INL/EXT-10-20102

Interim Report: Air Cooled Condensers for Next Generation Geothermal Power Plants Improved Binary Cycle Performance

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The INL is a U.S. Department of Energy National Laboratory
operated by Battelle Energy Alliance

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Prepared for the
U.S. Department of Energy
Assistant Secretary for Energy Efficiency and Renewable Energy
Under DOE Idaho Operations Office
Contract DE-AC07-05ID14517

Introduction

As geothermal resources that are more expensive to develop are utilized for power generation, there will be increased incentive to use more efficient power plants. This is expected to be the case with Enhanced Geothermal System (EGS) resources. These resources will likely require wells drilled to depths greater than encountered with hydrothermal resources, and will have the added costs for stimulation to create the subsurface reservoir. It is postulated that plants generating power from these resources will likely utilize the binary cycle technology where heat is rejected sensibly to the ambient. The consumptive use of a portion of the produced geothermal fluid for evaporative heat rejection in the conventional flash-steam conversion cycle is likely to preclude its use with EGS resources. This will be especially true in those areas where there is a high demand for finite supplies of water. Though they've no consumptive use of water, using air-cooling systems for heat rejection has disadvantages. These systems have higher capital costs, reduced power output (heat is rejected at the higher dry-bulb temperature), increased parasitics (fan power), and greater variability in power generation on both a diurnal and annual basis (larger variation in the dry-bulb temperature).

This is an interim report for the task ***'****A*ir-C*ooled* C*ondensers in Next*- G*eneration* C*onversion* S*ystems***'.** The work performed was specifically aimed at a plant that uses commercially available binary cycle technologies with an EGS resource. Concepts were evaluated that have the potential to increase performance, lower cost, or mitigate the adverse effects of off-design operation. The impact on both cost and performance were determined for the concepts considered, and the scenarios identified where a particular concept is best suited. Most, but not all, of the concepts evaluated are associated with the rejection of heat. This report specifically addresses three of the concepts evaluated: the use of recuperation, the use of turbine reheat, and the non-consumptive use of EGS make-up water to supplement heat rejection.

Design Base

Two locations; Grand Junction, Colorado and Houston, Texas; were chosen as the basis for geothermal binary plant design because of their proximity to areas having high heat flow based on review of Southern Methodist University's 2004 Surface Heat Flow Map. Neither of these locations has been evaluated for power generation using conventional hydrothermal resources. In addition to the proximity to high heat flow, the Houston area has geopressurized resources and co-produced fluids, which could also utilize the air-cooled binary technology for power generation.

For this study, the geothermal resource temperature used for the Grand Junction area was 200°C, while the resource temperature used for the Houston area was 150°C; for both locations it was assumed that the fluid produced at the indicated temperatures was subcooled.

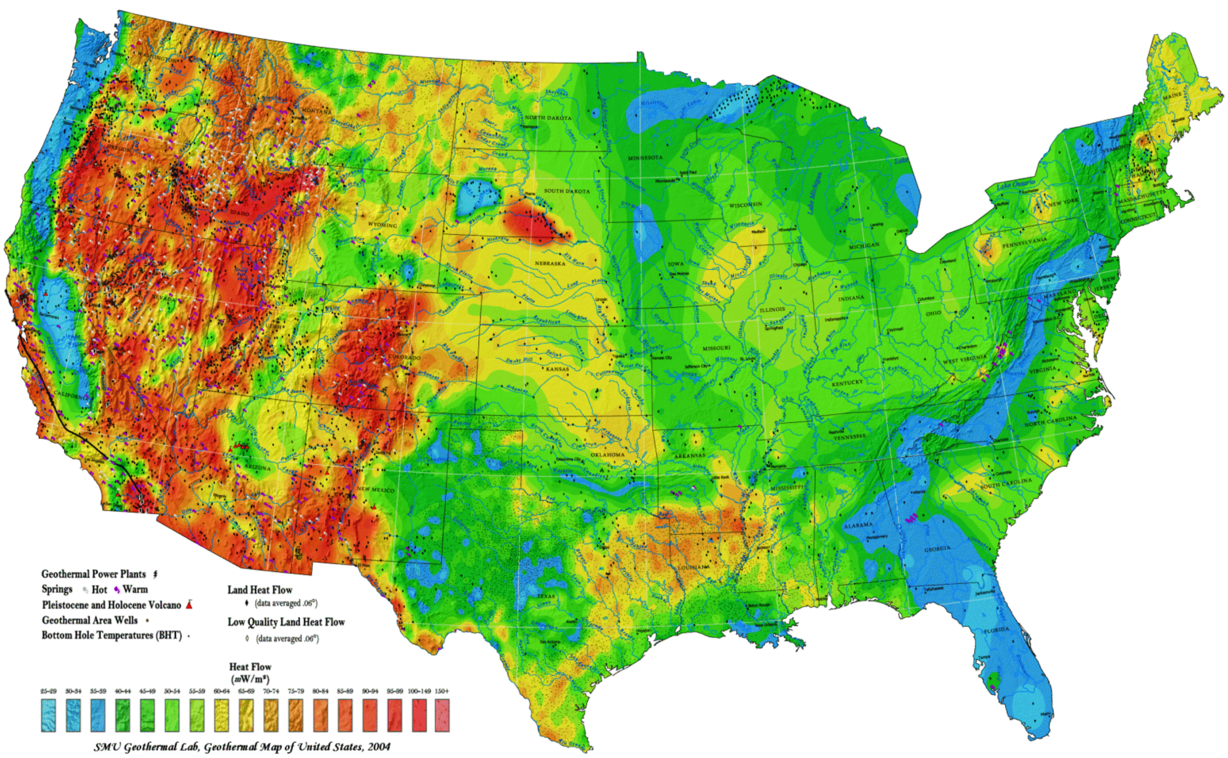

Figure 1. Southern Methodist University Geothermal Laboratory 2004 Surface Heat Flow Map

One reason for the selection of these specific locations was the availability of climatic data. Hourly ambient temperature data for the selected plant locations during the calendar year of 2009 were downloaded from the University of Utah's MesoWest web site. Houston temperature data from the Hooks Memorial Airport weather monitoring station (KDWH) and Grand Junction temperature data from the Walker Field weather monitoring station (KGJT) were implemented in determining the basis for the ambient temperature conditions at each location. This data was used to determine the minimum, median, and maximum temperatures occurring at each plant location during 2009. This temperature data was then used to create an array of eight, approximately evenly spaced, temperature points for each of the plant design locations that were used for input to fixed plant simulations to be described in detail later in this report.

Plant Design Configurations

The basic plant design evaluated is an air-cooled binary plant. A simple schematic for the plant is shown in Figure 2. In this plant the energy from the geothermal fluid is used to preheat, vaporize and superheat a pressurized secondary working fluid. The high pressure working fluid vapor is subsequently expanded in a turbine that drives an electrical generator. The low pressure working fluid vapor exiting the turbine is condensed in an air cooler and pumped back to the geothermal heat exchangers. For this study, it is assumed at the working fluid is vaporized at a single pressure; i.e., dual boiling cycles were not evaluated.

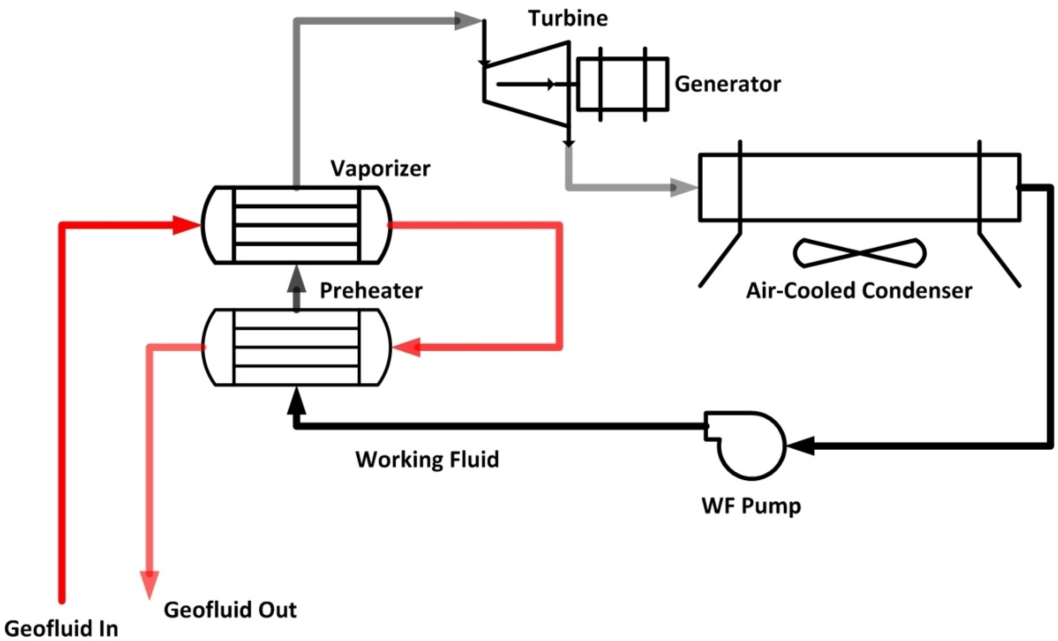

Figure 2. Simple air-cooled binary cycle schematic

This study assessed modifications to this basic plant design having the potential to improve performance when the plant operation deviates from the design resource and ambient conditions. The plant design configurations considered included the technologies described in the following sections.

Recuperation.

The recuperated plant design incorporates an additional heat exchanger that transfers heat from the turbine outlet stream to the pump outlet stream; the working fluid flows through both sides of the heat exchanger. A schematic of a binary cycle with recuperation is shown in Figure 3.

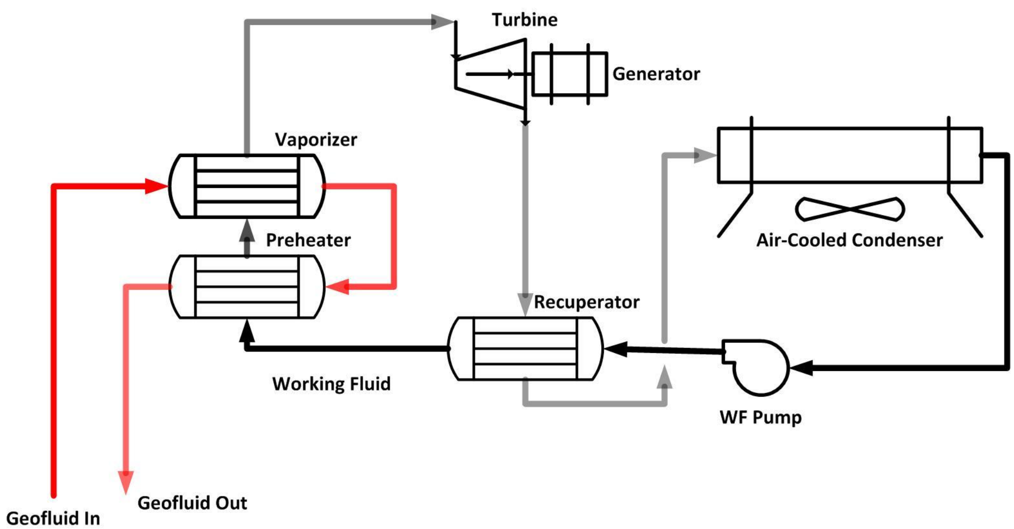

Figure 3. Binary cycle with recuperator

The advantage of using the recuperator is a decrease in the heat duty in both the air-cooled condenser and the geothermal heat exchangers. This can lead to a reduction in the cost of the condenser and geothermal heat exchangers, could allow more working fluid to be vaporized from a fixed geothermal fluid flow and consequently more power generated, or could reduce the fan power because of the reduced heat load resulting in more net power production from the plant. These benefits will be offset to some extent by the pressure drop associated with the recuperator, which increases the turbine exhaust pressure relative to the condenser pressure.

Makeup Water Condenser

When EGS resources are utilized, it is postulated that there will be subsurface water losses that will make it necessary to provide makeup water to the geothermal system. In the design considered, the make-up to the geothermal system is used to condense a portion of the turbine exhaust stream and reduce the air-cooled condenser duty. A schematic of a binary cycle with this condenser is shown in Figure 4.

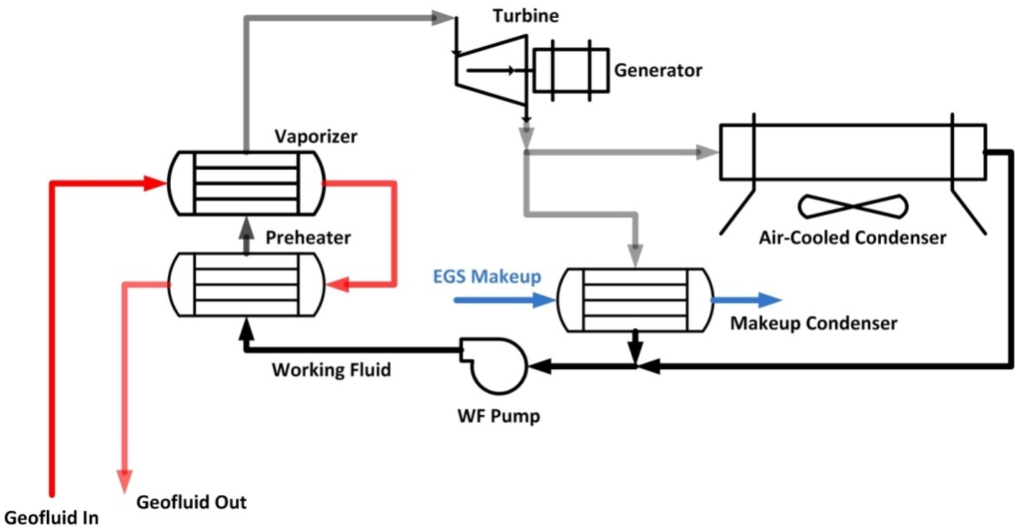

Figure 4. Binary cycle with makeup water condenser

As depicted above, the makeup water condenser operates in parallel with the air cooled condenser so as to not introduce additional pressure drop. Because the heat transfer coefficients for water are significantly higher than they are for air, heat rejection done by this condenser is expected to lower the total cost for heat rejection. In this design it is assumed that the makeup water is available at a mass flow rate up to 5% of the geothermal fluid flow and at a temperature equal to the design ambient temperature.

Reheat Turbine.

The reheat turbine plant design utilizes an additional heat exchanger and turbine. A schematic showing a binary cycle with this reheat is shown in Figure 5.

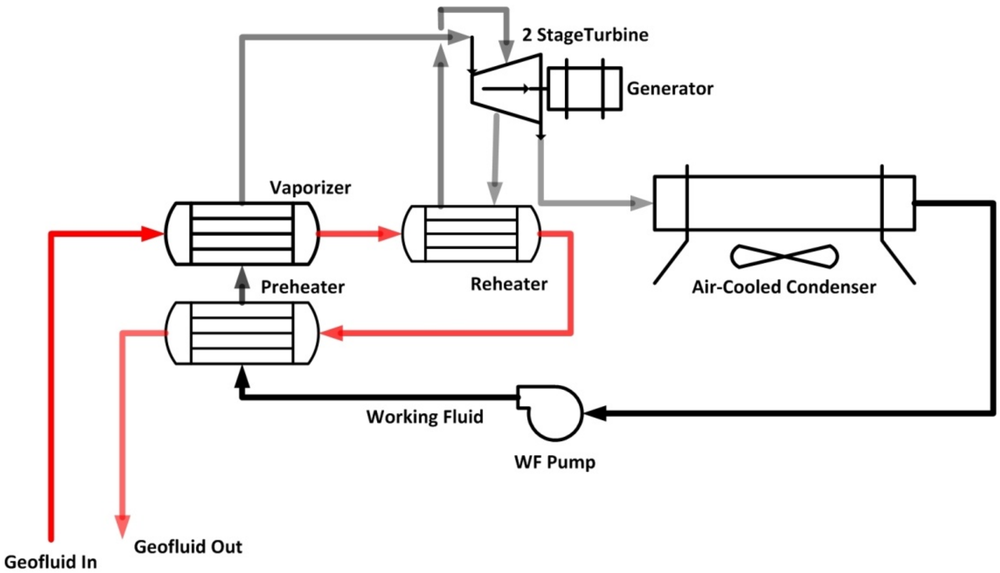

Figure 5. Binary cycle with reheat turbine

In this configuration design, the working fluid exits the vaporizer at a temperature such that the subsequent isentropic expansion in a turbine would enter the two phase region. Prior to this occurring, the working fluid exhausts the higher pressure turbine and is heated again to a temperature that assures the subsequent expansion in a low pressure turbine occurs completely outside of the two-phase region. This reheating is accomplished in the added heat exchanger using a portion of the cooled geothermal fluid exiting the vaporizer. The potential benefit of this concept is limited to those working fluids like propane and R134a that do not have the retrograde dew point line (on a T-s diagram). As a consequence it would be more likely used with lower temperature resources.

Plant Model Development

Binary plant design models were developed using AspenTech Aspen Plus version 2006.5 process simulation software. The process flow diagrams were constructed using multiple Aspen Plus unit operation (or block) models. Multistream heat exchanger blocks were used to model the preheater, boiler, air cooled condenser, as well as the recuperator, geothermal makeup water condenser, and reheat turbine heat exchanger in the analyses that included those options. Compressor/turbine blocks were used to model the working fluid turbine as well as the condenser fan. A pump block was used to model the pump and a valve block was implemented to model the control valve. Frictional pressure losses throughout the model were accounted by setting fixed pressure drops in component blocks. The Peng-Robinson property method was used to calculate working fluid and air properties, while the STEAM-TA property method (1967 ASME steam table correlations for thermodynamic properties, International Association for Properties of Steam (IAPS) correlations for transport properties) was used to calculate geothermal fluid properties. A flow diagram of one of the configurations evaluated is shown in Figure 6.

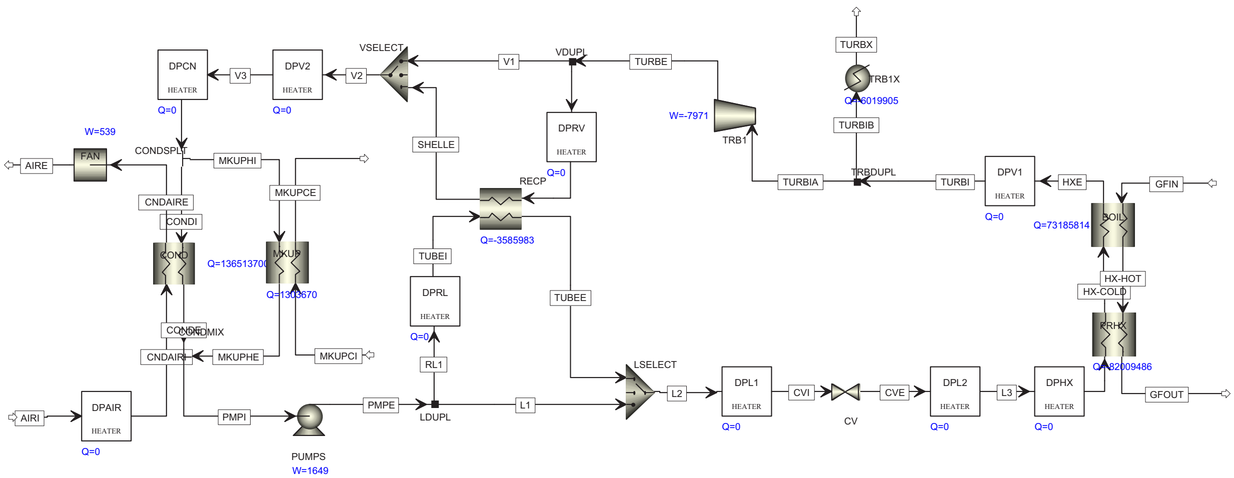

Figure 6. Binary geothermal plant process flow diagram with recuperation and makeup water condenser

The methodology for evaluating plant design configurations included setting several values in the plant to predetermined values. These values included heat exchanger minimum temperature approach (or pinch points), magnitude of frictional pressure losses (in all relevant blocks and process piping), the preheater outlet temperature, the extent of subcooling provided by the condenser, and the efficiencies of the turbine, pump, and fan. The preheater outlet design temperature was set equal to the bubble point temperature of the working fluid at the heater pressure; if the heater pressure was above the critical pressure, the working fluid's critical temperature was used. The specific quantities used for the remaining design values were determined through analysis of actual geothermal plant operating data and engineering judgment.

The plant design ambient conditions were set equal to the 2009 hourly temperature data median values obtained for each geographic location from the MesoWest web site. The geothermal fluid temperature was set equal to 150°C for the Houston, Texas plant design and 200°C for the Grand Junction, Colorado plant design.

Table 1. Resource and ambient design conditions

Location	Design Ambient Temperature	Design Geothermal Fluid Temperature
Houston, Texas	71.1°F (21.7°C)	302°F (150°C)

Grand Junction, Colorado	53.1°F (11.7°C)	392°F (200°C)

The preheater outlet temperature, condenser subcooling, block and piping frictional pressure losses, and block efficiencies were all set directly within the appropriate Aspen Plus simulation blocks. The equipment and piping design frictional pressure losses are detailed in Table 2 and the design efficiencies are detailed in Table 3. The condenser working fluid outlet was specified at a fixed value of 2°F below the saturation temperature.

Table 2. Design frictional pressure losses

Unit Operation/Process Piping Section	ΔP_{design} (psia)
Control Valve	2
Preheater and Boiler combined pressure drop	38
Recuperator vapor side (shell side)	1
Recuperator liquid side (tube side)	5
Condenser working fluid side	1
Condenser air side	calculated
Pump to control valve piping	5
Control valve to preheater piping	5
Boiler to turbine piping	3
Turbine to condenser piping	1

Table 2 lists the condenser air side design pressure drop value as "calculated". This value is calculated because neither the air flow rate, condenser size, or air temperature rise are fixed in the model. As a consequence it is unrealistic to use a fixed static pressure drop across the condenser tube bundle. To relate the air pressure drop to the condenser size and air flow, the following relationship was used.

$$\Delta P = \Delta P_D \cdot \left(\frac{Q}{Q_D} \cdot \frac{UA_D}{UA}\right)^2$$

where

ΔP, ΔP_D = pressure drop, design pressure drop = 0.260 in H_2O

Q, Q_D = actual volumetric flow rate, design actual volumetric flow rate

UA, UA_D = product of overall heat transfer coefficient U and heat exchanger area A, UA_D (subscript D corresponds to design case)

In this relationship, the parameters with the D subscript were taken from the design specification sheet for an air-cooled condenser in an operating binary plant. The UA term is the product of overall heat transfer coefficient and heat exchange area; if not quoted, it can be calculated by dividing the specified heat exchange duty by the log mean temperature difference (either specified or calculated). For the calculation in the simulation it is assumed that the U value remains constant so that the ratio of the Q/UA's is effectively the ratio of the air velocities for the condenser. The simulation pressure drop is then the reference pressure drop times the square of this air velocity ratio. It is recognized that the U will not remain constant and will vary with air velocity, however this is a more representative depiction of pressure drop than assuming a constant air-side pressure drop, and is considered to suffice for these studies.

Table 3. Design efficiency values

Simulation block	η_{design} (%)
Turbine, isentropic	83
Turbine, mechanical	94
Pump	80
Pump driver	98
Fan, isentropic	55
Fan, mechanical	90

The minimum temperature approach values are dependent variables that must be set by one or more independent variables that exist within the process simulation. The minimum temperature approach design values were obtained by implementing Aspen Plus design specs that manipulated the appropriate independent variables required to drive the minimum temperature approach values to the design conditions. Table 4 details the minimum temperature approach design values as well as the variables manipulated by the Aspen Plus design specs to achieve the design conditions.

Table 4. Minimum temperature approach design values

Heat Exchanger Block	Minimum T Approach (°F)	Variables manipulated to achieve design condition
Preheater and Boiler (pinch point resides in only one of the two blocks)	10	Working fluid mass flow rate
Condenser	15	Air mass flow rate
Recuperator	10	Recuperator duty
Geothermal Makeup Water Condenser	10	Fraction of working fluid condensed in makeup water condenser

Following specification of the aforementioned plant design values, the simulation net power output was maximized by implementing the Aspen Plus optimization feature. The Aspen Plus optimization was configured to vary the turbine outlet pressure, pump outlet pressure, and boiler outlet temperature so as to maximize the net power objective function:

$$P_{net} = P_{trb} - P_{pmp} - P_{fan}$$

where

P_{trb} = turbine power

P_{pmp} = pump power

P_{fan} = fan power

Completed plant design simulations yielded optimized design point operating conditions and power output as well as specifications for all equipment included in the plant design.

Design Constraints

In this analysis, the performance of an air-cooled binary plant was optimized, both with and without a constraint on the temperature of geothermal fluid leaving the plant. This constraint is common when using hydrothermal resources to preclude the precipitation of dissolved solids in the surface equipment. Typically it is imposed to prevent the precipitation of amorphous silica, which goes into solution in the subsurface as quartz. The solubility of both quartz and amorphous silica increase with the fluid temperature, hence as the resource temperature increases so does the minimum temperature needed to prevent silica precipitation. A temperature constraint was integrated into the model that was based on preventing silica precipitation. Based upon solubility equations for both quartz and amorphous silica (Gunnarsson and Arnorsson, 2000), a correlation was developed that predicted the temperature constraint based upon the production fluid temperature.

In evaluating binary plant performance an upper limit of 1,200 psi was placed on the working fluid system pressure. Operation at higher pressures was not considered to be likely because of the additional cost for system components when these pressures were used (costs increase not necessary as a result of higher component pressures, but also due to the increased pump and turbine-generator sizes).

In evaluating the plant performance with different working fluids, a requirement was imposed that the turbine expansion be 'dry', i.e., no portion of the turbine expansion occurs within the two-phase region. This requirement is commonly used by operators of commercial binary plants to prevent damage to turbine internals exposed to vapor having entrained droplets, as well as the adverse impact that 'wet' expansions have on turbine efficiency. Though there has been work that indicates this constraint is overly conservative (Mines, 2000), it has been imposed in these studies.

To assure that the turbine expansion remained dry, a minimum constraint was placed on the entropy of the vapor entering the turbine. This constraint is illustrated in Figure 7.

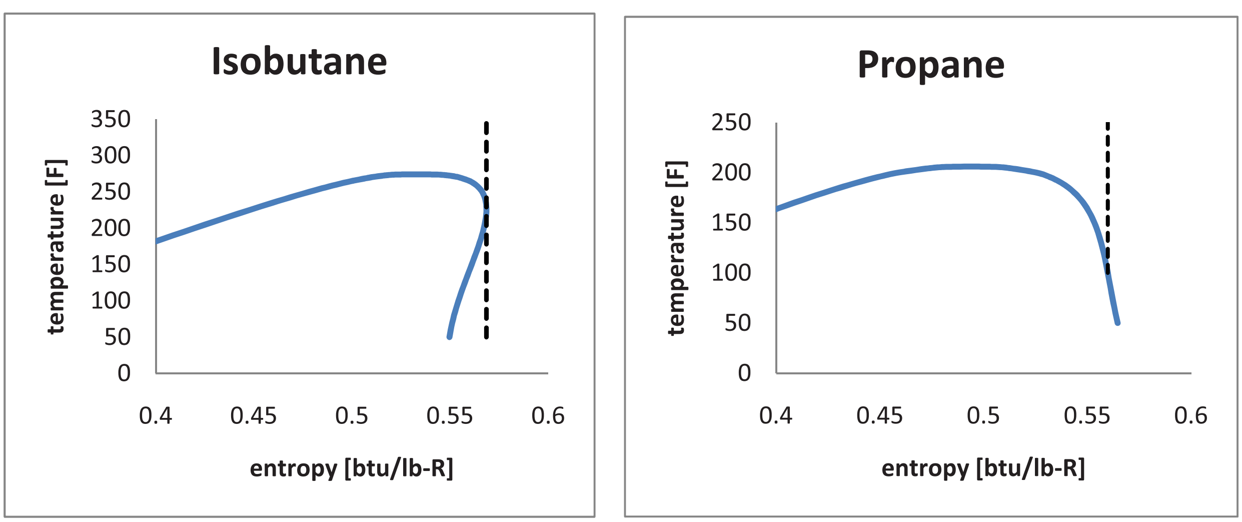

Figure 7. Inlet temperature constraint on vapor entering turbine

For those fluids having retrograde dew point curves like isobutane, for saturation conditions up to the maximum dew point entropy (dashed line in figure on the left) a minimum level of superheat (1°F) was imposed on the vapor entering the turbine. For saturation conditions above the maximum dew point entropy, as well as for supercritical pressures, the minimum turbine inlet temperature was established by the maximum dew point entropy. For fluids like propane where the dew point entropy increases with a decreasing temperature (figure on the right), the minimum turbine inlet entropy was established by the dew point entropy at the exhaust pressure (dashed line in right figure). The imposition of these constraints on the entropy (or superheat) of the vapor entering the turbine assured that the resulting expansion occurred outside of the two-phase region (to the right of the dashed lines in both of the above figures).

Working Fluids

Several different working fluids were investigated at each geographic plant location to identify the working fluid that would yield the maximum power output from a fixed geothermal fluid flow rate at the design resource and ambient temperature conditions. Working fluids evaluated included propane (C3), isobutane (iC4), n-butane (nC4), isopentane (iC5), R-134a, and R-245fa.

Fixed Plant Simulations

The plant design simulations previously described provided the basis for equipment sizing and performance specifications. The plant performance data associated with these plant design simulations correspond to a single operating point that is tied to the ambient design temperature and geothermal resource design temperature. Changes to the ambient or resource temperatures force the plant to operate at off-design conditions, impacting the plant performance or output. In order to assess the benefits of the concepts evaluated in an actual plant, the performance of various plant designs was simulated at off-design ambient and resource temperature conditions.

"Fixed plant" simulations were developed to calculate the performance of geothermal plants as a function of ambient temperature fluctuations and geothermal resource temperature decline. The fixed plant simulations replicate the operation of the optimal plant designs for each geographic location at off-design temperature conditions. For given off-design conditions, the fixed plant simulations determine maximal net power output by optimizing plant operating parameters analogous to those that would be controlled by the operator of an actual geothermal plant. These operating parameters include the working fluid mass flow rate, control valve pressure drop, and air mass flow rate. The plant operating point determined by these variables could be replicated by a geothermal plant operator by manipulating the control valve position, turbine vane position, and condenser fan operation (turning fans on or off).

The fixed plant simulations accounted for the physical effects and changes in equipment performance that were deemed to be the major factors affecting off-design plant performance. The principles and calculations used to determine off-design plant performance are described below. This plant performance data was then used to determine whether the concepts evaluated increased net power production over plant operating life for specified ambient and resource temperature versus time scenarios.

Pump Curve

The design pump operating point was specified by each plant design simulation. When the fixed plant simulation working fluid mass flow rate is changed, a new pump outlet pressure and pump efficiency must be calculated. Pump curves specific to each fixed plant simulation were required to calculate these pump outlet conditions.

The shape of these pump curves was determined using the spec sheet for the working fluid pumps in an operating commercial binary plant. A normalized pump curve, with the design point flow rate corresponding to an x-axis value of unity and the design point head and efficiency values occupying the corresponding unity positions on the dual y-axis plot, was generated by curve fitting the pump curve for the commercial plant. The fixed plant pump curve was then generated from this normalized pump curve by using the pump operating point from the plant design simulation as the design condition. The normalized pump curve is shown in Figure 8.

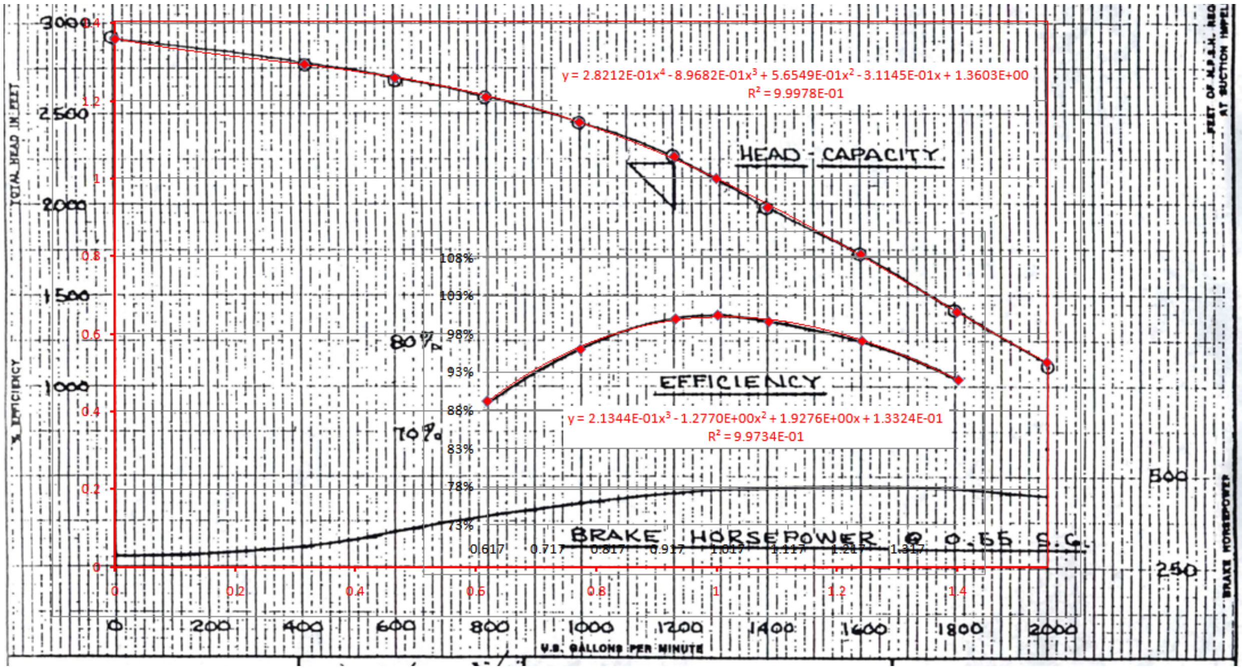

Figure 8. Normalized pump curve (head and pump efficiency vs. volumetric flow)

Turbine

The method used to characterize operation of the turbine quantifies how the turbine reacts as the operating conditions both in and out of the turbine deviate from those for which it was designed. The two parameters quantified by this methodology are effects on the flow rate of the working fluid through the turbine and on the turbine efficiency. For this analysis it is assumed that the turbine is a single stage, reaction turbine. A similar approach would have been used for an impulse turbine.

The working fluid flow through a turbine is choked, i.e., the vapor flow the throat of the turbine's nozzle is at the sonic velocity. For a constant throat area, the working fluid flow varies with both the density of the vapor at the throat and the sonic velocity. The vapor density varies directly with the pressure and indirectly with temperature, and if one wants to decrease flow through the turbine, it is necessary to increase the temperature or decrease pressure. Decreasing the turbine inlet pressure will decrease the necessary pump head and increase the flow produced by the pump. In order to operate a turbine at 'off-design' conditions, it becomes necessary to throttle flow either leaving the pump or entering the turbine, both of which increase cycle irreversibilities and lower plant performance. Operation at the off-design conditions are inevitable as the ambient or resource temperature deviate from the plant's design values.

As indicated, flow through the turbine is a function of both the sonic velocity and the fluid density at the throat of the turbine nozzle. One method of defining the throat conditions is to identify the pressure where the mass flux (flow/area, or velocity times density) is a maximum for an isentropic expansion from a given inlet condition. In this approach the velocity, V, is defined as

$$V = \left(2 \cdot g \cdot \Delta h_{isentropic}\right)^{0.5}$$

where

g is the acceleration due to gravity and

$\Delta h_{isentropic}$ is the isentropic enthalpy change from the inlet to a given pressure

As the pressure is reduced from the inlet condition, the velocity increases and the density (at this pressure and the entropy at the turbine inlet) decreases. At some reduced pressure, the product of this velocity and density goes through a maximum. This maximum defines operation at the throat of the turbine nozzle.

In the modeling performed a relationship was developed for defining the pressure at the throat of the nozzle.

$$P^* = 0.67 \cdot P_{trb,in} \cdot \left(\frac{P_{trb,in}}{P_{crit}}\right)^{0.2} \cdot \left(\frac{T_{trb,in}}{T_{crit}}\right)^{-1}$$

where

P = absolute pressure

T = absolute temperature

superscript “*” denotes turbine nozzle throat conditions

subscript “crit” denotes critical condition

Using this relationship to define the throat pressure, the enthalpy change and density could be defined for an isentropic expansion from the design turbine inlet condition and used to determine the mass flux, With the design flow rate established, this relationship allowed the total nozzle area in the turbine to be defined. Once this throat area was known, the mass flow rate could then be calculated for turbine inlet conditions other than design.

The analysis that was conducted assumed that the turbine had variable nozzle geometry, which allowed the throat area to be changed. This provides an additional degree of freedom in maximizing plant performance and is likely to be used in future plants using EGS resources. In establishing the design conditions, it was assumed that the turbine was oversized, i.e., the throat area at the design was 80% of the total available. This provision allowed additional flexibility in optimizing performance at the off-design condition, and is used in some commercial plant turbine designs.

The use of variable nozzle geometry reduces the efficiency of the turbine when operation requires a level of throttling that deviates from the optimal. Using similar information from different sources (Baljae, 1981 and SAE, 1969), a correlation was developed that related the change in turbine efficiency to the change in throat area resulting from manipulating the nozzle geometry.

In order to characterize the effect of varying inlet and exhaust conditions on turbine performance, information in Baljae was used to relate the turbine efficiency to a velocity ratio for a reaction turbine. The velocity ratio is the ratio of the tip speed to the spouting velocity, which is the velocity one would have if the potential energy defined by an isentropic expansion (enthalpy change) were converted to kinetic energy (velocity).

At off-design conditions, these relationships established the flow the turbine was able to pass, and the effect of those conditions on the turbine's efficiency, both in terms of changes in the inlet and exhaust conditions, as well as any throttling needed to adjust the flow to match the desired inlet pressure. These correlations were integrated into the model's depiction of the turbine, and along with Aspen Plus' ability to provide fluid properties allowed these efficiencies to be calculated.

Heat Exchangers

Changes to the fluid mass flow rates will affect the heat transfer occurring in each of the geothermal plant heat exchangers. Heat exchanger duty is a function of the temperature difference between the hot and cold sides of the heat exchanger; the overall heat transfer coefficient, which is a function of heat exchanger geometry, fluid properties, and flow regime; as well as heat transfer area.

Heat exchanger geometry and area are constant by definition in the fixed plant simulations. However, changes to fluid flow rates will affect the fluid flow regime, equipment operating pressures, and temperature differences throughout each heat exchanger; which in turn will affect fluid physical properties. Changes in the fluid flow regime were assumed to have a greater affect on the overall heat transfer coefficient than changes in fluid properties (viscosity, density, heat capacity, and thermal conductivity).

Table 5. Heat transfer coefficient correction factors

	Hot Side				Cold Side			
Heat Exchanger	Geometry	Fluid	Governing heat transfer correlation	Correlation Reynolds Number exponent	Geometry	Fluid	Governing heat transfer correlation	Correlation Reynolds Number exponent
Preheater	Tube	GF	Sieder-Tate	0.8	Shell	WF	Donohue	0.6
Boiler	Tube	GF	Sieder-Tate	0.8	Shell	WF	Donohue	0.6
Condenser	Tube	WF	Boyko and Kruzhilin	0.8	Fin	air	Zhukauskas	0.6
Recuperator	Shell	WF (vap)	Donohue	0.6	Tube	WF (liq)	Sieder-Tate	0.8
Makeup Water Condenser	Tube	WF	Boyko and Kruzhilin	0.8	Shell	GF	Donohue	0.6

The Sieder and Tate (Bell and Mueller, 2001), Donohue (1949), Boyko and Kruzhilin (1967), and Zhukauskas (1972) heat transfer correlations were used to determine the dependence of the heat transfer coefficients on fluid mass flow rates. A correction to the UA of each heat exchanger was applied to account for changes in fluid mass flow rates according to the following equation:

$$UA = UA_D \cdot \left[1 + R_{hot} \cdot \left(\left(\frac{\dot{m}_{hot}}{\dot{m}_{hot,D}}\right)^{exp_{hot}} - 1\right) + R_{cold} \cdot \left(\left(\frac{\dot{m}_{cold}}{\dot{m}_{cold,D}}\right)^{exp_{cold}} - 1\right)\right]$$

where

UA = product of overall heat transfer coefficient and heat exchange area

R = heat transfer resistance associated with specified side of heat exchanger, proportional to reciprocal of specified heat transfer coefficient

$\dot{m}$ = mass flow rate

exp = exponent of Reynolds number in governing heat transfer correlations listed in Table 5

subscript “D” denotes design condition value

subscripts “hot” and ”cold” denote hot and cold side of heat exchanger, respectively

In order to set the fixed plant UA equal to the value calculated using the above mass flow rate correction equation, one independent variable dependent on the UA of each heat exchanger was manipulated. These variables are described in Table 6.

Table 6. Fixed plant heat exchanger manipulated variables

Heat Exchanger Block	Variable manipulated to achieve corrected UA
Preheater	Working fluid outlet temperature
Boiler	Geothermal fluid outlet temperature
Condenser	Turbine outlet pressure
Recuperator	Recuperator duty
Makeup Water Condenser	Fraction of working fluid condensed in makeup water condenser

Frictional losses

In addition to affecting pump performance and heat transfer processes, changes in the fluid mass flow rates will also affect the magnitude of frictional losses in individual unit operations as well as in process piping. The fixed plant frictional losses were determined by applying the following equation to all defined pressure drops in the process.

$$\Delta P = \Delta P_D \cdot \left(\frac{\dot{m}/\rho}{\dot{m}_D/\rho_D}\right)^2$$

where

ΔP = unit operation or piping pressure drop

$\dot{m}$ = mass flow rate

ρ = fluid density

subscript "D" denotes design condition value

The quantity $\dot{m}/\rho$ is equal to the fluid volumetric flow rate. Since the plant geometry is fixed, the numerator and denominator of the term in parentheses in the expression above can be divided by the cross-sectional area of the unit operation or piping in question to yield a ratio of the fluid velocities. The change in pressure drop is therefore proportional to the square of the fluid velocity divided by the design fluid velocity.

Results

Base Plant Design – Optimal Performance

The optimal working fluid was evaluated for the two geograhic location scenarios both with and without the temperature constraint on the geothermal outlet temperature. This optimization was based upon the previously described assumptions relative to pinch points, component efficiencies and pressure drops. For each working fluid, the turbine exhaust, working fluid flow rate, boiler outlet temperature, and pump discharge pressure were varied until a maximum net power was found for the fixed geothermal flow rate.

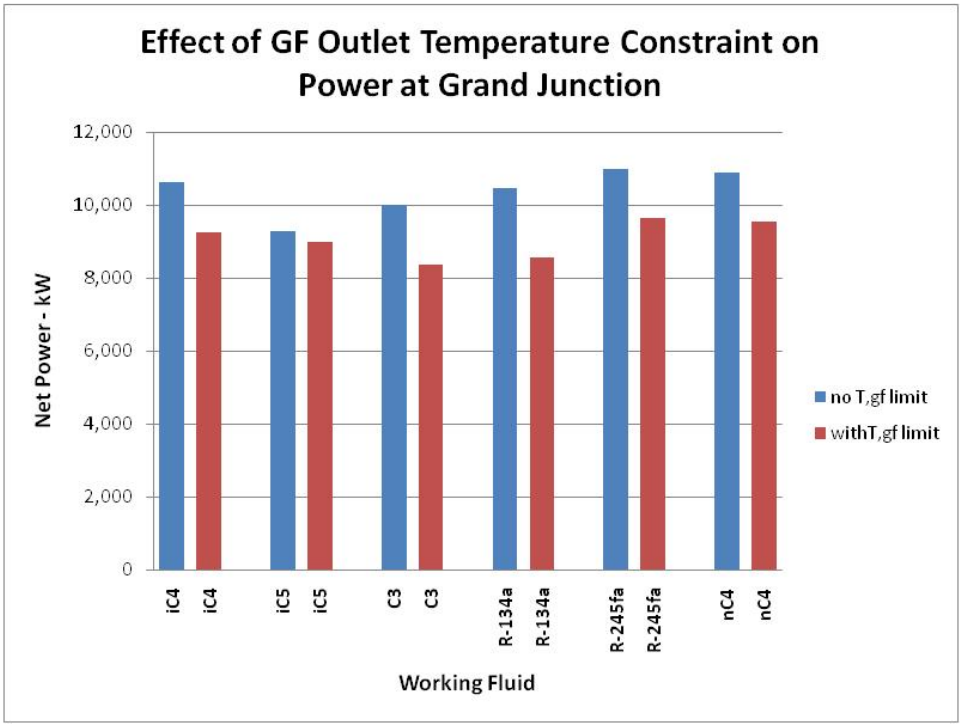

Figure 9. Baseline plant performance using different working fluids for Grand Junction design scenario

Results for the Grand Junction optimal plant performance with each fluid are shown in Figure 9. These results indicate the imposition of the geothermal fluid outlet temperature constraint adversely impacts the plant performance regardless of the working fluid used. Without a constraint, the use of R245fa and n-butane resulted in the highest net power for the assumptions made; these fluids also provided the highest levels of performance with the outlet temperature constraint imposed. A similar evaluation was made for these working fluids at the Houston location. Those results are shown in Figure 10.

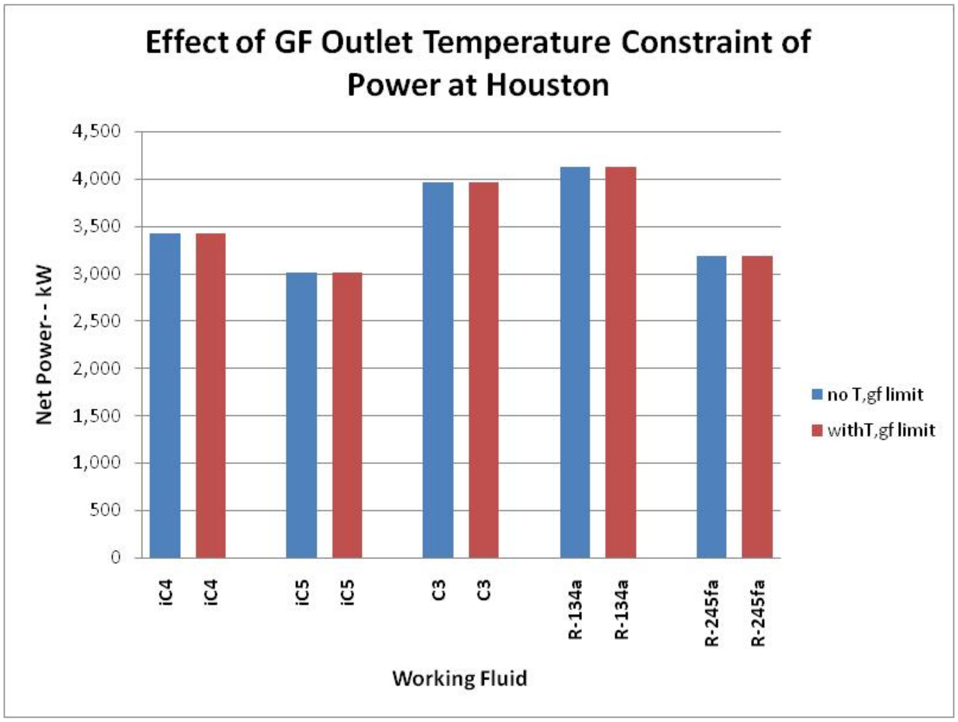

Figure 10. Baseline plant performance using different working fluids for Houston design scenario

At the Houston location the outlet temperature constraint had no effect on the power produced with any of the fluids considered. This is because for the optimized scenarios the geothermal temperature leaving the plant was greater than the calculated value at which silica precipitation would occur. At this resource design condition, the optimal working fluid was R134a, with propane's performance being slightly less.

For the design scenarios at both locations, the power cycles with the fluids having the optimal levels of performance operated at supercritical turbine inlet pressures (pressures above the critical pressure of the working fluid). To maximize the power produced for the assumptions made, a plant designed for the Grand Junction conditions would use either R245fa or n-butane, while a plant designed for the Houston conditions would use R134a.

Base Plant Design – Minimum Cost

In order to assess the impact of the plant design on capital costs, installed plant costs were developed using the equipment sizes determined for each scenario and working fluid. These costs estimates were developed using equipment costs generated previously with the ICARUS Process Evaluator (IPE) software package. Those estimates were brought to the present using the US Department of Labor, Bureau of Labor Statistic's Producer Price Indices for equipment, materials, and labor estimates. Costs for both locations are shown in Tables 7 and 8 for both assumptions relative to the outlet temperature constraint.

Table 7. Plant cost and performance for Grand Junction Scenario (with and without temperature constraint)

Fluid	No Constraint		With Constraint	
	Cost	Net Power (kW)	Cost	Net Power (kW)
Isobutane	$36,500,427	10,647	$29,626,859	9,252
Isopentane	$27,638,368	9,291	$6,036,885	9,005
Propane	$35,440,611	10,003	$28,479,057	8,361
R134a	$35,435,577	10,489	$27,650,213	8,575
R245fa	$33,828,787	10,984	$29,198,291	9,653
n-butane	$34,818,253	10,898	$29,581,785	9,567

Table 8. Plant cost and performance for Houston Scenario (with and without temperature constraint)

Fluid	No Constraint		With Constraint	
	Cost	Net Power (kW)	Cost	Net Power (kW)
Isobutane	$14,299,489	3,430	$14,299,489	3,430
Isopentane	$12,321,017	3,016	$12,321,017	3,016
Propane	$18,471,288	3,967	$18,471,288	3,967
R134a	$18,417,775	4,131	$18,417,775	4,131
R245fa	$12,917,357	3,196	$12,917,357	3,196

DOE bases its determination of the levelized cost of electricity (LCOE) for renewables on a fixed charge rate that is applied to the project's total capital cost. Using this methodology, the project having the lowest cost in terms of $ per kW will have the lowest LCOE. Assuming no cost for a well field, one would opt for the optimized base plant design using the isopentane working fluid at Grand Junction and R245fa at Houston as they would project the lowest cost in terms of $/kW (see Figure 11 and Figure 12).

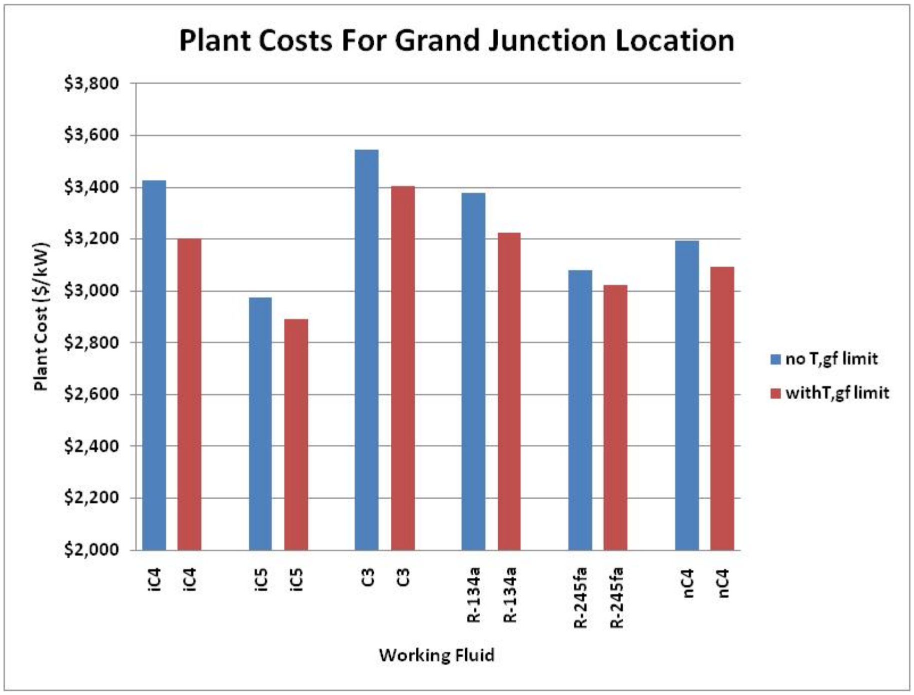

Figure 11. Baseline plant cost using different working fluids for Grand Junction design scenario

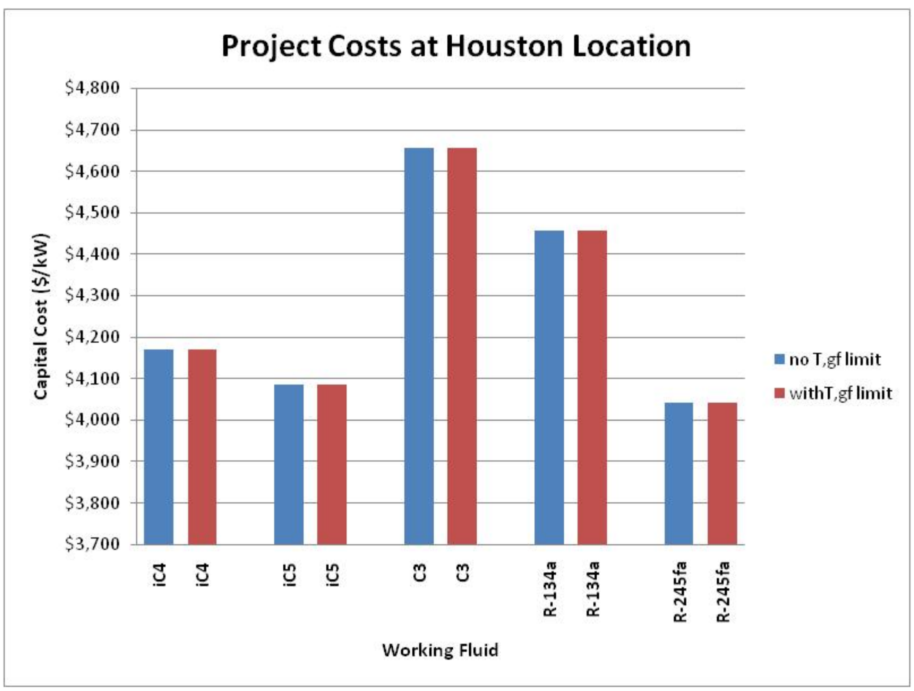

Figure 12. Baseline plant cost using different working fluids for Houston design scenario

Again, the costs in these figures assume that there is no cost associated with the well field. If one includes the well field costs, then the optimal fluid can change depending upon the magnitude of those costs. For both location scenarios, the geothermal flow rate was fixed at 1,000,000 lbs per hour, or 126 kg/s. This flow rate is more than the near term goal of 10 kg/s per well listed in DOE's Multiyear Research, Development and Demonstration Plan, which means it will be probable that more than one production well would be required to supply 126 kg/s. Assuming there are a minimum of two production wells and one injection well having drilling costs of $5,000,000 each and stimulation costs of $2,000,000 for each, the well field development costs would be $21,000,000. When this cost is included, the optimal project costs change as shown in Figure 13 and Figure 14.

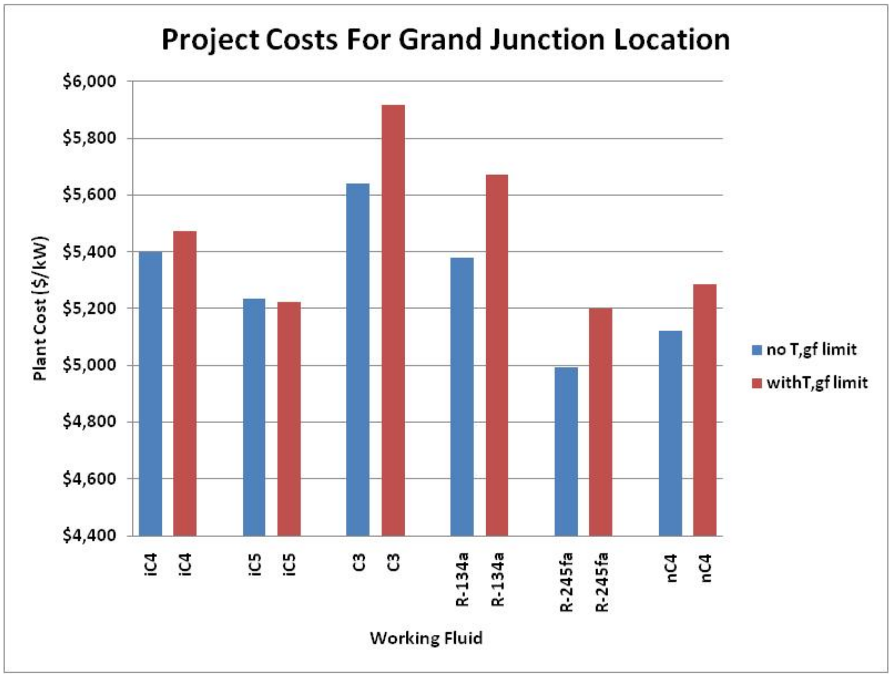

Figure 13. Baseline project cost with $21 Million field cost for Grand Junction design scenario

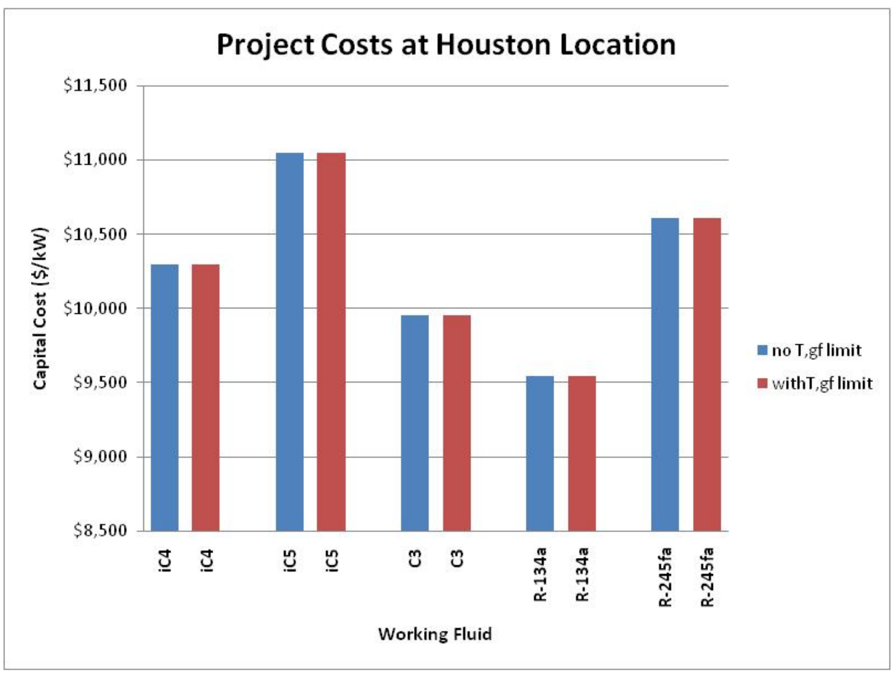

Figure 14. Baseline project cost with $21 Million field cost for Houston design scenario

When the assumed cost for developing the well field is included, the optimal design for both locations corresponds to that with the working fluid that provides the superior performance. For the Houston location, R134a would have an advantage when well field development costs exceed ~$6 Million; at Grand Junction, R245fa would have an advantage when well field costs exceed ~$6.3 Million with no geothermal fluid exit temperature limit and ~$18 Million with the temperature limit imposed.

Based on plant performance and the postulated costs for the EGS well field, R134fa would be the optimal fluid for the Houston design scenario, while R245fa would likely be the choice for the Grand Junction design scenario.

Recuperated Plant Design – Optimal Performance

A similar analysis was performed with the recuperated plant design. Grand Junction results are shown in Figure 15. At this location, the use of recuperation provides no performance benefit when there is no constraint placed on the outlet brine temperature. As was the case without recuperation, with the exception of those cycles using isopentane, optimal performance of the recuperated plant was achieved with supercritical cycles.

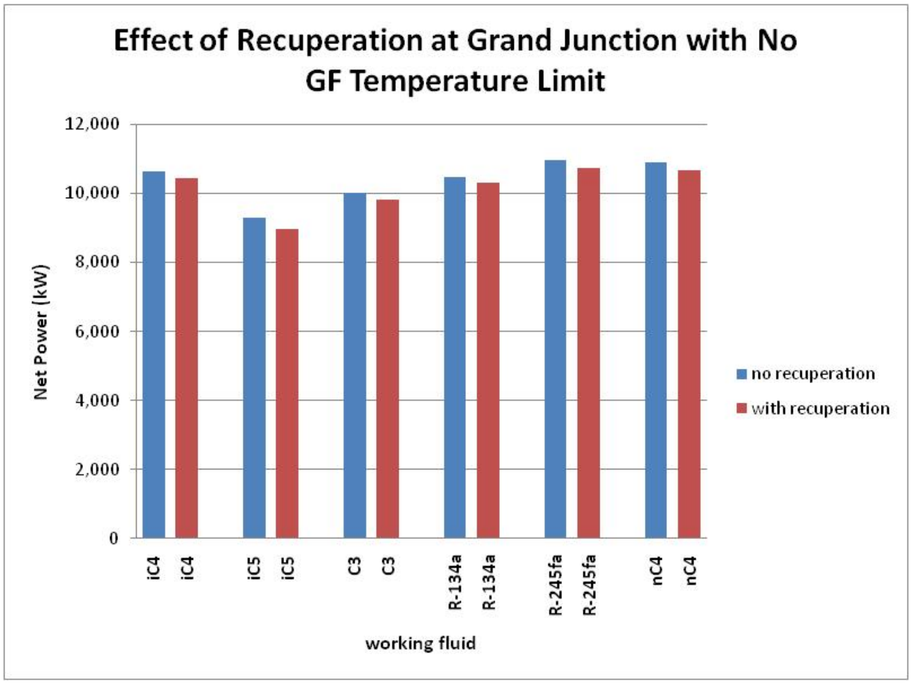

Figure 15. Recuperated plant performance for the Grand Junction scenario with no geothermal outlet temperature limit

Similar results are shown in Figure 16 for the Houston location. Results for Houston are similar to those for Grand Junction when no constraint is placed on the geothermal outlet temperature, in that recuperation provides no performance benefit for the assumptions made. Note that for the Houston scenario, the outlet temperature for the optimized conditions is always above the calculated temperature for the precipitation of amorphous silica.

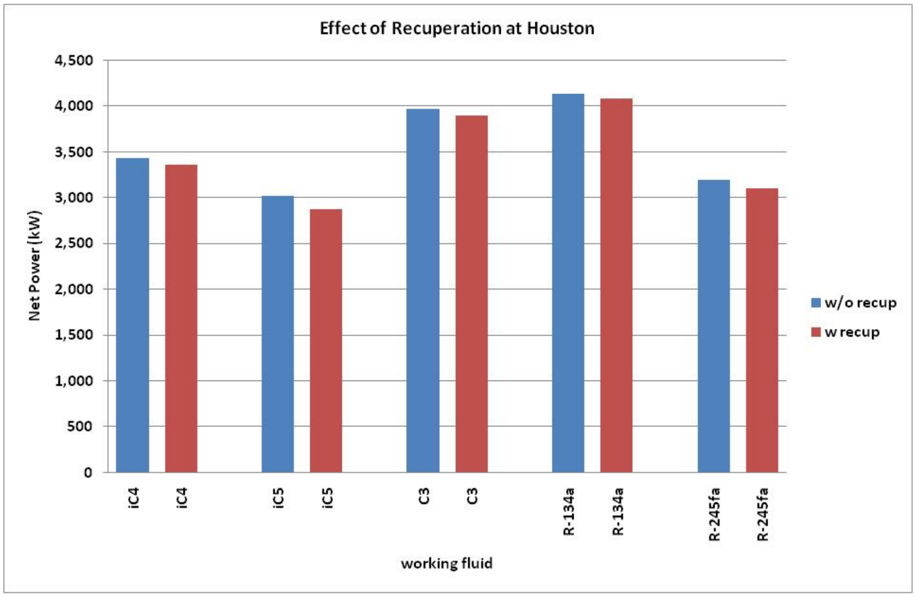

Figure 16. Recuperated plant performance for the Houston scenario

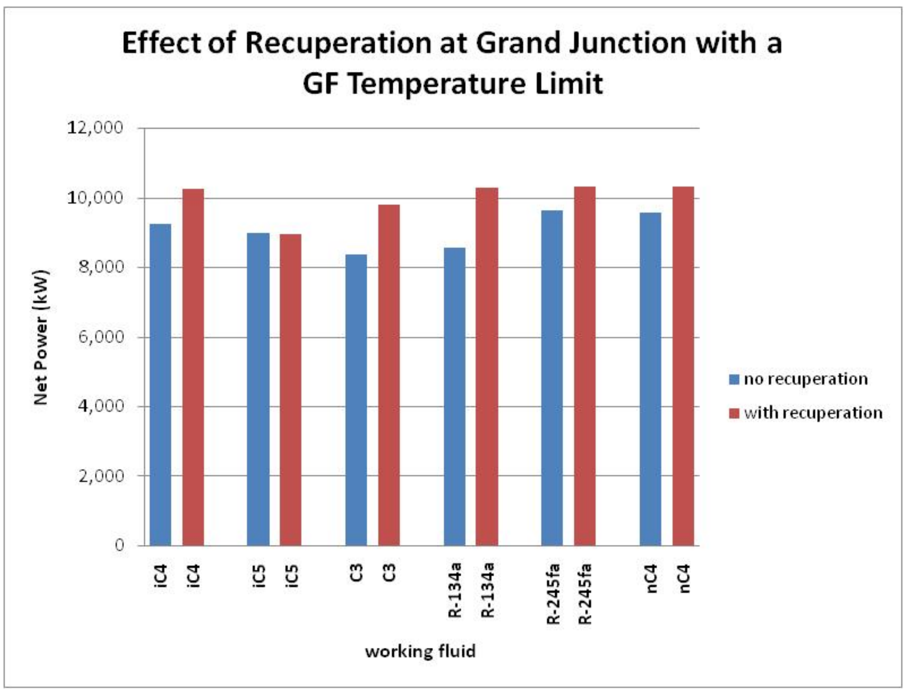

Figure 17. Recuperated plant performance for the Grand Junction scenario with an outlet temperature limit

When there is a limit on the geothermal outlet temperature, as is the case for the Grand Junction scenario, there a performance benefit from using recuperation. The magnitude of that benefit is shown in Figure 17. With the exception of isopentane, the performance of the optimized plant using recuperation at Grand Junction increased for the working fluids evaluated when a limit is place on the outlet temperature. The magnitude of the benefit varied, depending largely upon how much the imposition of the temperature limit affected plant performance. For the two fluids providing the higher levels of performance (R245fa and n-butane), recuperation increased power output by 7 to 8%. Note that these levels of performance with recuperation were still below the optimized performance of the plant when no limit is placed on the geothermal outlet temperature.

Recuperated Plant Design – Minimum Cost

The approach used to assess the cost of the recuperated plants is similar to that used for the baseline plant design and working fluid selection. Cost and performance of the plants are shown in the following tables for the recuperated and unrecuperated scenarios.

Table 9. Plant cost and performance for the Grand Junction Scenario

Fluid	No Recuperation, No Limit		No Recuperation; With Temperature Limit		With Recuperation; With Temperature Limit	
	Cost	Net Power (kW)	Cost	Net Power (kW)	Cost	Net Power (kW)
Isobutane	$36,500,427	10,647	$29,626,859	9,252	$32,939,520	10,254
Isopentane	$27,638,368	9,291	$6,036,885	9,005	$26,075,334	8,971
Propane	$35,440,611	10,003	$28,479,057	8,361	$32,420,617	9,816
R134a	$35,435,577	10,489	$27,650,213	8,575	$33,219,826	10,286
R245fa	$33,828,787	10,984	$29,198,291	9,653	$30,712,027	10,311
n-butane	$34,818,253	10,898	$29,581,785	9,567	$31,263,811	10,324

Table 10. Plant cost and performance for the Houston Scenario

Fluid	No Recuperation			With Recuperation		
	Cost	Net Power (kW)	$/kW	Cost	Net Power (kW)	$/kW
Isobutane	$14,299,489	3,430	$4,169	$13,947,535	3,356	$4,157
Isopentane	$12,321,017	3,016	$4,084	$11,599,136	2,874	$4035
Propane	$18,471,288	3,967	$4,657	$18,527,123	3,895	$4,756
R134a	$18,417,775	4,131	$4,458	$18,394,499	4,077	$4,512
R245fa	$12,917,357	3,196	$4,042	$12,431,899	3,096	$4,017

Again, the installed cost in $ per kW is indicative of the generation cost. The working fluids producing the minimum plant capital cost for Grand Junction are shown in Figure 18 both with and without recuperation.

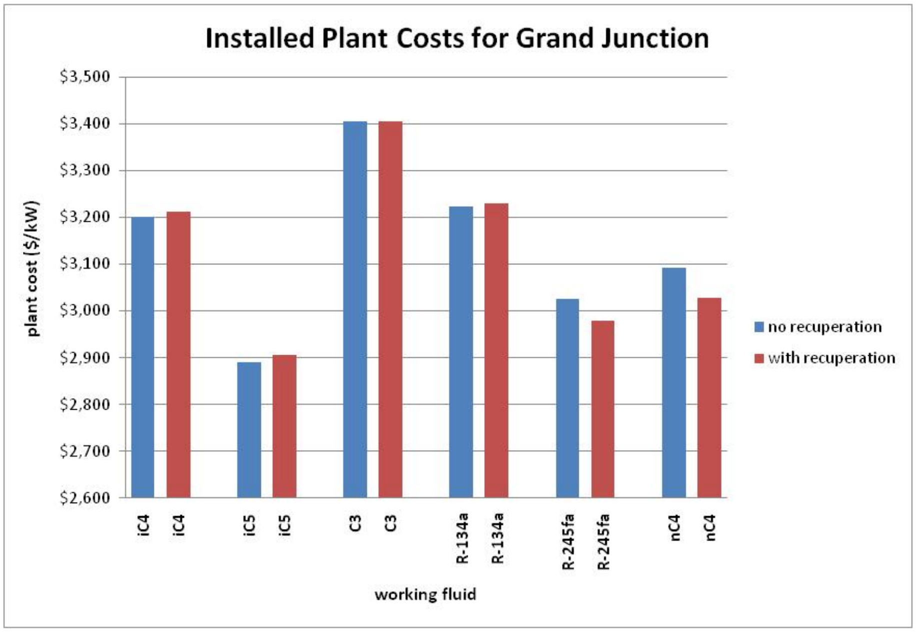

Figure 18. Recuperated plant cost for the Grand Junction scenario

As was the case for the base plant design, isopentane has the lower plant cost. Interestingly it is the only fluid whose cost with recuperation increased. Again, the important capital cost is the installed project cost that will also include the well field. These results are shown in Figure 19 for the project cost using the same assumptions for well field cost ($21 Million) that was used for the baseline plant evaluation.

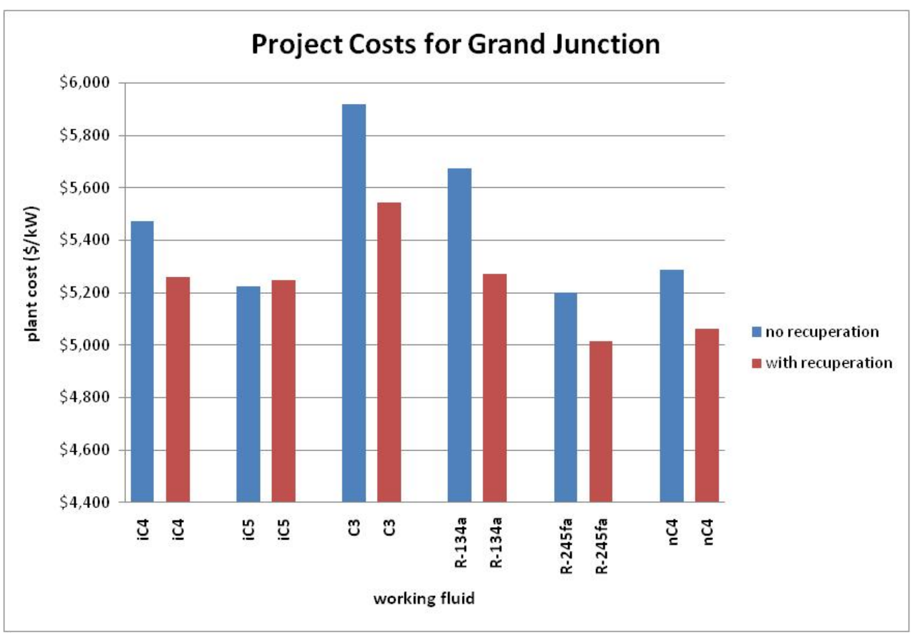

Figure 19. Project costs with recuperated plant for the Grand Junction scenario

For this magnitude of well field costs, the optimized plant with the R245fa working fluid would provide the minimum project capital costs and the minimum power generation cost. Again this is based upon the assumptions made relative to the well field costs. For this scenario, once the well field costs exceeded ~$18 Million, the unrecuperated plant using R245fa would have a cost advantage over the plant using isopentane; for the recuperated scenario, R245fa would have an advantage over isopentane once the well field costs exceeded ~$5 Million.

At the Houston location, recuperation generally increases the installed plant cost. The project costs for this location are shown in Figure 20 below with the EGS well field costs of $21 Million.

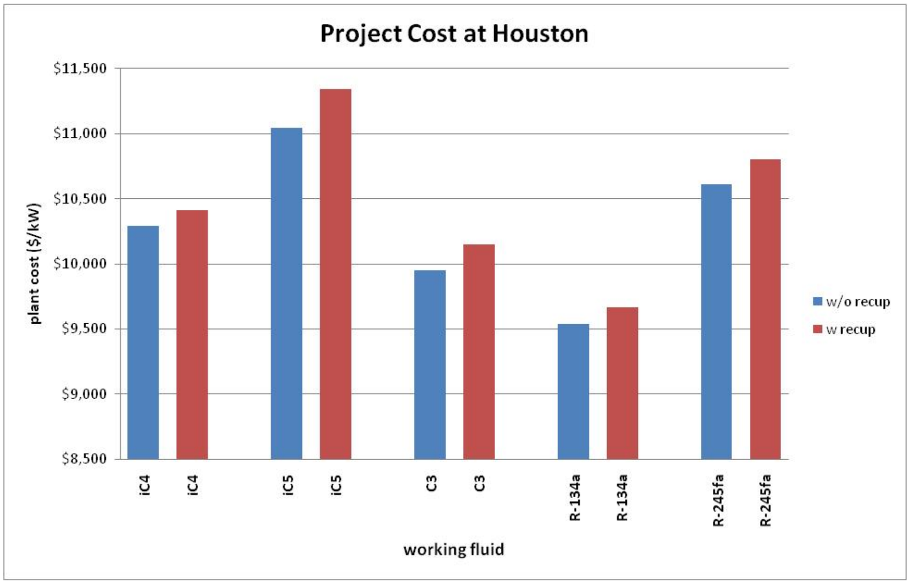

Figure 20. Project costs with recuperated plant for the Houston scenario

This shows that for this scenario and the assumptions used, recuperation provides no cost benefit. It also indicates that the optimized cycle with the R134a working fluid provides the minimum capital cost and power generation cost.

In conclusion, the assessment of recuperation indicated it provided no cost or performance benefit if no temperature limit is placed on the temperature of the geothermal fluid leaving the plant. If there is a limit placed on this temperature, recuperation can improve performance. At the design conditions for the Grand Junction location, recuperation increased plant output by ~7 to 8%. It also resulted in a slightly lower plant capital cost (~1.5%). When combined with a postulated EGS well field cost, it is estimated to lower the total project capital cost by ~3.5%.

Condenser Using EGS Make-up

Two configurations were considered for a condenser that utilized the EGS make-up water to augment heat rejection. The first configuration assumed the condenser using this makeup was in parallel with the air-cooled condenser; the second assumed the condensers were in series. The parallel configuration was ultimately selected because it was assumed that there would be a pressure drop associated with the make-up condenser. This pressure drop negated the ability to reduce the condensing temperature further with the series configuration.

The analysis was performed for the design scenarios at both geographic locations. For each location it was assumed that ground water would be used for make-up, and that this water would be at the average ambient temperature used for the design air temperature. The basis for the evaluation was the optimal plant design defined for each location with a temperature limit imposed on the geothermal fluid leaving the plant. In assessing the potential benefit of using the make-up condenser, the model was allowed to perform a similar optimization of process conditions under the same set of assumptions and using the optimal working fluid selected for the location's base plant design. The EGS make-up water flow rate was assumed to be 5% of the produced geothermal fluid flow, or 50,000 lb/hr (6.3 kg/s).

The results for each location's design scenario are shown in the table below.

Table 11. Costs and power generation for makeup water condenser scenarios

Location	Scenario	Fluid	Capital Cost	Net Power, kW	$/kW
Houston	No Recuperation, T-Limit	R134a	$19,235,291	4,294	$4,479
	No Recuperation, T-Limit, Make-up		$18,982,725	4,297	$4,417
Grand Junction	No Recuperation, No T-Limit	R245fa	$33,509,172	10,895	$3,075
	No Recuperation, No T-Limit, Make-up		$33,599,867	10,906	$3,081
	No Recuperation, T-Limit	R245fa	$29,192,482	9,621	$3,034
	No Recuperation, T-Limit, Make-up		$29,336,303	9,629	$3,046

These results indicate that using the make-up water to supplement the condensing provides a small increase in net power for scenarios considered at each location. For the lower temperature resource at the Houston location, there was a small decrease in capital cost, however with the higher temperature resource at Grand Junction, the capital costs increased slightly. A satisfactory explanation as to why the costs went up for one location and down for the other has not been found. It should be noted that for each of the scenarios in Table 11 the differences between the net power values calculated with and without the makeup water condenser fall within the convergence tolerance of the simulation optimizer, and that the model's optimization produces slightly different results each time the model is run. This may have contributed to the shift in the cost trend between locations.

These results indicate there is not a compelling reason to use to incorporate the make-up condenser at the design conditions for each location. The impact of using this condenser during 'off-design' operation is discussed in a subsequent section of this report.

Reheat Turbine

The potential for the use of turbine reheat to improve cycle performance was evaluated. When this concept is used, at an intermediate point in the turbine expansion, the vapor exits the turbine and is heated to a higher temperature using the geothermal fluid. This heated vapor is then directed back to the turbine and the expansion completed to the condenser pressure. This is illustrated in the example of a binary cycle with reheat shown in the figure below.

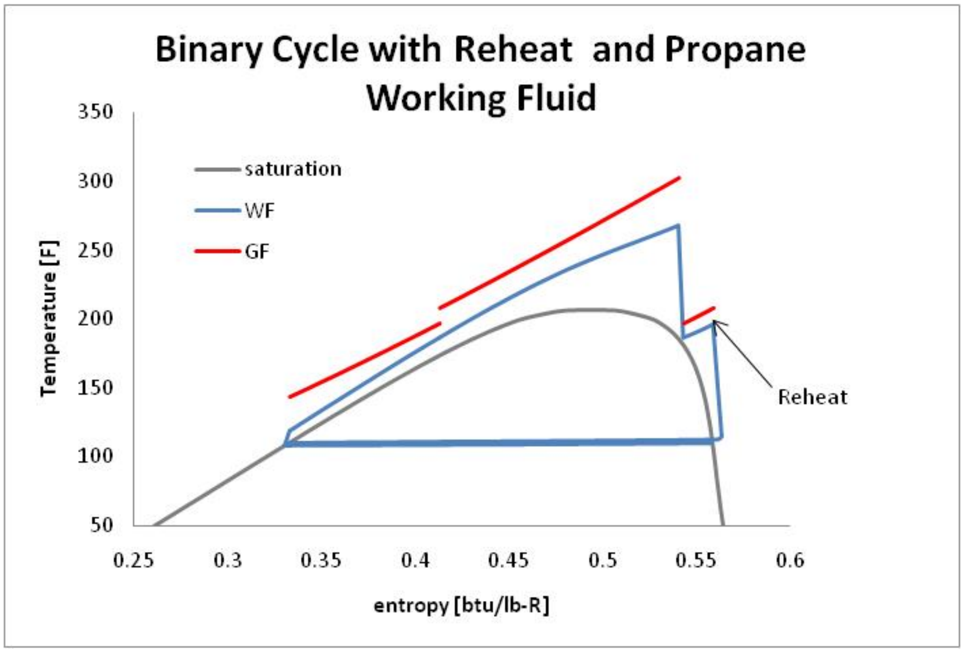

Figure 21. Temperature-entropy (T-s) diagram for propane binary cycle with reheat turbine

It was believed that this cycle could have a performance advantage when working fluids having vapors that tend to condense upon expansion are used; propane, R134a, and water are examples of these types of fluids. The analysis performed for the baseline plant suggested that these fluids would more likely be used with the lower temperature resources; hence the assessment of the potential benefit of this concept was done using the Houston location scenario. The working fluid selected for this evaluation was propane; it was selected primarily because for a given change in the dew point entropy it has a smaller change in the dew point temperature. It requires more superheating in the vapor entering the turbine in order to assure the expansion in a single stage turbine would remain 'dry".

In assessing the potential benefit, the same assumptions were used in defining the performance of the baseline plant. In addition, a pressure drop of 1 psi was assigned to the working fluid vapor side of the reheater. In lieu of separating a portion of the geothermal fluid flow to perform the reheating, the model assumed that all the geothermal flow passed through the reheater before entering the preheater. The temperature of the geothermal fluid entering the reheater was adjusted to assure that a 10°F approach temperature was achieved during the heat transfer process.

In order to assess the benefit of reheating, the base plant performance with propane was first optimized. The cycle with reheating was then evaluated by varying both the turbine inlet pressure and the intermediate pressure at which the reheating was performed. The model determined the temperature of the vapor entering the high pressure turbine stage (exiting the vaporizer) using the dew point entropy corresponding to the intermediate (reheat) pressure. The temperature of the vapor entering the second turbine stage was based on the dew point at the turbine exhaust pressure. For given turbine high and low (intermediate or reheat) pressures, the exhaust pressure was varied until a maximum net power was found. The working fluid flow rate and air flow rate were varied to achieve the desired pinch points in the heat addition and heat rejection processes. For this study, the model was not used to find an optimal low/intermediate/reheat pressure. Instead a parametric study was made to better assess those conditions which led to the optimal performance.

The expectation was that this cycle configuration would allow more working fluid to be vaporized and more power generated; this did occur. However, because the working fluid flow rate was higher and the optimal turbine pressures at the first stage were higher, the pumping power required was significantly higher. In addition, the cycle allowed more heat to be removed from the geothermal fluid, which increased the heat duty and fan power in the condenser. Consequently, for the design scenario evaluated a combination of first stage inlet pressure and intermediate/reheat pressure were not found that produced as much net power as the optimized base plant design using the propane working fluid. The optimal performance with reheating came within <1% of the baseline performance, but never exceed it.

The cycle analysis indicates a plant with reheating would require larger working fluid pumps (~45%), geothermal heat exchangers (~27%, exclusive of the reheater), turbine-generator (~10%) and a slightly larger air-cooled condenser (2%). Given there was no performance advantage, no detailed cost estimate was made for this configuration.

Though this preliminary assessment of reheating suggested no advantage for using this concept, there may be benefit with lower temperature resources.

Off-Design Performance

The modeling of the fixed plant configuration was used to assess the impact of the variations in both the ambient air temperature and the resource temperature on the performance of the optimal air-cooled binary plant design established for both locations. These analyses were made for the scenario where a constraint is placed the temperature leaving the plant, and examined plant designs that were unrecuperated, recuperated, and unrecuperated with condenser using the EGS make up water.

In assessing the impact of the air temperature on performance, the fixed plant output was predicted at the maximum and minimum hourly air temperatures at each location, as well as 6 intermediate temperatures (one of which was the design temperature). This was done at the design geothermal temperature, as well as for production scenarios where the fluid temperature had decreased by 5, 10, 15, 20 and 25°C from the initial design temperature. Plant output for the Grand Junction location is shown in the figure below at the design geothermal fluid temperature, as well as after a 10°C and 25°C decline in the production fluid temperature.

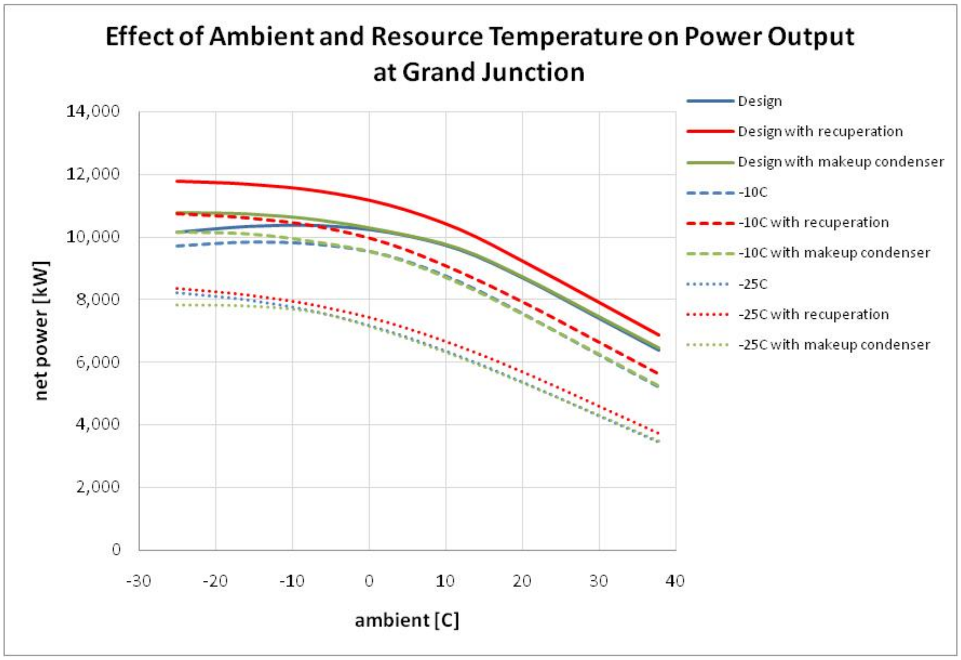

Figure 22. Effect of off-design performance on plant output for the Grand Junction scenario (R245fa working fluid)

These results indicate that recuperation provides more benefit at the lower ambient temperatures at the design resource temperature. However once the resource temperature begins to decline, the relative magnitude of this benefit decreases, and at the 25°C level of decline, there is little benefit at the lower air temperatures. In contrast, at the higher air temperatures, the relative benefit from using recuperation increases slightly as the resource temperature declines. The use of the makeup condenser provides some benefit at the design resource temperature when operating at the lower ambient temperatures. At the 25°C level of temperature decline, this advantage is negated. The makeup condenser provides a slight performance advantage at the higher ambient temperatures; like recuperation, this performance advantage remains relatively constant over this range of temperature decline.

Similar results are shown in the figure below for Houston. These results indicate that recuperation provides minimal benefit at off-design conditions. These results do indicate that using the makeup condenser when the ambient temperatures are low results in a performance penalty. This penalty occurs because the water temperature for Houston is always assumed to be the design ambient temperature (21.7°C). With the assumed pinch point in the condenser (5.6°C), the condenser temperature in the model is never allowed to drop below 27.3°C, which limits operation during the colder periods. It is believed that this effect also impacted the projected results for Grand Junction, however because of the temperature constraint on the geothermal fluid at this higher temperature, this impact was not as apparent. It is believed that when it is sufficiently cold that the makeup condenser will adversely impact performance. When this occurs, flow to the condenser would simply be shut off and operation proceed with only the air-cooled condenser; this operation was not simulated.

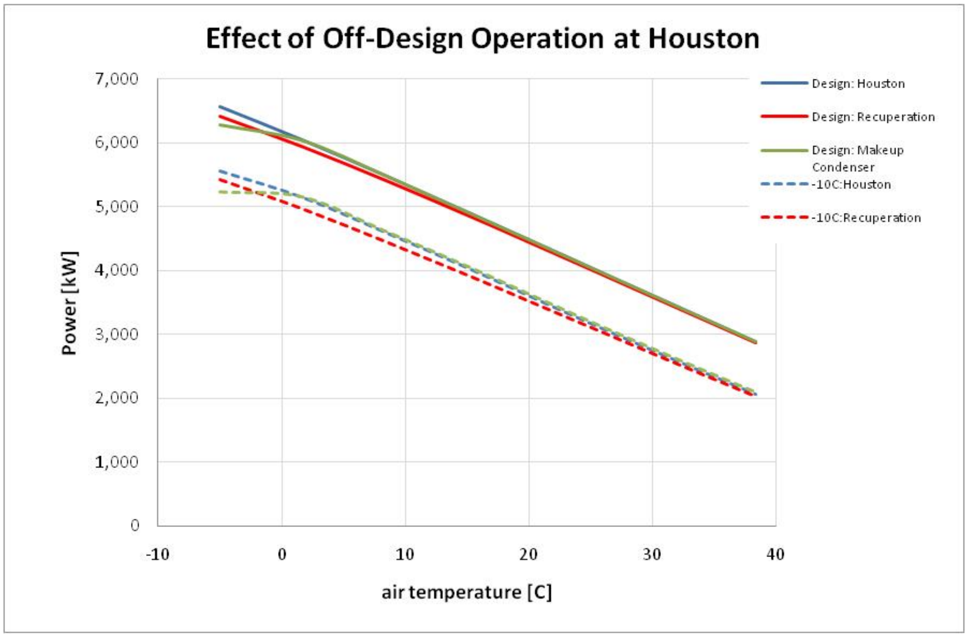

Figure 23. Effect of off-design performance on plant output for the Houston scenario (R134a working fluid)

It should be noted that it is not uncommon for the output from the plants to be limited by the operator during colder portions of the year. Typically the generators are sized for the design conditions, and are limited to the extent that they can be operated beyond that design generation capacity (10 to 20% are typical). If this upper limit of the generator capacity is reached, the operator will adjust the plant operation (generally by reducing air flow through the condenser) to not exceed this maximum limit. For the Grand Junction location and the design resource temperature, the maximum predicted generator output is ~13% more than at design. However for the Houston location the maximum predicted generator output is ~40% more than design, suggesting that at Houston it would be necessary to adjust operation to avoid operation beyond the generator capacity.

As an alternative, one might oversize the generator to allow maximum power to be produced during the colder portions of the year. At Houston, it is estimated that a generator sized to accommodate operation during the colder portions of the year would add an additional $100 per kW to the total plant cost, or ~$400,000.

The potential effects of both recuperation and the makeup condenser on output over the life of a plant are shown in the table below. This table shows the kW-hrs produced over a 30 year plant life for the plant using the Grand Junction design scenario at different rates of resource decline (it is assumed that the resource temperature declines at some annual percentage rate to achieve this end of life temperature). The results shown assume the plant operates continuously over the entire 30 year life. Though this is an unrealistic assumption, the results are indicative of the amount of additional power that might be generated. Also shown in this table is the ∆power (again in kW-hrs) that would result if either recuperation or the makeup condenser were used.

Table 12. Impact of concepts on total power produced over 30 yr life for Grand Junction design scenario

End of Life Temperature Decline	Base Plant: kW-hrs Produced over 30 yr life	Recuperation: ∆kW-hrs over 30 yr life	Make-up Condenser: ∆kW-hrs over 30 yr life
0°C	2,522,160,000	198,028,000	19,344,000

5°C	2,474,699,633	172,466,570	12,951,865
10°C	2,412,486,341	151,061,913	7,457,084
15°C	2,335,404,488	133,845,464	2,866,421
20°C	2,243,335,556	120,849,439	-813,199
25°C	2,136,158,040	112,106,856	-3,574,674

These results indicate that the performance advantage provided by the make-up condenser diminishes as the temperature declines, and at some point may adversely affect performance. These results also indicate the advantage from using recuperation will decline, though it does not appear to result in any adverse impact on performance for this scenario with the indicated levels of performance decline.

No similar evaluation was made for Houston as recuperation did not appear to provide any benefit at the design condition, and the benefit from the makeup condenser was marginal.

Conclusion

Though this study focused on specific design scenarios, both in terms of the ambient and resource temperatures as well as the assumptions made relative to component performance, some results would appear to be generic to all designs.

Probably most important is that if there is no temperature limit imposed on the geothermal fluid leaving the plant, it is difficult to produce any meaningful benefit from recuperation. If however this is a temperature constraint, recuperation appears to be a cost effective means of offsetting the adverse effects of this operational limit.

Turbine reheat may have the potential to improve performance for lower temperature resources, however for the resource conditions evaluated it was not possible to show any performance benefit for this concept relative to a power cycle that has already been optimized for the heat source and sink conditions. In addition, it appears that incorporating reheat into the cycle could increase capital costs beyond the added heat exchanger and turbine.

Using the EGS makeup water in a condenser that supplements the heat rejection in the air-cooled condenser provides only a slight performance benefit with the higher ambient temperatures. At the cooler temperatures it is probable that it would not be used as it could impose a lower temperature limit on the condensing temperature. The magnitude of this potential penalty will be dependent upon the temperature of the makeup water; this analysis assumed that it was the same as the annual median ambient temperature, and that value was kept constant for all ambient temperature scenarios. The magnitude of the benefit is small in part because the water flow rates are relatively small; the water cooled condenser accounted for ~1% of the total heat rejected for both design scenarios. While the benefit is small, the condenser is also small and should not be very expensive; the risk for incorporating the condenser into the plant operation should be small.

The results presented have assumed that turbines with variable nozzle geometry are used. If they are not used, then it is very improbable that the off-design output that was projected will be realized.

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