing cooling capital cost, reducing parasitic loads, and reducing footprint. Therefore, the project team also investigated alternative cooling solutions including:

- Using an absorption chiller from the waste heat of the brine stream
- Installing a recuperator to transfer heat from the outlet of the turbine to the injection, thus reducing the cooling load
- Using an absorption chiller with a recuperator
- Installing a bottoming cycle on the waste heat of the brine
- Adding a refrigeration cycle instead of an adiabatic cooler

Table 2 shows the LCOE results for the different cooling strategies. The absorption chiller removes the parasitic load of the adiabatic coolers at a smaller price and footprint. The downside is that the waste heat stream does not have enough heat for the chiller to take the entire cooling load. This would require an additional cooling tower system, which requires significant water due to evaporation (that will be higher than the adiabatic cooler). This alternative does have a smaller LCOE at $0.059/kWh. Alternatively, a refrigeration cycle eliminates all water consumption (when compared to the adiabatic cooler). The major downfall is that the refrigeration cycle requires a larger parasitic load to run, and greatly reduces the net power output. The recuperator does eliminate the size of the cooling system, but also negatively effects the thermosiphon, which also reduces net power output.

Table 2. LCOE for Alternative Cooling Strategies

	Capital Cost Turbine	Capital Cost Cooling	Capital Cost Well	Total Cost	Net Power (kW)	LCOE/kW-hr
Baseline Thermosiphon	$ 1,000,000.00	$ 2,299,411.26	$ 5,500,000.00	$ 8,799,411.26	1,813.00	$ 0.062
Absorption Chiller	$ 1,000,000.00	$ 1,500,508.63	$ 5,500,000.00	$ 8,000,508.63	1,753.10	$ 0.059
Recuperator	$1,000,000.00	$ 2,716,525.01	$ 5,500,000.00	$ 9,216,525.01	1,300.00	$ 0.091
Absorption/Recuperator	$ 1,000,000.00	$ 1,200,327.00	$ 5,500,000.00	$ 7,700,327.00	960.333	$ 0.105
Bottoming Cycle	$ 1,600,000.00	$ 4,008,620.00	$ 5,500,000.00	$11,108,620.00	1,758.50	$ 0.079
Refrigeration Cycle	$ 1,000,000.00	$ 2,098,981.00	$ 5,500,000.00	$ 8,598,981.00	1,580.18	$ 0.070

Conclusions

This study investigated the techno-economics of a geothermal sCO_2 thermosiphon power plant. The study utilized a physics-based model to simulate the thermodynamic performance of the plant and used cost modeling from previous experience, vendor quotes, and other economic-based studies. The technical and economic aspects were combined to estimate a LCOE. The study found the optimal LCOE (minimum) occurred when running the plant at approximately 60 kg/s, which was a different operating condition than the peak turbine output and peak net power. The thermosiphon was also compared to a binary plant utilizing sCO_2 and was shown to have a lower LCOE due to higher power output and lower surface equipment capital costs (even with a higher drilling cost). Finally, alternative cooling strategies were investigated for their effect on LCOE, which had similar and higher LCOE values when compared to the baseline that assumed adiabatic coolers.

The study shows that the geothermal thermosiphon has a cost competitive LCOE, estimated at $0.062/kWh. This is cost-competitive with both renewable and traditional sources, and provides a clean, renewable option for baseload energy. Future advances, simply by large-scale manufacturing, will further reduce the LCOE. The major techno-economic hurdles appear to stem from the cooling of the CO_2 to the dense phase. The large flow rates require a large cooling duty with a tight pinch to ambient conditions. Future studies will continue to investigate reducing cooling requirements while also looking at alternative cooling solutions that do not utilize large amounts of water.

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