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Solar-Geothermal Hybrid Cycle Analysis for Low Enthalpy Solar and Geothermal Resources

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ABSTRACT

Innovative solar-geothermal hybrid energy conversion systems were developed for low enthalpy solar and geothermal resources that take advantage of the potential synergies of solar thermal and geothermal power cycles. A range of power cycle configurations and working fluids were evaluated in terms of both thermodynamic and economic metrics. Accurate pure component property data and equation of state models were utilized using the Aspen Plus platform for mass and energy balance calculations. A parametric steady-state study was completed to examine the performance over the range of conditions resulting from diurnal and seasonal variations as well as geothermal reservoir depletion. The results of the diurnal and seasonal parametric studies were used to estimate the annualized electricity generation. A dynamic model was created for the selected hybrid system and used to examine the transient performance for a typical January day and a typical July day. The study was focused on determining the hybrid configuration that yielded the highest annualized electricity generation. The levelized cost of electricity (LCE) was estimated using equipment costing rules of thumb developed from Aspen HTFS and Aspen ICARUS software and from other sources.

1. INTRODUCTION

The objective of this study is to develop innovative solar-geothermal hybrid energy conversion systems for low enthalpy geothermal resources augmented with solar energy. The goal is to find hybrid solutions that take advantage of the potential synergies of solar thermal and geothermal power cycles. These synergies may be related to the differing diurnal cycles of geothermal and solar energy. They may also be related to using the low enthalpy (or exergy) geothermal energy to enhance the power generation of high enthalpy (or exergy) solar energy, or vice versa.

This study determined the hybrid configuration that yielded the highest annualized electricity generation. The levelized cost of electricity (LCE) was estimated using equipment costing rules of thumb developed from Aspen HTFS and Aspen ICARUS software and from other sources, such as the NREL Solar Advisor Model (SAM).

Several power cycle configurations and working fluids were evaluated in terms of both thermodynamic and economic metrics. Accurate pure component property data and equation of state models were utilized using the Aspen Plus platform for mass and energy balance calculations. A detailed model for the solar-geothermal system was developed. Turbine flexibility relative to vapor flow rate, temperature, pressure variations was analyzed. A parametric steady-state study was carried out to examine the performance over the range of conditions resulting from diurnal and seasonal variations. The results of the diurnal and seasonal parametric studies were grossly weighted to approximate a typical year in Stillwater, Nevada, and these results led to an estimate of the annualized electricity generation.

A dynamic model was then created for the selected hybrid system, and was used to examine the transient performance for a typical January day and a typical July day. The dynamic model approximated the thermal inertial of the heat exchangers and the working fluids in the exchangers, solar collectors, piping and in the storage tanks. This model was driven with a solar forcing function and an ambient temperature forcing function to approximate the typical winter and summer days. The dynamic model will not represent the moment of inertia of the turbine, since this time scale is not relevant to this study.

2. STEADY-STATE ANALYSIS OF HYBRID SYSTEMS

The following sections describe the design and steady-state analysis of solar-geothermal hybrid energy conversion systems. First, the design basis will be explained. Then several conceptual designs will be introduced, including metrics used to assess their performance. Next, the Aspen Plus computer models will be introduced with their respective assumptions. These models will then be parametrically analyzed to determine annualized electricity production and to estimate the annualized electricity generation.

2.1 Design Basis for hybrid cycle

The design is based on a R134a Supercritical Binary Cycle. This cycle was chosen because of its superior performance in prior internal studies when compared to other binary working fluids and subcritical cycles in low enthalpy geothermal resources. The cycle was constrained by the following parameters:

- Geothermal fluid mass flow: 100 kg/s
- Geothermal fluid temperature: 150°C
- Dead-state temperature (Air ambient temperature): 20 °C
- Turbine isentropic efficiency:
  - 85% for fully-vapor expansions
  - <85% when liquid is present (calculated from the Baumann rule)
- Turbine exit vapor quality ≥ 90%
- Mechanical/generator efficiency: 98%
- Pump efficiency: 80%
- Condenser subcooling: 2 °C

- Main heat exchanger log-mean-temperature-difference (LMTD): 5 °C
- No pressure drop in heat exchangers.

In addition, several rules defining optimum design parameters were built into the model. These were determined by running various cases and developing rules of thumb in the earlier study. They are:

- The air flow through the air cooled exchanger was set such that the air temperature rise is exactly half of the temperature difference between the working fluid condensing temperature and the ambient air temperature.
- The parasitic power to run the fans in the air-cooled condenser is very nearly constant and equal to 0.25 kW per kg/s of air flow through the ACC.
- The optimum configuration for the turbine expansion path was selected using the overall utilization efficiency as the criterion. The path was examined to determine that the working fluid did not pass through the critical point, did not cross the saturated liquid line, and that the exit vapor quality was at least 90% to minimize erosion on blades/vanes

This project departs from the earlier study with the addition of solar energy to the cycle. For the hybrid models, the following solar constraints were chosen for the initial analysis:

- The solar energy will come from parabolic trough collectors.
- Solar heat transfer fluid (HTF) is Therminol-VP1.
- Maximum solar HTF temperature: 390°C.
- Solar thermal heat storage method: Two-tank molten salt storage.
- Collector field size based on requirements for optimum cycle performance.

These constraints are based on estimates of which solar technology can be more useful for geothermal systems and proven technology in the solar thermal field. However, these constraints will be evaluated during the course of this study to verify that they are the best choice. For example, several solar heat transfer fluids are commercially available, with Therminol-VP1 currently the most popular choice.

A critical constraint on our system is the upper temperature limit of R134a. At temperatures above 200°C, R134a begins to decompose. Thus, care must be taken to prevent the R134a temperature from getting too high. In these models, R134a was never allowed to go above a bulk temperature of 180°C to ensure a safe margin. This temperature ceiling ultimately limits the amount of solar exergy that can be transferred to R134a at a given flow rate.

Part of the design basis examined the temperature of the brine returning to the reservoir. If it was below 70°C, the brine would be at risk of depositing minerals in the pipes or the formation. Thus, care was taken to ensure return brine temperature would exceed 70°C.

It is important to recognize that the steady-state model will be used for a dynamic simulation, and this will require the design basis to be refined. For example, pressure drops in all heat exchangers will need to be added and solar collectors will need to be modeled as heated pipes. However, the optimum equipment sizing will be based on steady-state results.

2.2 Conceptual Designs and Metrics for Comparing Hybrid Options

This section presents and discusses two hybrid solar-geothermal systems that were selected to be studied further after many preliminary designs were sketched and looked at. These were the solar-geothermal supercritical superheat binary hybrid (referred to as the "superheat hybrid"), and the solar-geothermal steam-flash/supercritical binary hybrid (referred to as the "flash hybrid"). A third system, the solar-geothermal supercritical preheat binary hybrid, or "preheat hybrid", was also selected. However, this system was dismissed after preliminary analysis because it had much less ability to use solar exergy compared to the flash hybrid.

The superheat hybrid concept is a simple modification to the basic R134a supercritical cycle, while the flash hybrid concept offers more possibilities with the addition of a steam cycle but with more complexity. Although the term "superheat" normally refers to raising the temperature of a saturated vapor into the superheated region, we will adopt this term for the present supercritical case. Both concepts use the same solar resource and follow the same design constraints set forth in the previous section.

2.2.1 Superheat Hybrid: Conceptual Design

An obvious means of integrating solar energy into a supercritical binary plant is to use a solar collector to raise the temperature of the R134a before it enters the turbines, thereby increasing the working fluid exergy and making it possible to increase the power from the turbine. Figure 2.1 shows the process flow diagram of this cycle. The brine heats up the working fluid (WF) via the low-temperature heat exchanger (LTH), and the WF picks up extra heat in the high-temperature solar heat exchanger (HTH).

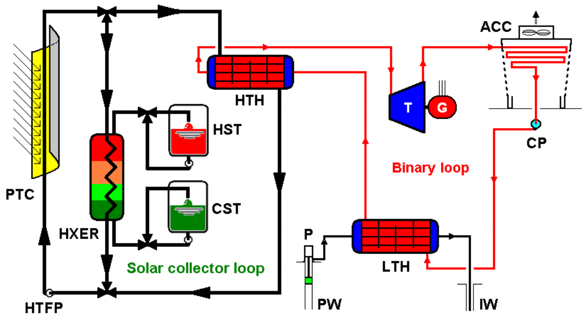

Figure 2.1: Solar-geothermal superheat hybrid concept.

In this concept, the R134a quickly approaches its upper temperature limit unless its flow rate is dramatically increased. This improves the gross power coming from the turbine (T), however, it greatly increases the parasitic loads from the WF circulating pump (CP) and the air cooled condenser (ACC) fans. Thus, there is an optimal flow rate for R134a that corresponds to an optimum net power produced. These results will be looked at in more detail in Section 2.4.

This concept requires active monitoring of the solar resource available in order to pump the optimum flow of WF through the loop: too little and the R134a can overheat, too much and the parasitic loads will decrease performance considerably. Thus, this concept is not easy to manage and will require a sophisticated control system.

3.2.2 Flash Hybrid: Conceptual Design

The flash hybrid plant is shown below in Figure 4.2. It is intended for low-to-moderate geothermal resources with pumped wells. The solar energy is used to raise the temperature of the pressurized geofluid to a sufficiently high value (roughly saturation) to allow flashing at an appropriate pressure. This generates steam and hot brine for use in a steam turbine and for heating the WF in the binary loop, respectively. The flash pressure is an adjustable, optimizable quantity, constrained by the temperature limit on the WF described above.

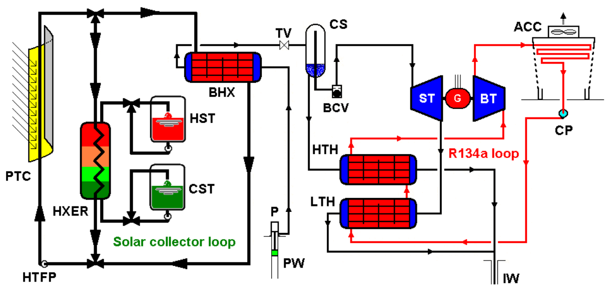

Figure 2.2: Solar-geothermal flash concept.

The geosteam drives one of the two turbines connected to a common generator, in much the same way as in the Ormat "combined cycle" plants. After leaving the back-pressure steam turbine (ST), the exhaust steam is condensed against the R134a working fluid in the bottoming binary, supercritical cycle in the low-temperature heat exchangers (LTH). The hot separated brine from the separator is used to impart the final heating to the working fluid in the high-temperature heat exchanger (HTH).

The exhaust pressure from ST is another adjustable, optimizable parameter, as is the R134a binary turbine (BT) inlet pressure.

When the solar energy is no longer available (i.e., when the sun sets, is obscured by clouds, and the thermal storage is depleted), the system will continue to operate as a pure geothermal plant, albeit at a lower power generation rate. The brine would be directed from the well pumps to the two R134a heaters via a bypass line, and the steam turbine would be disconnected from the generator by means of a clutch. Bypass lines, bypass valves and the clutch are omitted from Figure 2.2 for the sake of clarity.

Figure 2.3 shows the process diagram in temperature-entropy coordinates. The diagram is to scale for water substance and the R134a saturation curve is overlain roughly to scale; the s-axis has been stretched for the R134a to render it approximately within the range of water entropy values. The isobars pertain to water; the critical pressure for R134a is about 40.6 bar and its critical temperature is 101.2oC. The heat needed to raise the R134a from D to A is supplied in two steps: in the low-temperature heater (LTH) from D-E with heat coming from the condensing steam (6-7), and in the high-temperature heater (HTH) from E-A with heat coming from the separated, hot brine (5-8). States 7 and 8 are close in temperature but 8 is at a higher pressure. Since the condensing exhaust steam provides the lower-temperature heat, it could be at a somewhat lower temperature than shown in the figure. Furthermore, since R134a exhibits a normal condensation line, state A could be raised to a higher temperature than shown and prevent any moisture in the last stages of the turbine. This temperature is limited by the flash temperature, T3.

Night Operation: When the solar energy is not available, the only heat source is the brine at temperature T1. It will likely be necessary to lower the R134a mass flow rate in order to raise its temperature sufficient to avoid the critical point. Dropping the R134a pressure to subcritical at night would also solve this problem but might cause trouble with the turbine inlet conditions. With Aspen Plus we can investigate different ways to operate the plant at the night situation.

2.2.3 Metrics for Comparing Hybrid Designs

The selection of one hybrid design over another involves an assessment of the thermodynamic performance advantage coupled with the cost of the new equipment to accomplish the design. A practical aspect in the selection process concerns the need to keep the geothermal plant in continuous operation, both under hybrid and stand-alone conditions. It is impractical to shut in the wells serving the plant or to shut down a unit when the sun is not shining.

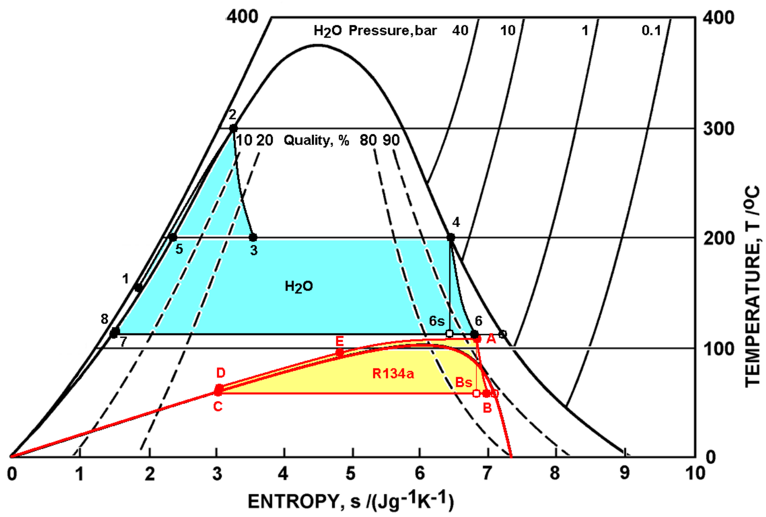

Figure 2.3: Process diagram in temperature-entropy coordinates.

The best criterion for comparing the thermodynamic performance is the utilization efficiency, η_u . This factor is based on the Second Law of thermodynamics and can be used for any power generating system, whether it operates cyclically or as a sequence of processes. It uses the exergy (or available work) as the basis for the performance assessment.

In its general form as applied to a simple system operating under steady, open conditions, the rate of exergy of a fluid at a given state 1 is given by:

$$\dot{E}_1 = \dot{m}[h_1 - h_0 - T_0(s_1 - s_0)] \tag{1}$$

where $\dot{m}$ is the mass flow rate, h is the enthalpy, s is the entropy, and T0 is the dead-state temperature. This equation may be applied directly in this form to calculate the exergy of the brine as it enters the power plant.

The calculation of the exergy of the solar energy input is not as direct. Ideally we would like to calculate the exergy associated with the electromagnetic solar radiation, but this is a controversial subject on which there is no general agreement. We can, however, base our calculations on the exergy of the heat transfer fluid (HTF) that circulates through the solar collectors, using the value of exergy that the HTF possesses when it leaves the solar field and enters the solar-geo heat exchanger, shown as BHX in Fig. 2.1 and as HTH in Fig. 2.2. This can be found from the following equation:

$$\dot{E}_{HTF} = \dot{m}_{HTF}[h_{1,HTF} - h_{0,HTF} - T_0(s_{1,HTF} - s_{0,HTF})] \tag{2}$$

We may also consider the change in the exergy of the HTF as it passes through the heat exchanger. This is the rate of exergy that the HTF releases when in thermal contact with either the R134a (superheat hybrid concept, Fig. 2.1) or the brine (flash hybrid concept, Fig. 2.2), and is given by:

$$\Delta\dot{E}_{HTF} = \dot{m}_{HTF}[h_{1,HTF} - h_{2,HTF} - T_0(s_{1,HTF} - s_{2,HTF})] \tag{3}$$

Since exergy is not a conserved quantity, a fraction of the exergy released by the HTF will be destroyed by various irreversibilities within the heat exchanger, and thus not be transferred to the brine or the R134a.

There are several possible definitions of utilization efficiency that are appropriate for our task. The first one is for a basic geothermal power plant, namely:

$$\eta_{u,GEO} = \frac{\dot{W}_{net}}{\dot{E}_{GEO}} \tag{4}$$

For a hybrid plant in which there are two distinct forms of exergy input, the definition that usually works best is:

$$\eta_{u,HYB} = \frac{\dot{W}_{net,HYB}}{\dot{E}_{GEO} + \dot{E}_{HTF}} \tag{5}$$

However, since not all solar exergy absorbed by the collectors is actually transferred to the power system (some will be stored in the thermal storage system and other exergy will be recirculated within the solar loop), an alternative utilization efficiency that better measures the performance of the plant is based on the amount of exergy transferred from the solar HTF to the power system. This utilization efficiency measures the ratio of net work from the hybrid plant to the exergy of the brine plus the change in exergy of the HTF, namely:

$$\eta_{u,HYB} = \frac{\dot{W}_{net,HYB}}{\dot{E}_{GEO} + \Delta\dot{E}_{HTF}} \tag{6}$$

The last of the efficiency equations is the familiar thermal efficiency:

$$\eta_{th,HYB} = \frac{\dot{W}_{net,HYB}}{\dot{Q}_{WF}} \tag{7}$$

In this case the denominator is the heat delivered to the cycle working fluid.

These measures will be combined with the cost of the additional systems to arrive at a thermodynamic-economic optimum design. Thus, the capital expense (CapEx) needed to generate a kilowatt (kW) of power, the operating and maintenance cost (O&M) per kWh of electricity, and the levelized cost of electricity per kWh (LCOE) can be used as criteria that combine thermodynamic performance and economics.

2.3 Flowsheet Modeling Assumptions

The Aspen Plus model flowsheets were built starting with a supercritical R134a binary model. For the flash hybrid model, a flash vessel, steam cycle, and solar cycle were added, as illustrated in Figure 2.4. The parameters for these new blocks and streams were determined by several sensitivity studies conducted on different critical parameters. The parameters in the binary cycle were taken from an earlier internal study, in which they were optimized to generate the highest utilization efficiency, with the exception of the WF flow rate, which was increased to accommodate the added energy input from the solar heat source. Once the parameters in the flash hybrid model were chosen, the same parameters were used for the superheat hybrid model.

The solar HTF was heated to 390^{o}C to model a typical parabolic trough solar collector, however this was enough heat to boil the brine. Thus, the brine pressure was increased to 60 bar to allow the brine to be heated to saturation (or 275^{o}C) in stream 2. From there, the brine is throttled down to 15.5 bar to obtain a vapor/liquid mix at the target temperature of 200^{o}C in stream 3. The 200^{o}C target temperature was chosen as the upper temperature limit for the brine to avoid overheating the R134a moving though HX2 (the high-temperature heat exchanger). In addition, the greater the temperature of the brine, the more heat passed to the binary WF, and the more power generated in the BT.

A sensitivity study was conducted to test how varying incoming brine pressure would affect overall plant power. As brine pressure increased, more heat could be passed to it from the solar-brine heat exchanger and thus more steam could be generated to power the steam turbine. Unfortunately, this increase in pressure creates greater parasitics for extraction pumping and would require a very thick-walled heat exchanger with the solar HTF. Therefore, 60 bar was chosen as a compromise between increasing power produced and increasing costs incurred as the incoming brine pressure is increased.

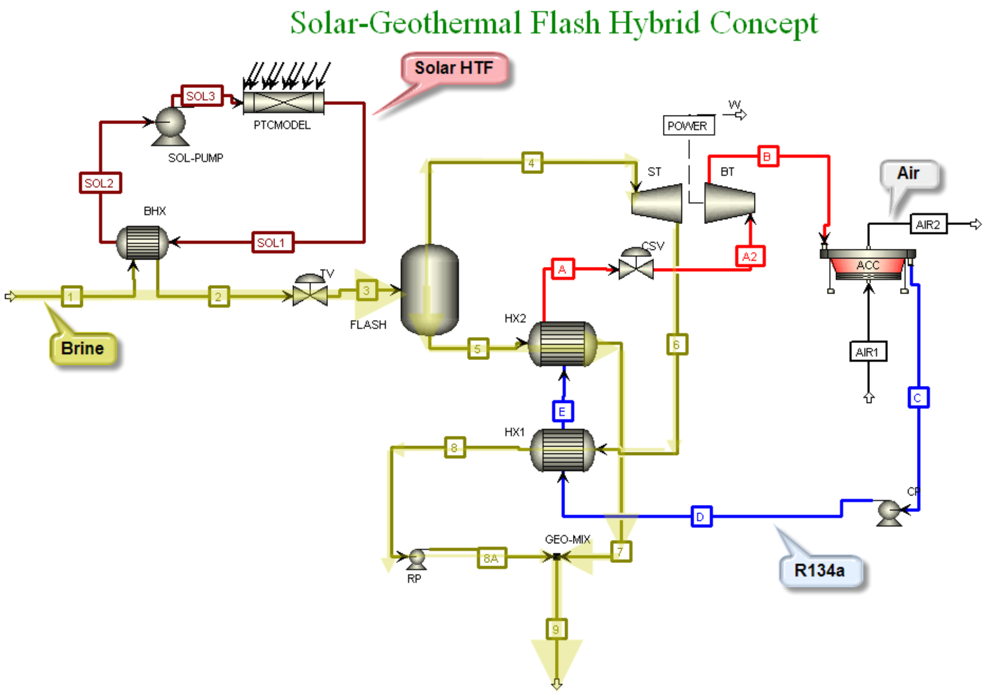

Figure 2.4: Solar-geothermal flash hybrid Aspen Flowsheet.

Other smaller sensitivity studies were conducted to verify the optimal R134a loop pressure and the best turbine outlet conditions. These showed the original values from earlier studies to be optimal and verified our assumptions to use them. However, the previous heat exchanger designs could not be used because of the new configuration. These heat exchangers they were designed to exchange as much heat as possible as long as R134a temperature was never allowed to go above 180°C and the pinch LMTD never went below 5°C.

Some of the model components are not physically needed, but help the model converge properly. One example of this is the control valve between the last heat exchanger and the binary turbine. There is an inherent ambiguity as to whether a supercritical fluid is a liquid or a vapor. The CSV enables the simulator to deal with this ambiguity and ensure that the fluid entering the turbine is identified as a vapor.

2.4 Hybrid Model Preliminary Results

This section discusses the Aspen model preliminary results for the candidate solar-geothermal hybrid systems. Each model was evaluated under different binary WF flow rates. The results show that the flash hybrid performed best in net power, thermal efficiency, and the alternative utilization efficiency, and slightly underperformed in the standard utilization efficiency.

2.4.1 Superheat Hybrid: Preliminary Results

The results in Figure 2.5 show that the model has a very flat net power response to a change in the WF flow rate. As the flow rate is increased, the gross power, the parasitic pumping, and the fan power also increase. Thus, this model is generating close to its maximum net power at 11.4 MW.

The range of flow rates were chosen based on temperature constraints in the binary loop. At low flow rates, the R134a gets very hot after gaining heat from the brine-solar heat exchanger and crosses over to unacceptable temperatures. At high flow rates, the R134a WF takes all the heat from the brine, which drops the brine to unacceptably low temperatures that present risks of mineral scaling.

Table 2.1 shows how this model performed in power generation and efficiency. The thermal efficiency, at 11.4%, was highest at low flow rates. The two utilization efficiencies measured 13.1% and 29.1% at the highest flow rates. This big difference in utilization efficiencies shows that the solar exergy passed to the WF is being used well, however a lot of solar exergy is also not passed to the WF. Note that the exergy remaining in the solar HTF is not lost; it is recycled to the solar collection field and for reheating to 390°C.

Table 2.1: Solar-geothermal superheat hybrid model results.

R134a Flow Rate	Binary Turbine Power	Parasitic Power	Net Power	Thermal Eff.	Util. Eff.	Alt. Util. Eff.
kg/s	MW	MW	MW	%	%	%
350.00	17.07	6.05	11.03	11.4	12.7	28.2
365.00	17.33	6.22	11.11	11.3	12.8	28.4
380.00	17.58	6.40	11.18	11.1	12.9	28.6
395.00	17.82	6.58	11.25	11.0	13.0	28.8
410.00	18.06	6.75	11.31	10.9	13.0	29.0
425.00	18.29	6.93	11.36	10.8	13.1	29.1

2.4.2 Flash Hybrid: Preliminary Results

The flash hybrid model was set with identical design variables as the superheat model and the R134a flow rate was varied to produce the results in Figure 4.6. This figure shows a net power leveling off at near 12.1 MW at flow rates lower than those used for the superheat hybrid. The reason the model flow rate stops at 265 kg/s is that after this point, the brine return temperature is too low.

At 12.07 MW net power, the flash hybrid outperforms the superheat hybrid by 6.25%. The thermal efficiency, as can be seen in Table 2.2, is up to 18%, which is over 50% better than that of the superheat cycle. The higher thermal efficiency gives this model a distinct advantage.

Table 2.2: Solar-geothermal flash hybrid model results (Steam Turbine Power = 4.39 MW).

R134a Flow Rate	Binary Turbine Power	Parasitic Power	Net Power	Thermal Eff.	Util. Eff.	Alt. Util. Eff.
kg/s	MW	MW	MW	%	%	%
240.00	11.32	4.08	11.63	18.0	12.2	29.0
245.00	11.55	4.16	11.79	17.9	12.4	29.4
250.00	11.71	4.23	11.87	17.8	12.5	29.6
255.00	11.85	4.30	11.94	17.7	12.5	29.8
260.00	11.98	4.37	12.00	17.5	12.6	29.9
265.00	12.12	4.44	12.07	17.4	12.7	30.1

The results presented here show that the flash hybrid model performed best in power generation and thermodynamic measures. Therefore, the flash hybrid model will be studied further in the next step in this study.

2.5 Parametric Analysis

In this section we would apply the power output for each viable hybrid system over a full typical year to determine the total energy (electricity) that can be generated on an annual basis. The results for this section are not available as of May 31st 2009, but to demonstrate expected results, here is the parametric analysis for three previously considered hybrid systems.

Historic solar and temperature data for Fallon, NV, were extracted from the National Solar Radiation Database. Figure 2.7 shows a typical June 21 day at Fallon: the insolation in W/m^2 and the air temperature in ^{o}F are plotted for each hour of the day.

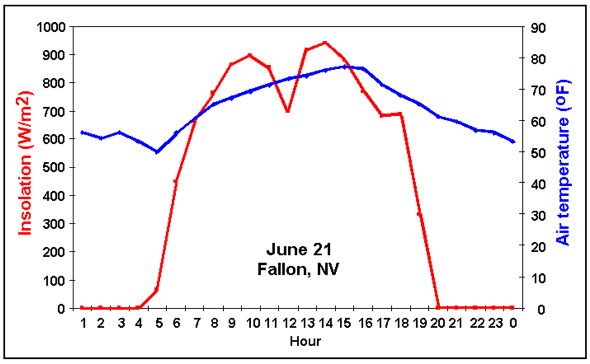

Figure 2.7: Hourly insolation and air temperature for a typical June 21 day at Fallon, NV.

Typical months are pieced together in the NSRD to form a typical year. For each hour of the year, the net power was calculated as generated under two scenarios:

Geothermal power only: The geothermal binary plant operated alone with no assist from the sun.

Hybrid power: The geothermal and solar inputs were combined according to the hybrid configuration under study for whatever amount of incident sunlight was available.

The final step involved summing the entire year's production of energy for each system. The results are presented in a form that shows the output from geothermal only, from the hybrid system, and the incremental annual energy generation that results from the use of the solar collector system.

Figure 2.8 presents an example of a monthly power output for the geothermal-only case as a function of the air temperature. The output is for a single unit, both levels, for the current plant configuration (i.e., no preheaters).

2.5.1 Hybrid System Operating Results

The power output for one of the previous hybrid systems, which used solar energy to preheat the geothermal fluid for a subcritical binary cycle, is shown in Figure 2.9, along with the insolation and air temperature for each day of the typical April month. Figure 2.10 shows a 3-D plot of power over the full range of the parametric study.

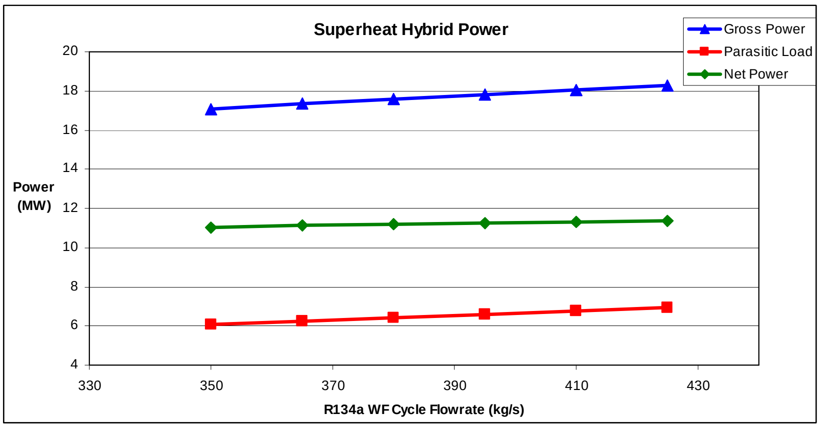

Figure 2.5: Solar-geothermal superheat hybrid model results.

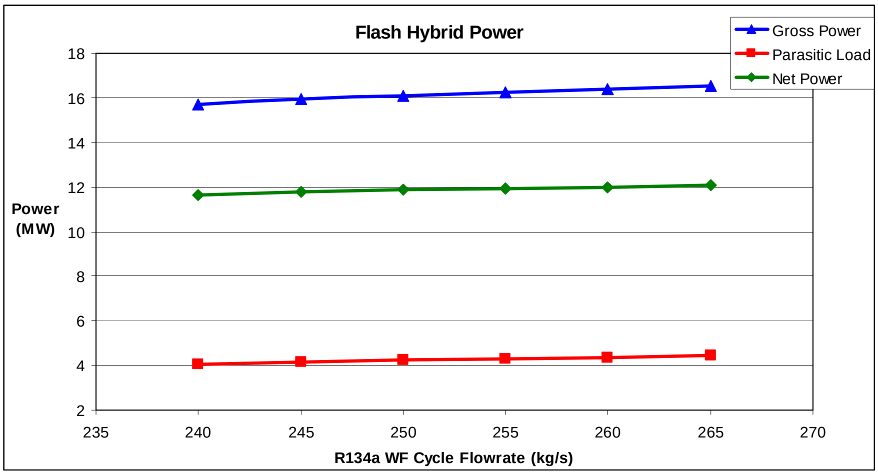

Figure 2.6: Solar-geothermal flash hybrid model results.

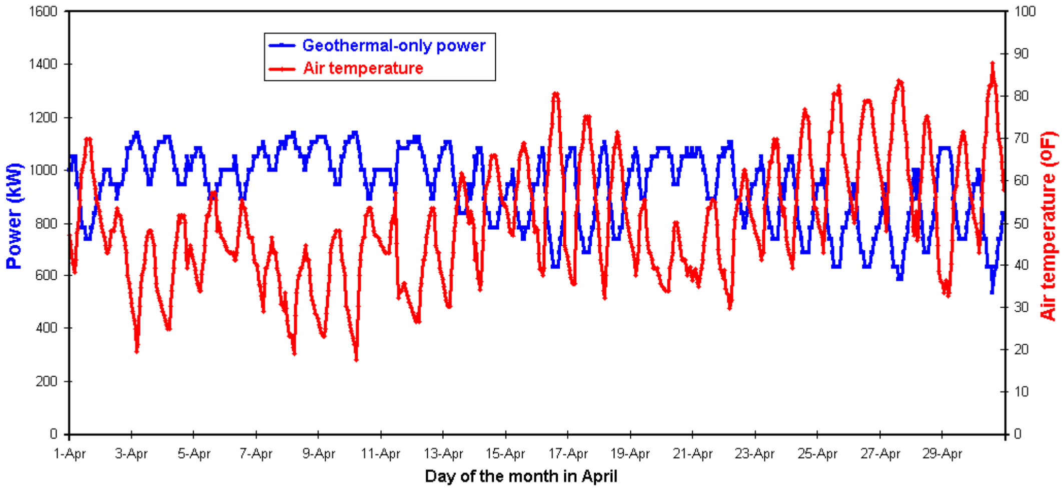

Figure 2.8: Current plant power output as a function of air temperature for a typical month of April in Fallon. NV.

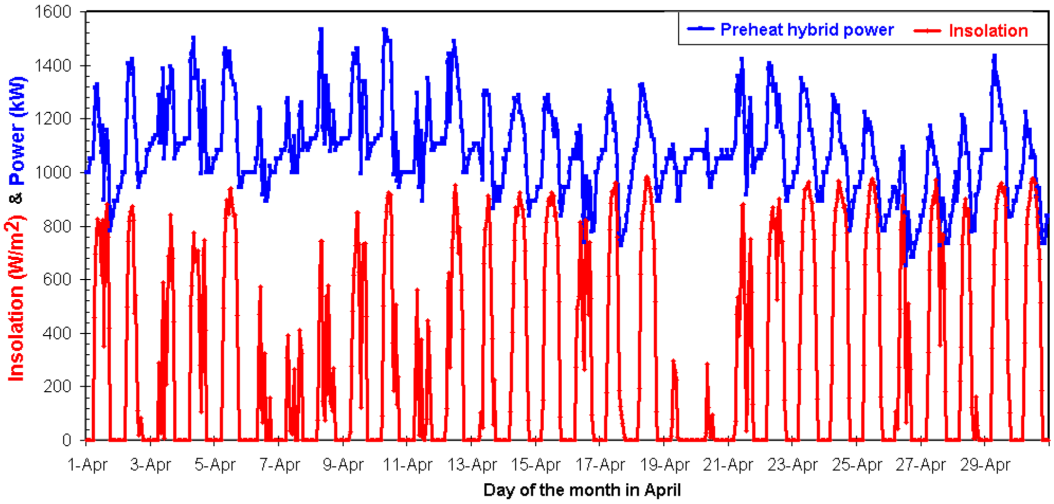

Figure 2.9: Power output for Preheat system over the typical April month.

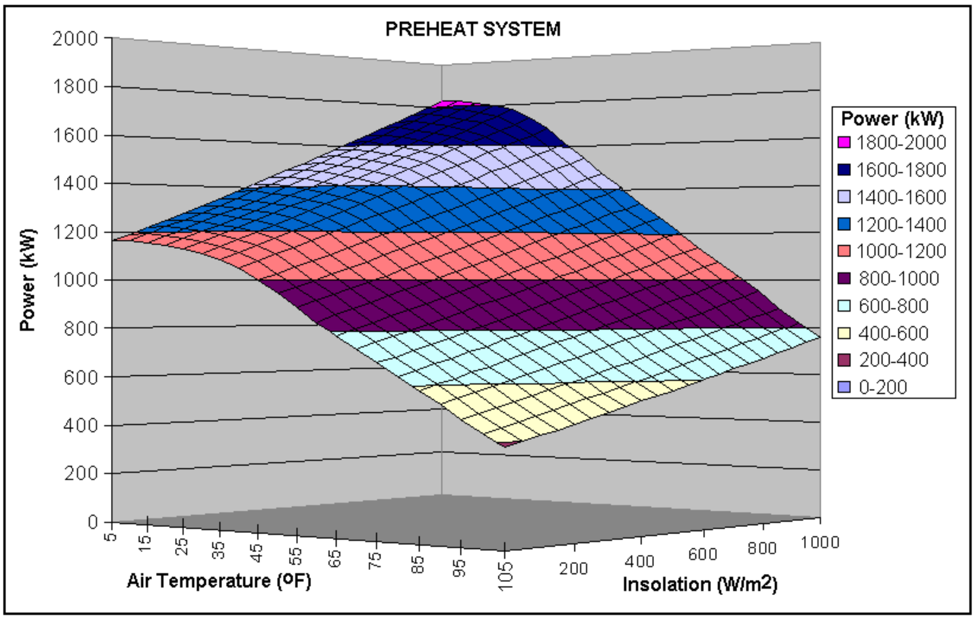

Figure 2.10: Power output for the Preheat system over the range of air temperature and solar insolation values studied.

The monthly electricity generated from this system is given in Figure 2.11. The solar contribution to the total generation varies from about 8% in winter to about 20% in the summer. The total annual generation comes to 8,980.6 MWh and the solar portion amounts to 1,202.5 MWh. In the month of July, the worst for generation of any binary plant, the Preheat system generates 25% more electricity than the base geothermal plant. In July, the Preheat system can produce about 67% of its peak (winter) generation. The base geothermal plant can produce only 60% of its peak generation in July.

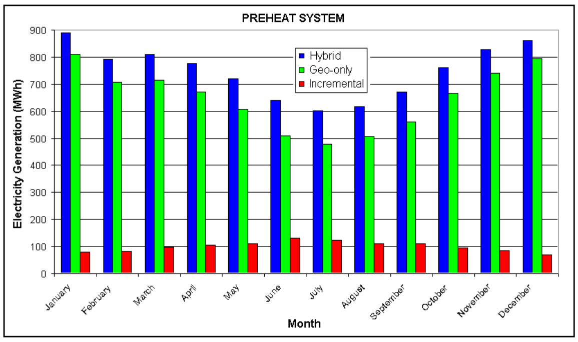

Figure 2.11: Monthly electricity generation for a typical year for the Preheat system.

3. DYNAMIC ANALYSIS

Dynamic analysis of the solar-geothermal flash hybrid model is underway. It will be used to examine the transient performance for a typical January day and a typical July day. A conventional two tank solar storage unit will be added to see how its thermal inertia affects the model's transient performance when the sun goes down or behind a cloud. For the solar collectors, a plug flow heated pipe model will be used to dynamically model the heating along the length of the pipes, including heat loss to the environment. The model utilizes realistic controllers to maintain optimal and safe operation.

4. SOLAR-THERMAL COLLECTOR ANALYSIS

There are a variety of solar collector options depending on the desired working temperature. These are the typical designs for the ranges indicated:

- greater than 400°C - power tower or dish;
- 350-390°C - conventional parabolic trough or linear Fresnel;
- 150-300°C - parabolic trough with water and some linear Fresnel;
- 100-200°C - evacuated-tube flat-plate;
- less than 150°C - flat-plate for home heating and hot water.

There are four main criteria for choosing the solar collectors in the hybrid plant: cost per unit area, efficiency, durability, and operation and maintenance.

Based on market availability, the short-term options are conventional parabolic trough or compact linear Fresnel collectors. Conventional parabolic trough collectors are of the type installed in Nevada Solar One that use a heat-transfer oil such as Therminol-VP1. Table 5.1 lists the major suppliers of conventional parabolic trough collectors and relevant information about performance and company experience.

Another current market option is the compact linear Fresnel reflector (CLFR), currently produced by Ausra. The reported 2008 cost of the system is $3000/kW, but is expected to decrease to $1500/kW in a few years. The heat-transfer fluid is water, which can reach over 285°C at 70 atm. These collectors have been used in the Kimberlina plant, which went online in October 2008; Ausra is working on early-stage projects with PG&E and FP&L. A more experimental option for the future is the CLFR produced by the Solar Power Group which plans to install 10 MWe in Spain in 2009.

Table 4.1: Suppliers of parabolic trough collectors and supporting information.

Suppliers	Performance and Experience
Solucar / Abengoa Solar	Peak η = 0.75
Solargenix	Used in Nevada Solar One
Solar Millennium	Peak η = 0.8 for *Flagsol* collector; 4 years experience
Solel	η = 0.8; test loop demonstrated
Alcoa	Research and Development
Sener / ACS Cobra	Platforma Solar de Almeria
Acciona	Peak η = 0.77, 1 MW in Arizona, 64 MW in Nevada
ENEA (storage included)	Peak η = 0.78; used in Archimedes project (ENEL)

The Solar Advisor Model (SAM), built by the National Renewable Energy Laboratory (NREL), will be utilized to evaluate a variety of technology-specific cost models. For example, one can combine different collector types, receiver types, heat transfer fluids, and storage types to evaluate the most cost effective combinations which satisfy the requirements of the Aspen model. This tool will be used to help select the most promising solar components that are commercially available.

In earlier studies, the following assumptions were used:

- The collectors are conventional parabolic trough.
- Collector efficiency: 70%. Note that the general equation for collector efficiency is given by: $\eta = \beta(\tau,\alpha,P) - h(T_c - T_a)/IC = 0.55$, where $\beta(\tau,\alpha,P)$ is a property of glass cover transmission, reflection, and surface absorption; h is the heat transfer coefficient; C is concentration ratio; I is the irradiance; T_c is the collector temperature; and T_a is the ambient temperature. The 70% value was agreed to in the absence of reliable values for the variables in the efficiency equation.
- The cost per unit area of collector: $250/m^2 for the base case economic analysis.
- $dQ/dt = IA\eta$, where dQ/dt is the heat added from solar and A is the area.

5. THERMO-ECONOMIC ASSESSMENT

When the results of the parametric study are generated, this section will closely follow the thermo-economic assessment of the three previously studied hybrid systems.

The two graphs below, Figures 5.1 and 5.2, give the levelized cost of the incremental electricity (LCOE) gained through the use of the solar energy in the hybrid system. It is the sum of the capital cost for the new solar system (but includes only the solar collector cost represented as a certain cost in US$/m^2 of collector area) plus a nominal $0.02/kWh for operating and maintenance costs.

For this economic assessment, we assumed that all costs were incurred during the first year, followed by 25 years of revenue. The cost of hybridization of the existing Stillwater geothermal plant is dominated by the solar collection system. Therefore the costs presented in this report are based on the solar collector system. The additional costs for interfacing the geothermal and solar collector systems, such as piping, heat exchangers, valves, control systems and pumps are not included since these costs are small relative to the substantial uncertainty regarding the solar collector costs. In addition, no taxes or tax credits were factored into these calculations. Given these assumptions, the actual cost of electricity may differ from what is shown here. The results presented in the report should be used for relative economics for comparing alternative systems.

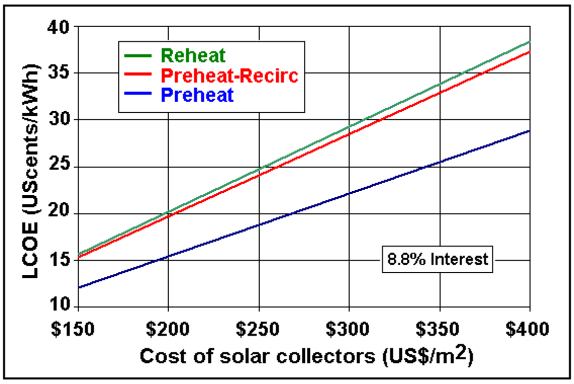

Figure 5.1: Cost of electricity versus solar collector cost.

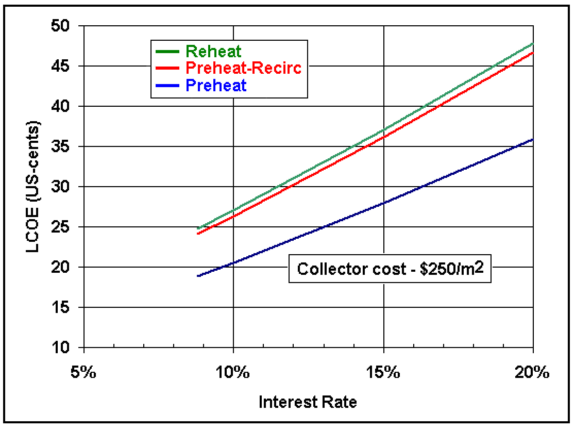

Figure 5.2: Cost of electricity versus interest rate.

The results show that the incremental unit cost of electricity ranges from about $0.19-0.25/kWh for a collector cost of $250/m^2 and an interest rate of 8.8%. Of the three viable hybrid systems, the Preheat system produces the lowest incremental LCOE. This cost may be compared to the current cost to generate a kWh from an existing plant to determine if adding the solar field to form a geothermal-solar hybrid plant makes economic sense.

6. CONCLUSION

Several power cycle configurations and working fluids were evaluated in terms of both thermodynamic and economic metrics. Accurate pure component property data and equation of state models were utilized using the Aspen Plus platform for mass and energy balance calculations. A detailed model for the solar-geothermal system was developed. Turbine flexibility relative to vapor flow rate, temperature, pressure variations was analyzed. A parametric steady-state study was carried out to examine the performance over the range of conditions resulting from diurnal and seasonal variations. The results of the diurnal and seasonal parametric studies were grossly weighted to approximate a typical year in Stillwater, Nevada, and these results led to an estimate of the annualized electricity generation.

In previous considered hybrid cycles, the lesson learned was that for the best hybrid configuration, the low-exergy, low-cost heat source (the geothermal energy) should be used to the maximum extent possible within its temperature limits, while the high-exergy, high-cost heat source (the solar energy) should be used only to extend the temperature above the temperature of the low-cost heat source.

The main conclusion from the current analysis is that among the hybrid configurations evaluated, the flash-geothermal hybrid system showed the most promise and was selected for dynamic analysis. Though this analysis has not been finished, it is expected to reveal how this model would perform under a typical winter and summer day, thereby reveal economic performance.