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Geothermal Power Cycle Modeling - Final Report

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1 PROJECT OBJECTIVES

The team at Southwest Research Institute® (SwRI®) was tasked with assessing options for the surface power cycle. Potential cycles were modeled that fit the reservoir predictions provided.

The technical tasks are as follows: Task 2 was the initial modeling of the candidate power cycles, the supercritical carbon dioxide (sCO_2) cycle for on-design conditions over the reservoir production life; Task 3 was interfacing with the work being done on the reservoir and site model, which includes providing feedback to the external team building the reservoir model; Task 4 progressed the sCO_2 power cycle modeling to include estimates of off-design, parasitic loads, and variation in the cycle performance versus well produced temperature.

Per the objectives for Task 2, the team used process software to model cycle performance based on the provided reservoir conditions. Initial modeling began on two candidate power cycles. These systems were evaluated for steady-state, on-design performance. Sweeping over the years of reservoir operation, the team highlighted a potential selection for the design year that maximizes system power production.

Per the objectives for Task 3, the SwRI team interfaced with the reservoir model, providing input on site and reservoir conditions that impact the power cycle modeling. This task is complete, and the SwRI team has received the final reservoir conditions for Task 4.

Per the objectives for Task 4, the SwRI team selected a size for the sCO_2 power cycle equipment. Per direction by the project team, we kept the design to a single speed, but with a compressor that can accommodate a wide range of flows as the turbine inlet temperature changed. This resulted in reduced performance of the sCO_2 system and a profile for power for the single-speed design. Future work can look at a variable-speed design that allows the system to perform better as well as temperature changes.

2 GEOTHERMAL SURFACE POWER SYSTEM MODELING

The results reported in this section are based on the well conditions produced by ResFrac. The well results included the geothermal source mass flow, pressure, and temperature to enter the surface plant over a thirty-year period. The mass flow rate from the producer is shown in Figure 1. Figure 2 shows the temperature at the outlet of the producer well. The data provided by ResFrac was discretized every 30 days over the period. For the initial time zero, the first 6 months were averaged to avoid the initial high flow rates that correspond to fracking the subsurface. The producer wellhead pressure was maintained at 13.8 MPa throughout the simulation. The well conditions shown below were optimized to maximize EGS fluid production while maintaining a feasible source temperature.

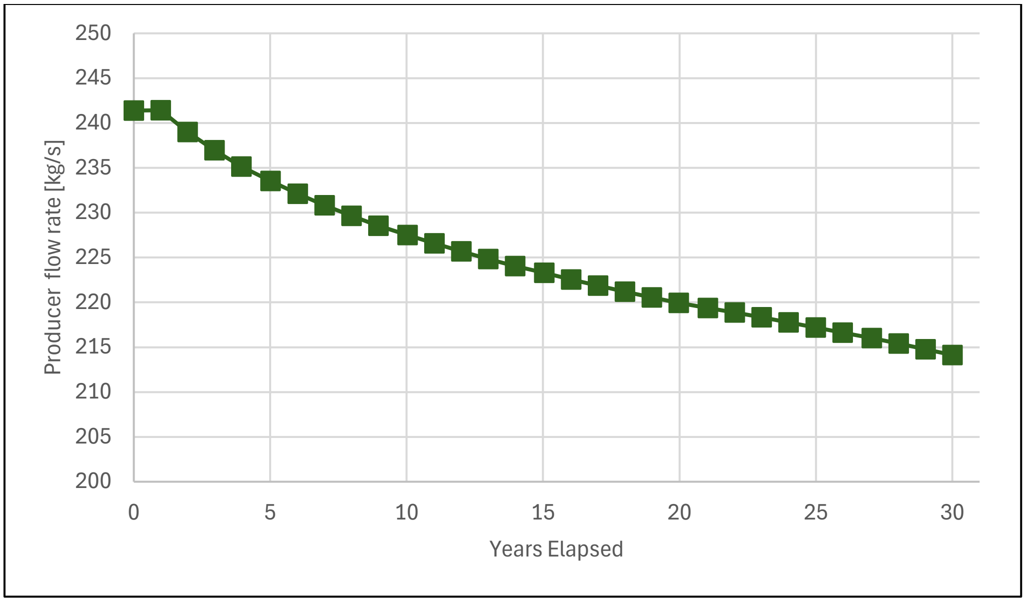

Figure 1. Producer Wellhead Mass Flow Rate over Time

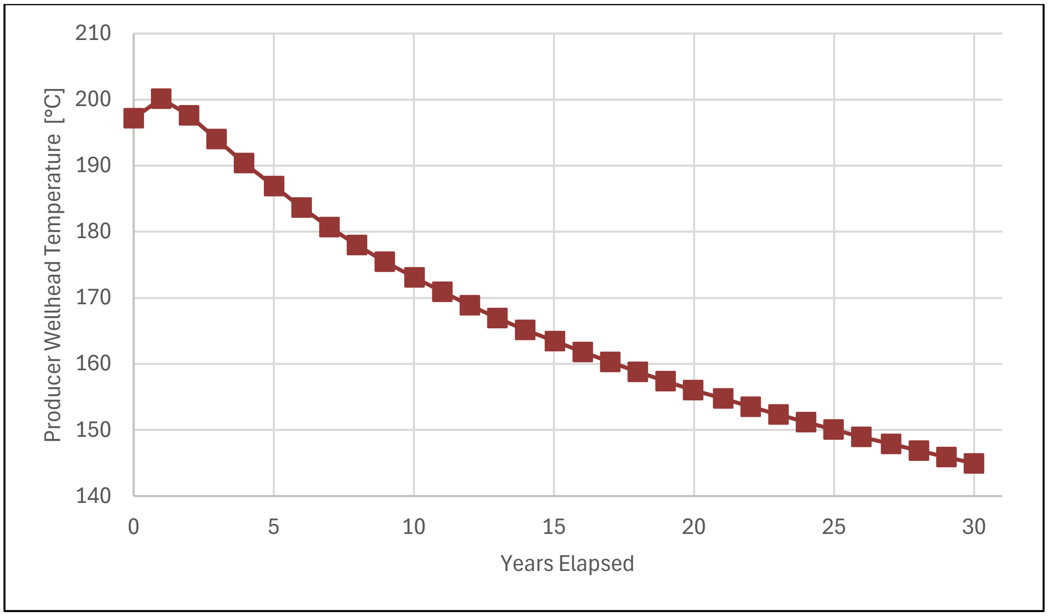

Figure 2. Producer Wellhead Temperature over Time

A closed-loop thermodynamic cycle model was created to explore the geothermal resource. The cycle was modeled as a simple cycle utilizing 4 basic components due to the low-temperature heat source, as shown in Figure 3. The simple cycle configuration consists of four main components: a compressor or pump to increase the pressure of the fluid by adding work to the fluid, a heat exchanger to interface with the geothermal source and adds heat to the fluid, a turbine to expand the fluid and extract work from the fluid, and a second heat exchanger to reject the remaining heat from the fluid.

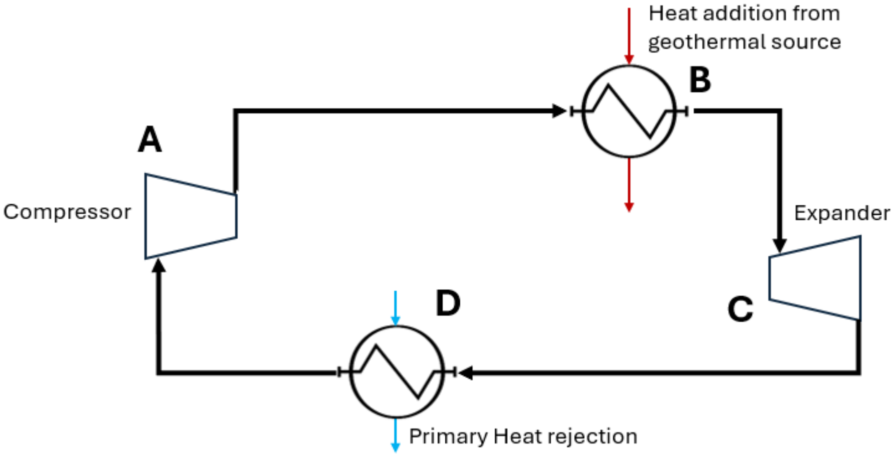

Figure 3. Thermodynamic Cycle Diagram for the Simple Cycle Configuration

2.1 sCO$_2$ Cycle Modeling (Design)

Initial property calculations and design studies in worksheets and tables were performed using the REFPROP database from the National Institute of Standards and Technology [1]. The sCO$_2$ cycle sweeps and design performance estimates were performed using Numerical Propulsion System Simulation (NPSS) [2]. Machinery and heat exchanger assumptions for the sCO$_2$ cycle are shown below in Table 1. The CO$_2$ mass flow rate and turbine inlet pressure were swept over a range of values to determine the design point that provided the maximum net power output, assuming the equipment was designed for various years throughout the lifespan of the geothermal resource.

Table 1. Key Cycle Assumptions for the sCO₂ Cycle

Parameter	Value
Primary Working Fluid	CO_2
Geothermal Fluid	Water-Brine mixture
Ambient Air Temperature	3°C / 35°C
Compressor Suction Pressure	80 bar_a
Turbine Isentropic Efficiency	92%
Compressor Isentropic Efficiency	84%
Generator and Motor Efficiency	95%
Gearbox Efficiency	98%
EGS Fluid Outlet Temperature	50°C
Heat Exchanger Approach Temperature	10°C
Cooler Coefficient of Performance (COP)	100

Preliminary cycle results are shown in Figure 4. The results show that if the cycle was optimized for the resource conditions at Year 15, for example, around 8-9 MW of power could be output from the cycle by designing the equipment for a mass flow rate of 420 kg/s and a turbine inlet pressure of 260 bar. Each parameter in the design sweep is represented by a gradient from blue to red on the other graph. Thus, the highest net power across a range of parameters will push designs toward a higher turbine inlet pressure and a lower mass flow rate.

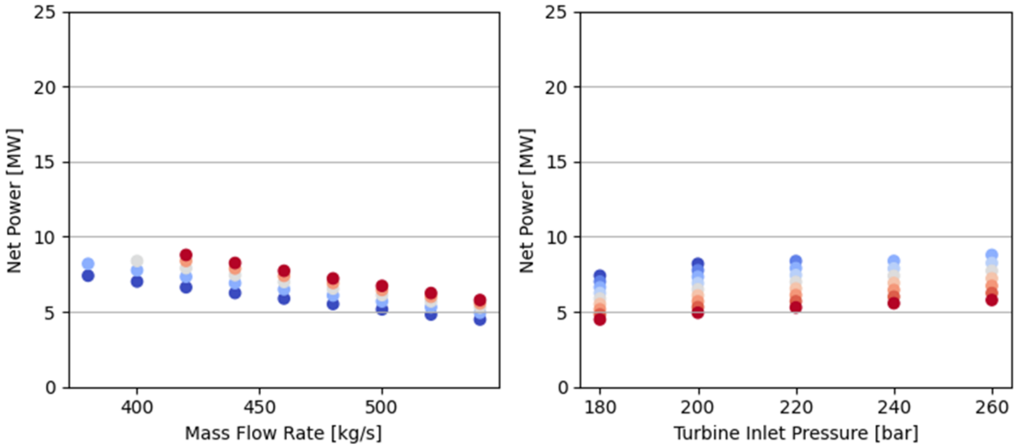

Figure 4. Net Power Output for Well Conditions Representative of Year 15

The cycles can further be analyzed by examining the pressure-enthalpy (Figure 5) and the temperature-entropy diagrams (Figure 6). The PH diagram shows how the power output of the cycle (defined as the area inside the cycle) decreases as the cycle was designed for later years.

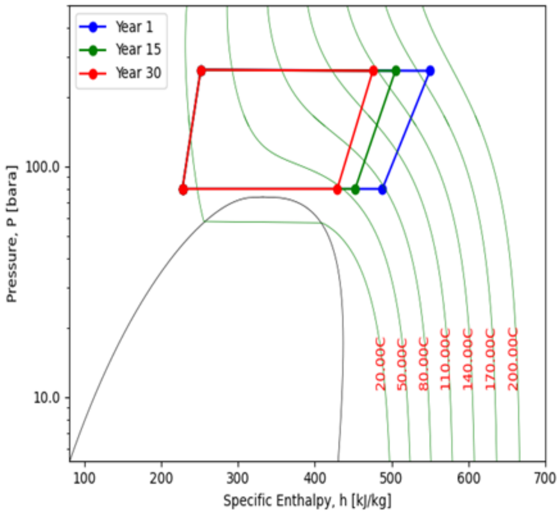

Figure 5. Pressure-Enthalpy Diagram for sCO_2 Simple Cycle Model at Various Years

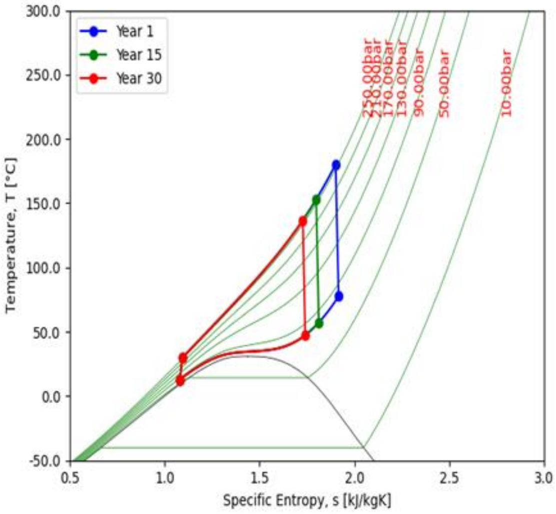

Figure 6. Temperature-Entropy Diagram for sCO_2 Simple Cycle Model at Various Years

Per direction from the client, the team chose year 1 as the design point, maximizing power production when well temperatures are highest. A summary of the major cycle conditions at key points is provided in Table 2. This represents the conditions of the CO_2 power cycle.

Table 2. Year 1 Design Cycle Point Summary

Cycle Point	T (°C)	P (bara)	h (kJ/kg)	s (kJ/kgK)	Phase
Compressor Inlet	13.0	80.0	227.62	1.08	Subcooled liquid
Heater Inlet	29.8	262.6	251.46	1.09	Subcooled liquid
Turbine Inlet	180.2	260.0	550.05	1.90	Supercritical
Cooler Inlet	77.6	80.0	488.04	1.92	Supercritical

For year 1, the design performance is summarized in Table 3. The shaft power is reduced by inefficiency in generators, motors, and gearboxes. Combined with the power loss to run the air cooler, the net power in year 1 is 15.2 MW. The CO_2 mass flow rate for this system is 515 kg/s.

Table 3. Year 1 Design Power Performance Summary

Compressor Shaft Power [kW]	12,274
Turbine Shaft Power [kW]	31,933
Generator Electric Power [kWe]	29,730
Compressor Motor Electric Power [kWe]	13,183
Air Cooler Power [kWe]	1,341
Net Plant Power [kWe]	**15,206**

Figure 7 represents the design space of efficiencies from year 1 to year 30. Note that this is the projected performance if the system maintains its design efficiency. This is not a representative estimate of the system performance because machinery will drop in efficiency as the system moves into an off-design condition. The team will assess the basic off-design performance in the next section of the report.

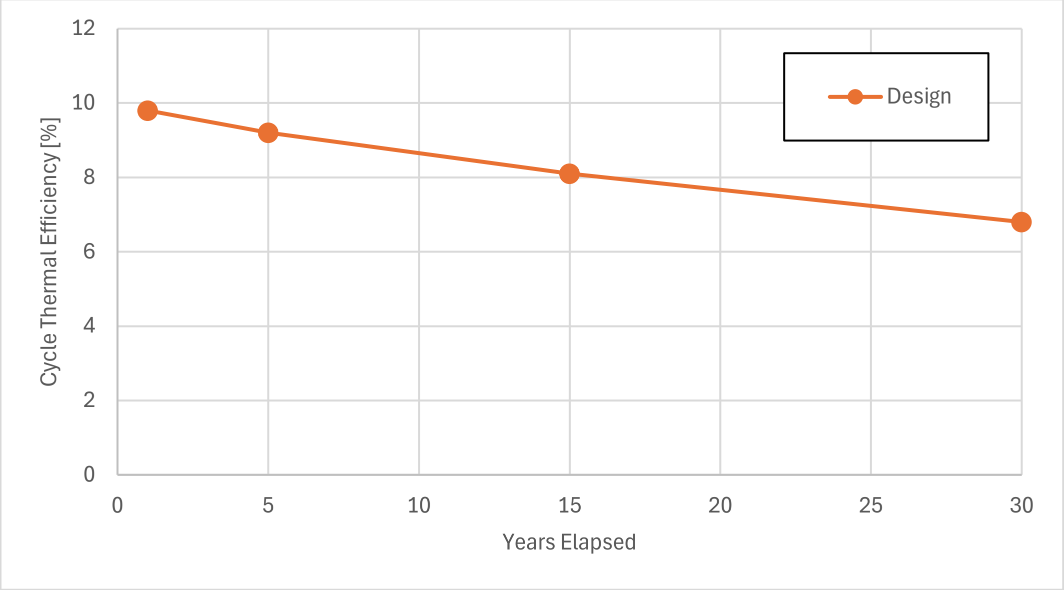

Figure 7. Design Cycle Efficiency over the 30-Year Well Conditions

Figure 8 compares the cumulative energy output of the cycle for the two main ambient temperature configurations. The cycle would lose about 68% of the overall energy output at the higher ambient temperature when compared to the lower ambient temperature. Note that turbomachinery efficiencies were held constant for this analysis.

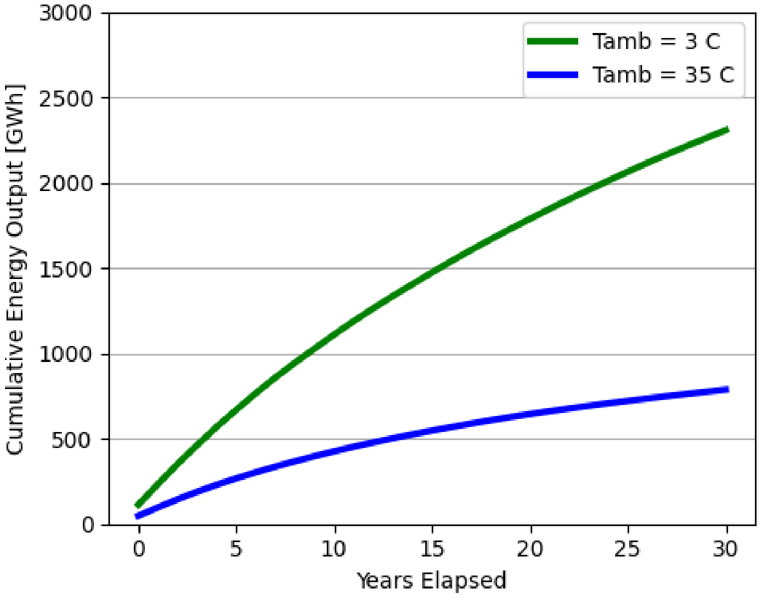

Figure 8. Cumulative Energy Output Over Time for sCO_2 Simple Cycle Model

2.2 sCO_2 Cycle Modeling (Off-Design)

Off-design analysis of geothermal power cycles requires careful consideration of the chosen design point due to the transient nature of the geothermal well. This means the turbomachinery will primarily operate away from its optimal design point for most of its operational lifetime. A design point analysis is where the well conditions and the corresponding heat rate are from the well at its lifetime maximum. In this case, that point is within year 1, shortly after the well begins production. For the off-design cycle analysis, the turbine off-design map provided a new set of inlet conditions that maximized isentropic efficiency, given the new EGS fluid conditions and its corresponding heat input through the primary heat exchanger.

The team used internal resources to develop a turbine map for the system. **Error! Reference source not found.** shows the efficiency versus performance of the turbine over two varying parameters. These parameters, based on experience, are the change in turbine pressure ratio versus the speed divided by the turbine inlet temperature. During the analysis it was noted that since the turbine was single-speed, the inlet temperature and mass flow rate of the cycle were correlated. Thus, the turbine drove the mass flow and pressure required.

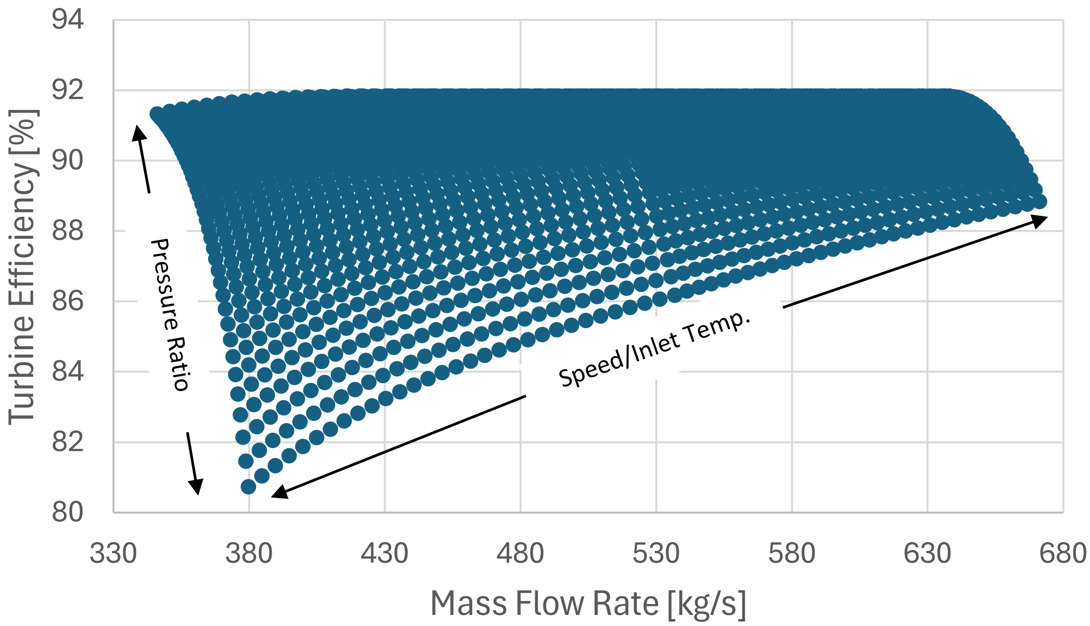

Figure 9. Off-design Turbine Map

The compressor off-design map was generated based on sCO_2 compressor designs documented in the literature [3]. An example of flow coefficient plotted against both isentropic efficiency and isentropic flow coefficient is given below in Figure 10. These maps were chosen because they were demonstrated in a lab environment on sCO_2 and they represent a wide range of operating conditions while maintaining good efficiency relative to the design point.

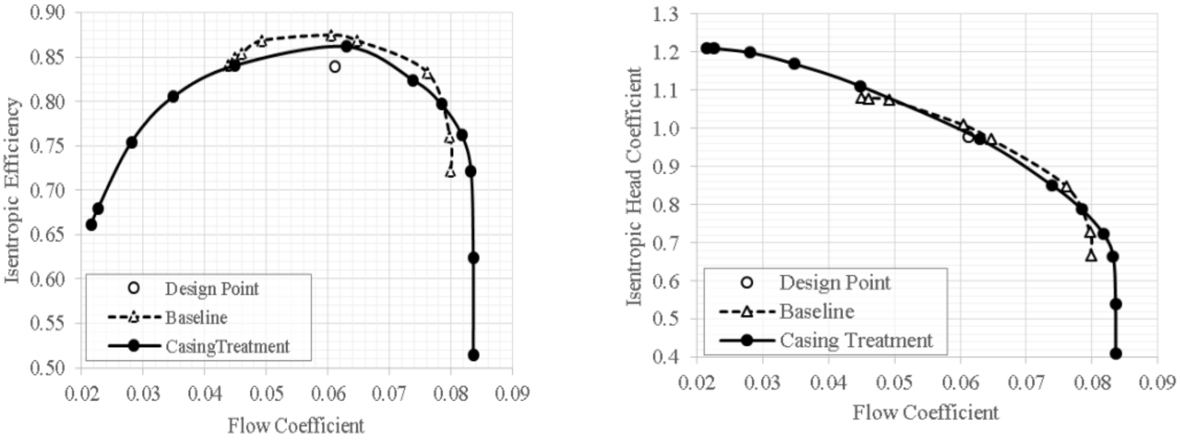

Figure 10. Compressor Performance Plots [3]

For the following analysis, the compressor was assumed to be a single speed. This allows the flow coefficient to be proportional to the mass flow rate through the compressor, and the head coefficient proportional to the isentropic head rise through the compressor. The compressor performance plots were non-dimensionalized (shown in Figure 11 and Figure 12), and the results were applied using the following parameters in Table 4.

Table 4. Compressor Design Point Parameters for Year 1

Parameter	Value
Isentropic Compressor Efficiency	84%
CO_2 flow rate	515 kg/s
Isentropic Head Rise	20.02 kJ/kg

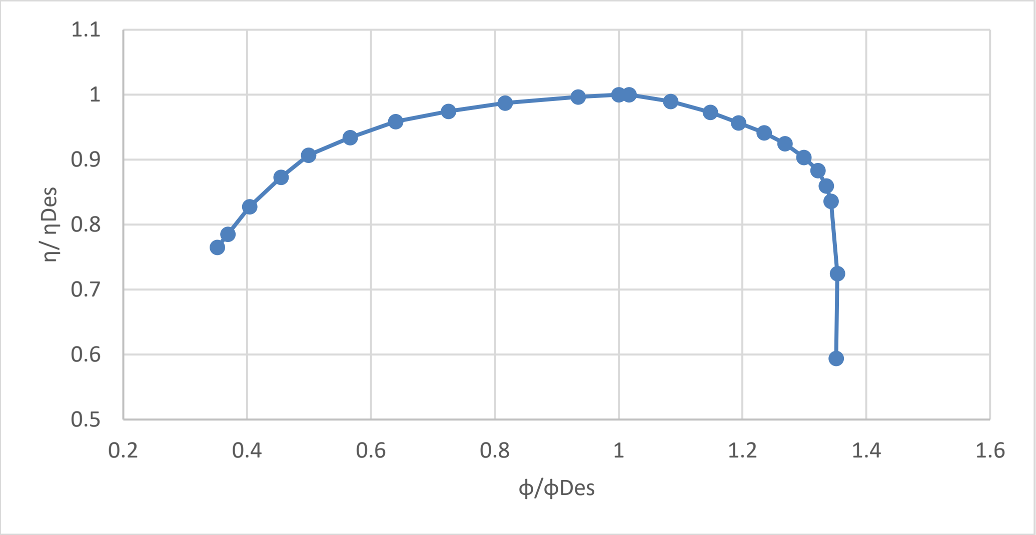

Figure 11. Nondimensionalized Efficiency Map for a Single-Speed Compressor

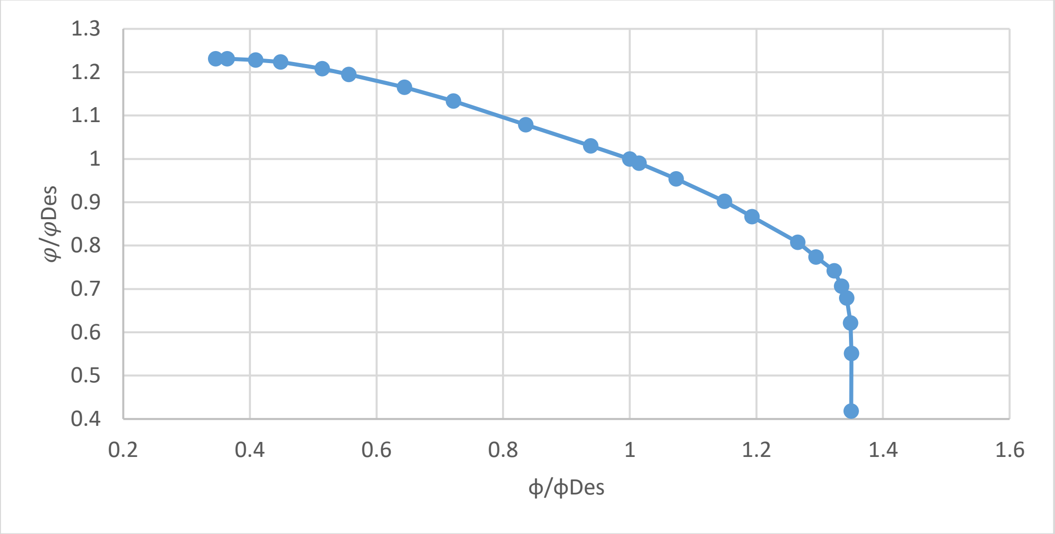

Figure 12. Nondimensionalized Head Rise Map for a Single-Speed Compressor

Results from this off-design analysis are shown in Figure 13 and Figure 14. As the turbine dictated inlet flow and pressure, the compressor provided excess head rise as the CO_2 flow demand decreased over time. As a result, it was necessary to introduce a valve upstream of the primary heat exchanger in order to meet the turbine's inlet pressure demands. This solution is lossy because the higher pressure produced by the compressor is dropped across a valve. Adopting a variable speed system will alleviate this issue and improve system performance. The shape of the compressor off-design map results in operating at points with a much larger than required head rise to meet the flow demand.

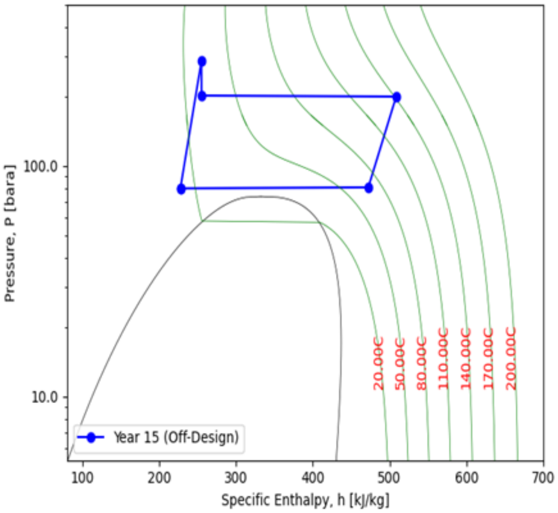

Figure 13. Pressure-Enthalpy Diagram for sCO_2 Simple Cycle Off-Design Model with Single Speed Compression

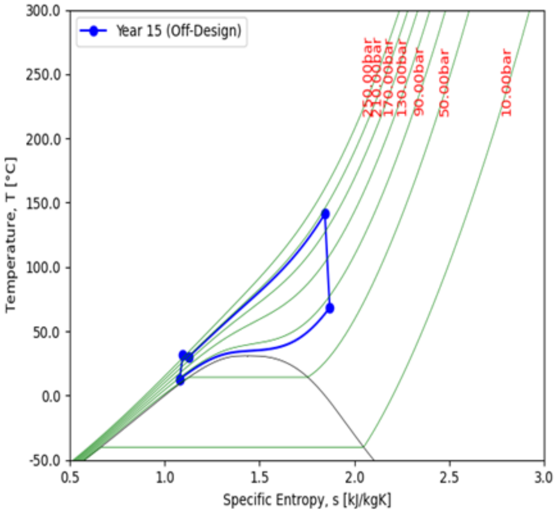

Figure 14. Temperature-Entropy Diagram for sCO_2 Simple Off-Design Model with Single Speed Compression

Figure 15 shows the cycle thermal efficiency over time. The design trendline does not consider efficiency fall off while the off-design trendline incorporates the single-speed design for year 1 and the off-design performance maps outlined above. Given the sharp decline in the single-speed compression efficiency, a variable speed compressor is likely needed to maintain higher turbomachinery efficiencies while still meeting flow and head rise demands. Note that its efficiency does include the air cooler performance loss, but does not include any estimate of the well circulating pump power.

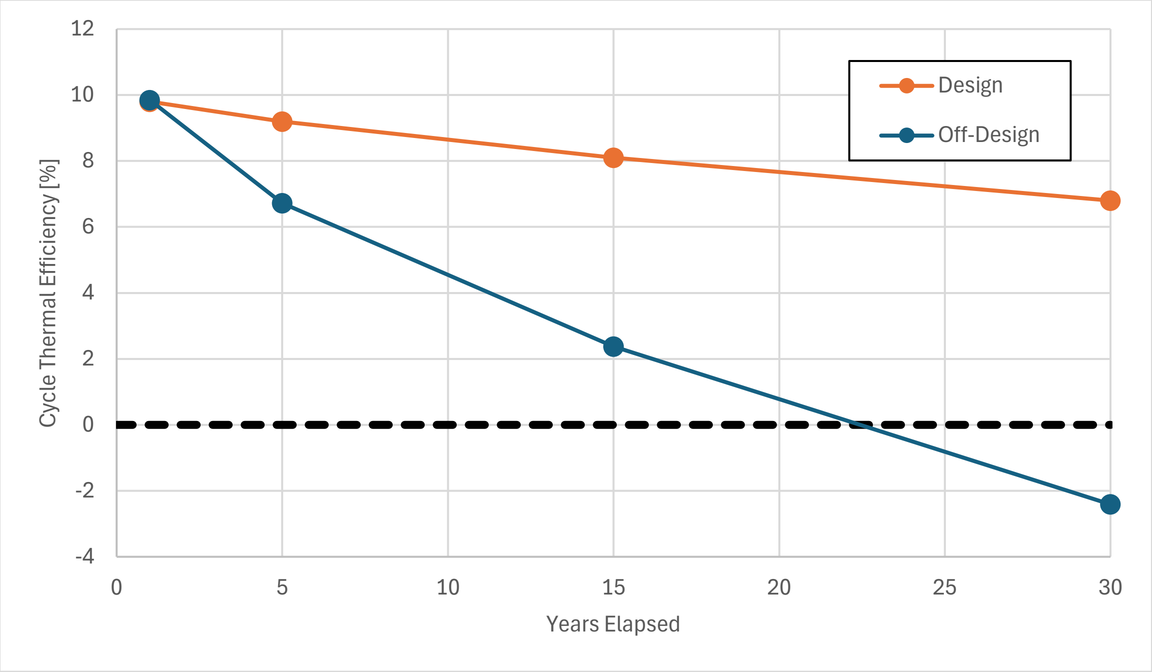

Figure 15. Overall Cycle Efficiency Over Time

3 ESTIMATE OF SCO2 POWER SYSTEM COST

Based on the sizing of the sCO2 power system, an estimate of the cost was developed. The cost correlations are derived from Weiland et al. [4], which gathered cost information for sCO2 equipment and published the associated cost correlations. The cost correlation can be generalized by equation (1).

$$Cost = C_o + KA^n \tag{1}$$

The input correlations for the various cycle components are shown in Table 5. Note that the correlation for recuperators was used for the cost of the primary heater. This is due to the fact that the other cost correlations were for combustion-based heaters that included natural gas burners, for example. Since the geothermal water to high-pressure CO2 primary heater will need to be pressure-rated for the sCO2 pressures, using recuperator costs should be representative for this degree of analysis. The cost correlations are for 2017 U.S. dollars. To bring the costs to

2025 dollars, a 2.85% inflation per year rate is applied, which results in an increase in equipment costs from the literature by about 25.2%.

Table 5. Sizing Values and Cost Correlation Parameters

	sCO_2			
Equipment Sizing	Sizing (A)	Constant (Co)	Constant (K)	Exponent (n)
Cooler, Sizing: UA (W/K)	5,240,500	0	32.88	0.75
Compressor, Sizing: Power (MW)	12.274	0	1,230,000	0.3992
Heater, Sizing: UA (W/K)	7,673,900	0	49.45	0.7544
Turbine, Sizing: Power (MW)	31.933	0	406,200	0.8
Gearbox, Sizing: Power (MW)	31.933	0	177,200	0.2434
Generator, Sizing: Power (MW)	29.730	0	108,900	0.5463
Motor, Sizing: Power (MW)	13.183	0	131,400	0.5611

The summary of costs is provided in **Error! Reference source not found.**. The literature provided equipment costs, which were then built into a total installed cost. A baseline cost estimate for indirect sCO2 cycles [5] was used to estimate the ratio between the equipment cost and the erected cost, which is the equipment plus labor, connective piping, site work, and other items required to erect the system. Once the erected cost is known, an engineering, procurement, and construction (EPC) contractor fee of 20% is added and a 10% contingency. This results in a total installed cost of about $78.8 million at about $5,183/kW.

Table 6. Cost Results for an sCO2 System

2025 USD	sCO_2
Nominal Size (kW)	15,200
Cooler/Condenser	$ 4,509,179
Compressor/Pump	$ 4,190,429
Primary Heater/Evaporator	$ 9,679,628
Turbine	$ 8,123,928
Gearbox	$ 515,502
Generator	$ 869,898
Motor	$ 699,319
Major Equipment Cost (USD)	$ 28,587,883
Equipment Cost (USD/kW)	1,881
Erected Cost/Equipment Cost	2.12
Erected Cost (USD)	$ 60,606,311
20% EPC Fee (USD)	$ 12,121,262
10% Contingency (USD)	$ 6,060,631
Total Installed Cost (USD)	**$ 78,788,205**
Cost/Capacity (USD/kW)	5,183

4 REFERENCES

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