October 1983

EFFECTS OF VAPORIZER AND EVAPORATIVE-CONDENSER SIZE
ON GEOFLUID EFFECTIVENESS AND COST OF ELECTRICITY
FOR GEOTHERMAL BINARY POWER PLANTS

EGG-GTH--6376

DE84 002386

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Operated by the U.S. Department of Energy

This is an informal report intended for use as a preliminary or working document

Prepared for the
U. S. DEPARTMENT OF ENERGY
Under DOE Contract No. DE-AC07-76ID01570

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O. J. Demuth

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ABSTRACT

A special study was conducted in support of the DOE/DGHT Conversion Technology Program to investigate the influences of minimum approach temperature differences occurring in supercritical-heater/vaporizer and evaporative-condenser heat rejection systems on geothermal-electric binary power plant performance and cost of electricity. For the systems investigated optimum pinch points for minimizing cost of electricity were estimated to range from 5 to 7°F for the heater vaporizer. The minimum approach of condensing temperature to wet-bulb temperature for evaporative condensers was estimated to be about 30°F in order to achieve the lowest cost of electricity.

Preface

This report presents the results of a special study performed under the direction of the Department of Energy Conversion Technology Program Manager, Ray LaSala. The study was made to improve the quality of advanced geothermal plant cycle analyses and conceptual systems evaluations, which are conducted as part of the mainstream DOE/DGHT Conversion Technology Program effort.

The work was supported by the U. S. Department of Energy, Assistant Secretary for Resource Application, Office of Geothermal, under DOE Contract No. DE-AC07-76ID01570.

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I. SUMMARY

1. Background

In the conduct of scoping-type performance and conceptual systems analyses for advanced geothermal plants, the preselection of near-optimum approach temperature differences in the heat addition and rejection systems can reduce the analysis effort by a major extent. The special study described in this report was conducted to complement the primary effort of the DOE/DGHT Conversion Technology Program by providing some general insight relative to near-optimum magnitudes of these approach temperature differences. The major objective of the study was to determine approximate values of the minimum approach temperature differences (pinch points) for supercritical heater/vaporizer and evaporative-condenser heat rejection systems which will result in maximum geofluid effectiveness, and separately, minimum cost of electricity.

2. Approach

To accomplish the objectives, cycle performance and capital cost estimates were made for a number of plants employing supercritical cycles with isobutane (iC_4) and 0.96 isobutane/0.04 heptane (C_7) working fluids, and evaporative-condenser heat rejection systems. Geothermal fluid at a temperature of 360°F (slightly subcooled) was assumed to be provided to the plants. For the analyses, sizes of the heater-vaporizer and evaporative-condenser systems were varied, and the effects on cost of electricity determined. A 160°F geofluid-outlet-temperature constraint (to prevent silica precipitation) was assumed for the evaporative condenser investigations; for the heater-vaporizer investigations cases both with and without the geofluid outlet constraint were analysed.

3. Conclusions

The results of the study and the conclusions reached are summarized below:

1. For the unrecuperated $0.96\text{ iC}_4/0.04\text{ C}_7$ plants without an outlet geofluid temperature constraint, the minimum cost of electricity was predicted for a heater/vaporizer pinch point temperature difference of about 5°F .
2. For the $0.96\text{ iC}_4/0.04\text{ C}_7$ plants having a 160°F minimum-temperature constraint on the outlet geofluid and using turbine-exhaust recuperators, the minimum cost of electricity was predicted for a heater/vaporizer pinch point of approximately 7°F ; the required recuperator pinch point for this condition was about 4°F .
3. Use of a turbine exhaust recuperator with the plants having an outlet geofluid temperature constraint of 160°F was predicted to result in a cost of electricity essentially equivalent (about 1% lower) to that produced by the unrecuperated plants without the temperature constraint.
4. For the recuperated, $0.96\text{ iC}_4/0.04\text{ C}_7$ plants using evaporative-condenser heat rejection and having the geofluid-outlet-temperature constraint, maximum geofluid effectiveness was predicted when the approach of the condensing temperature to wet bulb temperature was approximately 15°F . For those same plants, minimum cost of electricity was predicted for a corresponding approach temperature difference of about 30°F .
5. Similar trends, relative to the the approach of condensing temperature to wet bulb temperature, were found for the outlet-temperature-constrained isobutane plants using evaporative-condenser heat rejection. The maximum geofluid effectiveness predicted for the isobutane plants was about 10% less than that predicted for the recuperated (and outlet-temperature constrained) $0.96\text{ iC}_4/0.04\text{ C}_7$ plants; the minimum cost of electricity for the isobutane plants was predicted to be 6% higher than for the $0.96\text{ iC}_4/0.04\text{ C}_7$ plants.

6. Market-penetration studies, based on comprehensive investment-decision models, have been applied to compare the consequences of selecting a 15°F pinch point temperature difference for the heater/vaporizer in a 0.96 iC₄/0.04 C₇ plant, rather than a 5°F pinch point, which was predicted to be near optimum. The resulting estimates indicated that in a "free market", the selection of the 15°F pinch point could result in a reduction of about 35% in geothermal resource utilization (production of electrical power) in the year 2000 from geothermal resources in the 350 to 400°F temperature range. Similar evaluations indicated that selection of an evaporative condenser producing an approach of the condensing temperature to wet bulb temperature of 45°F, rather than a near-optimum value of 30°F, could result in a corresponding reduction in resource utilization of about 26%.

II. INTRODUCTION

In the optimization of binary geothermal power plants, tradeoffs are made of the sizes and costs of the working-fluid-vaporization and heat-rejection systems, against net plant geofluid effectiveness (watt-hr/lbm geofluid) and cost of electricity (mills/kWhr). For example, as the size of the working-fluid vaporizer increases, its cost increases, but the mean temperature difference between the fluids involved in the heat exchange process decreases, resulting in a corresponding decrease in thermodynamic irreversibility and an increase in net plant geofluid effectiveness. At some point the increase in component cost becomes equally as influential on cost of electricity as does the increase in geofluid effectiveness, and an optimum component size is reached relative to cost of electricity. Similar relationships exist between heat-rejection system size and cost of electricity.

The goals of the Conversion Technology Program require that thermodynamic cycle analyses and conceptual designs, of a preliminary nature, be conducted to evaluate advanced binary-cycle components and/or plant configurations and working fluids. These studies generally involve perturbing one feature or component at a time. An example of such a study might consist of comparing working fluids used in single-heating supercritical cycles utilizing a given geothermal resource. In principle, the entire plant should be optimized for each fluid before the comparison of fluids can be complete; for the simplest plants vaporizer pressure, vaporizer size, and heat-rejection system size would then be optimized simultaneously. If magnitudes of pinch points (minimum approach temperature difference) between the geofluid and working fluid for the vaporizer, and between the working fluid and ambient wet bulb temperature for the heat-rejection system can be "fixed", the three-parameter plant optimization for each working fluid can be reduced to optimization of a single parameter (vaporizer pressure). The significance of such simplification is increased for studies involving more complex cycles containing features such as multiple boiling, regenerative preheating, or turbine bleed.

The major objectives of this special study were: to determine approximate magnitudes of (1) the working-fluid vaporizer pinch point, and (2) the minimum approach of condensing temperature to wet-bulb temperature for evaporative condensers, which result in maximum net geofluid effectiveness, and separately, in minimum cost of electricity. The results are intended to provide guidance for the selection of suitable approach temperature differences for heat-addition and heat rejection components which can be used for scoping-type advanced plant systems and conceptual-design analyses.

III. APPROACH AND ASSUMPTIONS

1. General

The special study presented in this report was conducted in order to establish near optimum values of minimum approach temperature differences for heat addition and rejection systems for use in systems analyses or conceptual designs of advanced binary geothermal plants. The study was made assuming a liquid-dominated hydrothermal resource having a temperature at the plant inlet of 360°F. Moderately advanced plant features were included in the study such as a mixed-hydrocarbon working fluid (0.96 isobutane/0.04 heptane) vaporized at a supercritical pressure, a turbine-exhaust recuperator for regenerative working fluid preheating, and an evaporative condenser heat-rejection system. Power plant state points were selected from previous analyses (Reference 1) corresponding to near optimum conditions of working-fluid composition and turbine inlet pressure. Turbine expansion processes were selected to avoid the moisture region, and piping and component pressure drops, as well as heat losses to the surroundings, were not included.

For the determination of effects of heater-vaporizer pinch point magnitudes, the basic effort consisted of first conducting a number of thermodynamic cycle analyses for a mixed hydrocarbon working fluid in which the minimum approach temperature difference (pinch point) for the working-fluid vaporizer was varied, and the remainder of each of the plants configured and sized consistently. Next, a "value analysis" (Reference 2) was conducted to estimate the effect of vaporizer pinch point on cost of electricity which accounts for the influences of pinch point on both geofluid effectiveness and capital cost of the various major plant components.

The same general approach was used in the determination of the effect of evaporative condenser size and approach to wet bulb temperature on plant performance and cost of electricity. This second part of the study was conducted for both isobutane (iC_4) and the mixed hydrocarbon working fluid (0.96 iC_4 /0.04 heptane (C_7)).

2. Cycle Analyses

Thermodynamic cycle analyses were conducted to calculate $\frac{\Delta \text{kWh}}{\text{kWh}}$, which was defined as the fractional increment in electrical energy produced annually for a given plant, relative to a reference plant which used the same geofluid flow. The analyses included determination of turbine power, working-fluid pumping parasitics, and the parasitic losses introduced by the evaporative-condenser heat-rejection system. (The evaporative-condenser parasitic losses were obtained from manufacturers' information as functions of the approach of condensing temperature to wet-bulb temperature for both isobutane and 0.96 isobutane/0.04 heptane working fluids; the losses and costs as used are shown in Table 6.) The ambient wet bulb temperature was held constant at 60°F for the calculations; the effects of varying wet-bulb temperature during the year (daily and seasonal) were assumed to be of secondary importance in the comparisons. The reference plant selected is sketched schematically in Figure 1, and its characteristics are shown in Table 1 below:

TABLE 1. CHARACTERISTICS OF THE REFERENCE PLANT

Working Fluid	Isobutane
Turbine Inlet Pressure (heater pressure)	1000 psia*
Geofluid Outlet Temperature	160°F*
Condensing Temperature (evaporative condenser)	105°F
No Regenerative Preheating	
Plant Net Power Output = 47MW at 5.75×10^6 lbm/hr geofluid	

*The 160°F geofluid outlet temperature was selected to avoid silica precipitation. The turbine inlet pressure of 1000 psia was found to result in an 11°F heater pinch point at the 160°F geofluid outlet temperature.

Figure 1: Simple Binary Geothermal Cycle
with Evaporative Condenser

2.1 Effects of Heater Pinch Point. For the part of the study investigating effects of heater size and pinch point, cycle analyses were conducted for several values of heater pinch point for two types of plants, both using a 0.96 isobutane/0.04 heptane (by mass) working fluid. Heater pressure was held at 600 psia (found to be near optimum in Reference 1), and condensing bubble-point temperature was held at 90°F (found to be near optimum in the second part of the present study). Geofluid flow was held fixed at the value used for the reference isobutane plant.

For the first type of plant the geofluid outlet temperature was unconstrained, and regenerative working fluid preheating was not utilized. Values of working fluid flow, turbine power, working fluid pumping parasitic power and evaporative condenser parasitic losses in the form of Whr/lbm geofluid were determined for each heater pinch point. The schematic of the reference plant (Figure 1) applies to this plant as well.

For the second type of plant the outlet geofluid temperature was constrained at 160°F to correspond to conditions where silica precipitation constitutes a potential problem, but is avoided by imposing the temperature constraint. For this plant type, turbine exhaust recuperators were assumed to be utilized for regenerative preheating of the working fluid; a schematic sketch is shown in Figure 2. Cycle calculations conducted for these plants were similar to those made for the first plant type. For each pinch point selected for the heater/vaporizer, a corresponding pinch point and size was found for the turbine exhaust recuperator; in addition, the turbine power and the various parasitic losses were evaluated. Heater pressure and condensing bubble-point temperature were held at 600 psia and 90°F, as was done for the first plant type.

2.2 Effects of Approach to Wet Bulb Temperature. For the second part of the study, which evaluated the effects of evaporative condenser size and approach to wet-bulb temperature, cycle analyses were

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Figure 2: Binary Geothermal Cycle with Turbine Exhaust Recuperator

conducted in a similar fashion to those evaluating effects of heater pinch point. For these analyses the plant outlet geofluid temperature was constrained at 160°F, and again, two types of plants were examined. First, simple isobutane plants without regenerative preheating, as shown in Figure 1, were analysed. For these, the heater pressure and geofluid flow were held at 1000 psia and 5.75×10^6 lbm/hr as was done for the reference plant.

The second type of plant studied employed the 0.96 isobutane/0.04 heptane mixture and a turbine exhaust recuperator. As for the heater pinch point studies, the heater pressure was held at 600 psia. The schematic diagram shown in Figure 2 represents these cycles. For both types of plants several condensing (bubble point) temperatures were selected, and cycle analyses were conducted as discussed for the heater-pinch-point study; these analyses resulted in values of net plant power (geofluid effectiveness), working fluid flows, heater size and recuperator (if used) size. Table 2 summarizes the plant types considered for both parts of the study.

TABLE 2. TYPES OF PLANTS INVESTIGATED

Component Investigated	Working Fluid	Heater Pressure, psia	Condensing/ Bubble Point Temperature, °F	Outlet Geofluid Temperature, °F	Turbine Exhaust Recuperator
(Reference Plant)	Isobutane	1000	105	160	No
Heater/Vaporizer	0.96 Isobutane/ 0.04 Heptane	600	90	Not Constrained	No
Heater/Vaporizer	0.96 Isobutane/ 0.04 Heptane	600	90	160	Yes
Evaporative Condenser	Isobutane	1000	75-105	160	No
Evaporative Condenser	0.96 Isobutane/ 0.04 Heptane	600	70-105	160	Yes

3. Value Analyses

Since optimum component sizes and configurations, relative to cost of electricity, are not the same as those optima relative to geofluid effectiveness value analyses were conducted to determine how each magnitude of minimum approach-temperature difference (pinch point) selected for the cycle analyses influenced the cost of electricity. The general approach described in Reference 2 was followed.

3.1 Evaluation Method. The basic approach involved estimating differences in performance as well as differences in capital and operating costs between a reference geothermal plant and modified plants for which cycle analyses had been conducted. (The reference plant is specified in Table 1.) These differences in cost and performance, all of which influence the real value of the "improvement", were then related to the common denominator, cost of power. To apply the method, the cost of electrical power for the reference plant was subdivided into a number of fractional contributions which included the specific areas influenced by the plant modifications. The fractional changes in cost for each of these areas (for which modifications were required) were estimated, along with the change in electrical energy produced by the modified plant. If the modified design required cost changes for n contributions to the cost of electricity, and resulted in a fractional change in the annual energy produced, the fractional change in cost of electricity, relative to the reference design, was calculated as:

$$\frac{\Delta \text{COE}}{\text{COE}} = \frac{\frac{-\Delta \text{kWh}}{\text{kWh}} + \sum_{1}^n \frac{\text{COE}_n}{\text{COE}} \left(\frac{\Delta C}{C} \right)_n}{1 + \frac{\Delta \text{kWh}}{\text{kWh}}} \quad (1)$$

Where: $\frac{\text{COE}_n}{\text{COE}}$ is the fractional contribution to cost of electricity (for the reference plant) of the n 'th contribution for which a fractional change in cost, $\left(\frac{\Delta C}{C} \right)_n$, is required for the modified

design, and $\frac{\Delta \text{kWh}}{\text{kWh}}$ is the fractional change in electrical energy produced annually.

To use the above expression a cost breakdown of a "reference" plant was required. Several sources for the capital cost distributions for similar plants were used to arrive at such a breakdown: the actual cost of the Raft River plant and equipment, loan guarantee proposal and review evaluations, the Heber binary plant estimates, and Holt-Procon studies. The major components of power cost are: plant capital, fuel, and operation and maintenance costs. A composite of information available from those sources yielded very similar results when proportioning the busbar power cost. These results are summarized in Table 3 for a binary cycle operating from a hydrothermal resource of about 360°F.

TABLE 3. BUSBAR POWER COST DISTRIBUTION

Item	Percent of Total Power Costs
Geothermal Fuel	50
Plant Capital	42*
Plant Operations and Maintenance (O&M)	8**

*Transmission charges are not included.

**Make up cooling water costs were estimated to correspond to 2 of the 8% O&M cost contribution.

Table 4 shows a breakdown of the capital costs by systems, major component, and labor that was derived from the various estimates, and used for the reference plant. From this breakdown it is seen that 50% of the plant costs are directly tied to the major systems; the balance is about equally divided between the

various other required items (fire, electrical, instrumentation, flare, holding ponds, control and maintenance building, etc.) and indirect charges (administration, conceptual engineering and reviews, preliminary and detailed design, construction management, inspection, etc.).

TABLE 4. PLANT CAPITAL COST DISTRIBUTION

System		Component	
Working Fluid	0.20	Heaters/boilers	0.09
		Pumps	0.01
		Balance (piping, etc.)	0.07
		Labor	0.03
Turbine-Generator	0.15	Turbine	0.035
		Generator	0.035
		Piping, Structure, etc.	0.06
		Labor	0.02
Heat-Rejection	0.15	Condenser	0.06
		Cooling Tower (wet tower)	0.025
		Pumps	0.01
		Pipes, Valves, etc.	0.015
		Water Treatment Equipment	0.025
		Labor	0.015
Geothermal	0.03	Boost Pump, Valves, Pipes	0.02
		Labor	0.01
Miscellaneous	0.21	Instrument & Control System, Fire Control, etc.	0.13
		Labor	0.08
Indirect General Administration, Engineering, Title I, II, III, Construction Mgmt.	0.26	Labor	0.26
TOTAL	1.00	TOTAL	1.00

Some of the capital costs from the sources mentioned required adjustment before constructing Table 4. For example, Admiralty Brass used as tubing in the Raft River geothermal heat exchangers is rather unique to Raft River because of the low hydrogen sulfide content of the water; other plants used carbon steel heat exchangers. Where such variances from general practice were found, the estimates have been modified. For the present study the cost of the evaporative condenser for the reference plant was assumed to correspond to the same fraction of the total capital cost as the heat-rejection system cost (0.15) shown in Table 4.

The specific procedure followed for both the heater and the evaporative-condenser value-analysis studies was, first, to estimate the fractional changes (relative to the reference plant) in component sizes and costs required for each of the plants for the different heater (or condenser) pinch points selected during the cycle analyses. Next, the fractional change in annual electrical energy generated, $\frac{\Delta \text{kWh}}{\text{kWh}}$, was taken from the appropriate cycle analysis, and the various values of $(\frac{\Delta C}{C})_n$ for the affected components found using cost distributions from Tables 3 and 4. For these studies the geofluid flow and cost were assumed to be fixed and equal to the values for the reference plant, so net plant power output varied slightly from plant to plant; differences between the plant outputs and 47MW (reference plant output) were assumed not to influence the cost of power in mills/kWh. The final step for each calculation of the fractional change in cost-of-electricity, $\frac{\Delta \text{COE}}{\text{COE}}$, relative to the reference plant, was to substitute the proper numerical values into Equation (1).

- 3.2 Evaluation Assumptions. The "reference" plant for each comparison was assumed to provide electricity with contributors to its cost subdivided as presented in Tables 3 and 4. The fractional changes in cost for the various cost contributors were estimated based on the approaches and assumptions discussed in subsequent paragraphs.

Heater. The major factor driving the cost of the heater, for a modification which does not require a change in tubing material, is the heat transfer surface area. The product of heat transfer surface area and overall heat transfer coefficient for the modified plant (UA_{mod}) was estimated relative to the reference case (UA_{ref}) using the relationship:

$$\frac{UA_{mod}}{UA_{ref}} = \frac{Q_{mod}}{Q_{ref}} \frac{\Delta T_{ref}}{\Delta T_{mod}}$$

where: Q_{mod} and Q_{ref} are the rates of heat transferred (Btu/hr) in the modified and reference heaters;

ΔT_{mod} and ΔT_{ref} are the log mean temperature differences for the two heaters ($^{\circ}F$).

Values of Q and ΔT were determined from thermodynamic cycle analyses for the modified and reference plants. To obtain more realistic approximations for the values of UA , separate contributions were determined corresponding to regions of the heaters at temperatures below and above the location of the minimum approach temperature (pinch point). The separate contributions to UA were combined to determine the total UA values for the modified and reference heaters.

For the cases considered, the heating processes were supercritical, maintaining single-phase statepoints. Accordingly, overall heat transfer coefficients for the modified and reference heaters were assumed to be equal. The surface area and cost of the heaters were, therefore, taken as being proportional to UA . The fractional change in cost, relative to the reference heater, $\frac{\Delta C}{C}$, was:

$$\frac{\Delta C}{C} = \frac{UA_{mod}}{UA_{ref}} - 1$$

The modified heater pressure (shell-side) for the cases in which the working fluid was 0.96 isobutane/0.04 heptane was 600 psia as

compared to 1000 psia for the reference heater. For those cases, a 21% correction to the cost (Reference 3, p. 84) was applied.

The contribution to cost of electricity $\left(\frac{COE_n}{COE}\right)$ for the reference-plant heater, used in Equation (1), was estimated from Tables 3 and 4 to be $0.42 \times 0.09 = 0.038$.

Evaporative Condenser. Evaporative condenser parasitic loss, cost, and makeup water requirements were obtained from condenser manufacturers as functions of the approach of the condensing temperature to the wet-bulb temperature (held constant for the study) of 60°F. The information was obtained for both isobutane and 0.96 isobutane/0.04 heptane based on condensing enthalpy changes provided to the manufacturers as shown in Table 5 below. All cases requested were for a wet-bulb temperature of 60°F, and for working fluid side pressure drops less than 2 psia.

TABLE 5. INFORMATION PROVIDED TO
EVAPORATIVE-CONDENSER MANUFACTURERS

Working Fluid	iC ₄	iC ₄	iC ₄	^(a) 0.96 iC ₄ / 0.04 C ₇	^(a) 0.96 iC ₄ / 0.04 C ₇	^(a) 0.96 iC ₄ / .04 C ₇
Condensing (Bubble Point) Temp. °F	105	90	75	105	90	75
Bubble Point Enthalpy Btu/lbm	-782.8	-791.5	-800.0	-776.1	-785.1	-793.6
Dew Point Temp. °F	105	90	75	>127	>112	>98
Dew Point Enthalpy Btu/lbm	-648.4	-653.1	-658.0	>-637.8	>-642.4	>-648.4
Hydrocarbon Inlet Temp. °F	135	124	113	127 ^(b)	112 ^(b)	98 ^(b)
Inlet Enthalpy Btu/lbm	-634.6	-638.0	-641.5	-637.8 ^(b)	-642.4 ^(b)	-648.4 ^(b)

(a) The condensing heat transfer coefficient for the mixture was estimated to equal 0.6 of that for isobutane.

(b) For the mixed hydrocarbon working fluid, (0.96 isobutane/0.04 heptane) desuperheating and some condensing was assumed to be accomplished in a turbine-exhaust-recuperator.

Table 6, below, summarizes the size, cost, parasitic loss, and makeup water requirements assembled from the manufacturers' responses to our inquiry. The magnitudes of the fractional changes in capital cost relative to the reference plant $\left(\frac{\Delta C}{C}\right)$, for the evaporative condensers, were found from values in Table 6.

As previously mentioned the evaporative condenser in the reference plant was assumed to correspond to the heat-rejection-system capital cost fraction of 0.15 from Table 4. The magnitude of $\frac{COE_n}{COE}$ for the reference plant evaporative condenser was taken as $0.15 \times 0.42 = 0.063$.

TABLE 6. EVAPORATIVE CONDENSER COST AND PERFORMANCE DATA

Working Fluid	Condensing Temp °F	Surface Area ft ² hr/Btu	Cost \$/ft ²	Parasitic Losses Whr/Btu	Makeup Water lbm/Btu
iC ₄	105	5.52 x 10 ⁻⁴	7.58	0.00292	0.00125
iC ₄	90	8.97 x 10 ⁻⁴	↓	0.00475	
iC ₄	75	19.45 x 10 ⁻⁴		0.01031	
0.96iC ₄ /0.04C ₇	105	5.81 x 10 ⁻⁴		0.00307	
0.96iC ₄ /0.04C ₇	90	8.41 x 10 ⁻⁴		0.00444	
0.96iC ₄ /0.04C ₇	75	14.83 x 10 ⁻⁴		0.00784	
0.96iC ₄ /0.04C ₇	70	20.52 x 10 ⁻⁴		0.01084	↓

Notes: (1) Wet Bulb Temperature = 60°F.

(2) Costs in table are based on hot-dip galvanized steel.

(3) Makeup water estimated for five "cycles of concentration".

Recuperator. The turbine exhaust recuperator provides some preheating of the working fluid using energy extracted from the turbine exhaust flow. This unit was not included in the reference plant but was included in some of the plants employing the mixed hydrocarbon working fluid. For the cases considered, desuperheating plus some condensation of the turbine-exhaust vapor was accomplished in the recuperator.

For the 600 psia mixed hydrocarbon cycle having a condensing temperature of 105°F, the recuperator was found to have a pinch point of 9°F. Previous estimates (Reference 2) indicated that the cost of such a recuperator should be about 28% of the cost of a conventional surface (shell and tube) water-cooled condenser; the labor for installing the recuperator was assumed to be 30% of that for installing a conventional heat rejection system. Using those estimates and the appropriate component cost fractions from Table 4, the fractional increment in capital cost resulting from use of the recuperator for the 105°F condensing temperature was: $\frac{\Delta C_{cap}}{C_{cap}} = 0.28 \times 0.06 + 0.30 \times 0.015 = 0.021$. For each of the other recuperators, capital cost was assumed to be proportional to the UA product for the recuperator, and the labor required for installation was assumed fixed. From Table 3 the value of $\frac{COE_n}{COE} \left(\frac{\Delta C}{C} \right)_n$ for each of the recuperators was estimated to be $0.42 \times \frac{\Delta C_{cap}}{C_{cap}}$.

Turbine. For these estimates the turbine cost was assumed to be proportional to the 3/2 power of the turbine exhaust area (a simplification of Equation 63, Reference 3). The turbine exhaust area was assumed to be proportional to the working fluid flow times the turbine exhaust specific volume times a correction factor which assumes that the turbine exhaust velocity is a constant fraction of sonic velocity (exhaust dynamic pressure, $\frac{1}{2} \rho_{ex} u_{ex}^2$, is a constant fraction of exhaust stagnation pressure). The fractional change in turbine cost was estimated as:

$$\frac{\Delta C}{C} = \left(\frac{A_{ex \text{ mod}}}{A_{ex}} \right)^{1.5} - 1$$

where $A_{ex \text{ mod}}$ and A_{ex} are the turbine exhaust areas for the modified and reference turbines. $\frac{CO_{En}}{COE}$ for the turbine was found from Tables 3 and 4 to equal $0.035 \times 0.42 = 0.015$ (for the reference plant).

Turbine Piping. The turbine piping cost was assumed to be directly proportional to the turbine exhaust area. Then for turbine piping:

$$\frac{\Delta C}{C} = \frac{A_{ex \text{ mod}}}{A_{ex}} - 1.$$

Turbine piping was assumed to represent 2% of the plant capital cost, and $\frac{CO_{En}}{COE}$ for the turbine piping in the reference plant was taken as $0.02 \times 0.42 = 0.0084$.

Generator. The cost of the generator and turbine-generator supporting structure was approximated as being proportional to the 0.7 power of the maximum power produced by the plant (Reference 3, p. 88). For the generator:

$$\frac{\Delta C}{C} = \left(\frac{MW_{mod}}{MW_{ref}} \right)^{0.7} - 1$$

where MW_{mod} and MW_{ref} are the turbine outputs for the modified and reference plants. The turbine generator supporting structure was assumed to represent 4% of the plant capital cost, and the resulting $\frac{CO_{En}}{COE}$ for this category was estimated as $(.04 + .035) \times 0.42 = 0.032$.

Miscellaneous Working Fluid Components. These components represent the working fluid pumps, piping, valves, etc. Their total cost was assumed to be proportional to the working fluid flow rate W_{WF} . The change in cost for this category is:

$$\frac{\Delta C}{C} = \frac{W_{WF \text{ mod}}}{W_{WF \text{ ref}}} - 1.$$

$\frac{CO_{En}}{COE}$ was estimated as $0.08 \times 0.42 = 0.034$.

Cooling Water Makeup. Costs for providing makeup cooling water were assumed to be proportional to the heat rejected. The change in cost of makeup cooling water was estimated as:

$$\frac{\Delta C}{C} = \frac{\text{Heat Rejected}_{\text{mod}}}{\text{Heat Rejected}_{\text{ref}}} - 1$$

As indicated in Table 3 the cost of providing makeup cooling water for the reference plant was assumed to contribute 25% of the operating and maintenance costs, or 2% of the total cost of electricity; $\frac{\text{COEn}}{\text{COE}} = 0.02$ for the reference plant.

Indirect Capital. This category pertains to the indirect costs of plant construction for general administration, engineering, Title I, II, III, construction management, etc. Table 4 indicates that the indirect capital costs for the basic plant correspond to about 35% of the direct capital costs. For the modified designs the increment in indirect capital was assumed to be 10% of the increment in direct capital costs rather than the full 35%. The increase in indirect capital costs was estimated as:

$$\frac{\Delta C_{\text{indirect}}}{C_{\text{cap}}} = \frac{\Delta C_{\text{direct}}}{C_{\text{cap}}} \times 0.10$$

where: $\Delta C_{\text{indirect}}$ and ΔC_{direct} are the increases in indirect and direct capital costs associated with the modification, and C_{cap} is the total capital cost of the reference plant.

$\frac{\text{COEn}}{\text{COE}} \left(\frac{\Delta C}{C} \right)_n$ for indirect capital then becomes:

$$\frac{\Delta C_{\text{direct}}}{C_{\text{cap}}} \times 0.1 \times 0.42 = \frac{\Delta C_{\text{direct}}}{C_{\text{cap}}} \times 0.042.$$

IV. RESULTS OF ANALYSES

As discussed previously in Section III, two major categories of analyses were conducted. The first category consists of the evaluation of effects of the working-fluid heater/vaporizer pinch point on plant geofluid effectiveness and cost of electricity. The second category pertains to a similar evaluation relative to effects of the approach of condensing temperature to wet-bulb temperature in the evaporative condenser.

1. Effects of Heater/Vaporizer Pinch Point

- 1.1 Summary of Cycles Analysed. This part of the study consists of comparisons of geofluid effectiveness and cost of electricity calculated for a number of selected plant cycles in which the working-fluid heater/vaporizer pinch point was varied between about 2.5 and 19°F. The cycles considered were supercritical and utilized a 0.96 isobutane/0.04 heptane working fluid operating at a heater pressure of 600 psia; plant inlet geofluid was assumed to be at 360°F. Plants with and without geofluid outlet temperature constraints were included, which correspond to cycles both with and without turbine exhaust recuperators. Turbine inlet (heater outlet) temperature and condensing (or bubble point) temperature were assumed fixed at 310.5 and 90°F for all cases.
- 1.2 Presentation of Results. For each of the cycles selected (types shown in Table 2) values of the fractional changes in energy produced, $\frac{\Delta kWh}{kWh}$, and cost of electricity, $\frac{\Delta COE}{COE}$, relative to the isobutane plant (defined in Table 1) are plotted in Figure 3 as functions of heater/vaporizer pinch point temperature difference. Note that since the geofluid flows were held the same for the perturbed plants as for the reference plant, fractional changes in energy produced are equal to fractional changes in net plant geofluid effectiveness (Wh/lbm geofluid). Curves are shown both for plants having the outlet geofluid temperature constrained to 160°F (with recuperators), and for plants without the temperature constraint (without recuperators).

Figure 3. Effects of Heater Pinch Point on Energy Produced and Cost of Electricity, Resource Temp. = 360°F, Wet Bulb Temp. = 60°F

In order to maintain the fixed heater and condenser outlet working-fluid temperatures for the different heater pinch points selected for the cases having the 160°F geofluid outlet temperature, the pinch point temperature differences in the recuperators had to take on different values. These values are shown near the several dots along the bottom curve. The recuperator pinch point increased as heater pinch was increased, and at the highest heater pinch point shown (18.8°F) no recuperator was required. Sizes and costs of other plant components were affected by changes (at constant geofluid flow) in the heater pinch point, as well. The effects of changes in individual major plant components on their contributions to the total fractional change in cost of electricity, $\frac{\Delta \text{COE}}{\text{COE}}$, are shown in Table 7 for each of the cases evaluated. Note that the first column in Table 7 pertains to plants both with and without recuperators.

As mentioned in Section II, frictional pressure drops in piping and components were neglected, generally; however, for the heater pinch point at minimum cost of electricity for the plants without the geofluid outlet temperature constraint (about 4°F), the working-fluid pressure drop in the heater was predicted to be about 45% larger than the 20 psi pressure drop estimated for the heater having a more conventional pinch point of 9°F. The fractional changes in geofluid effectiveness and cost of electricity predicted for this increment in pressure drop of $0.45 \times 20 = 9$ psi were -0.0023 and +0.0020, respectively. This effect on cost of electricity is shown by the short dashed lines in the lower left corner of the plot.

- 1.3 Interpretation of Results. For the assumptions made in the analyses, the plants (both with and without recuperators) show increasing power output as heater/vaporizer pinch point is decreased; the cost of electricity, however, is predicted to reach a minimum value. Figure 3 shows that a heater/vaporizer pinch point of about 4°F corresponds to the minimum cost of electricity for the plants without recuperators when changes in component pressure drops were neglected; accounting for increases in working fluid

TABLE 7. EFFECTS OF HEATER PINCH POINT ON ENERGY PRODUCED
AND COST OF ELECTRICITY, 0.96 ISOBUTANE/0.04 HEPTANE

Geofluid Outlet Constraint	160°F					None			
Heater Pinch Point, °F	19	16	9	7	6	9	6	4	3
Recuperator Pinch Point, °F	No Recup	33	9	4	1	No Recup			
ΔkWh/kWh	-0.003	0.027	0.121	0.141	0.156	0.108	0.140	0.157	0.173
Energy Output ΔCOE/COE	0.003	-0.026	-0.108	-0.123	-0.135	-0.098	-0.123	-0.136	-0.148
Heater ΔCOE/COE	-0.021	-0.019	-0.009	-0.006	0.007	-0.008	-0.001	0.005	0.014
Evaporative Condenser ΔCOE/COE	0.033	0.032	0.028	0.026	0.026	0.039	0.041	0.042	0.043
Recuperator ΔCOE/COE	0	0.002	0.007	0.012	0.025	0			
Turbine ΔCOE/COE	0.005	0.005	0.007	0.008	0.008	0.007	0.008	0.008	0.009
Turbine Piping ΔCOE/COE	0.002	0.002	0.003	0.003	0.003	0.003	0.003	0.003	0.003
Generator ΔCOE/COE	-0.002	-0.001	0.001	0.001	0.001	0.001	0.001	0.001	0.002
Misc. W.F. Components ΔCOE/COE	-0.003	-0.002	0.000	0.001	0.001	0.000	0.001	0.002	0.002
Cooling Water ΔCOE/COE	-0.000	-0.000	-0.000	-0.001	-0.001	0.002	0.003	0.002	0.003
Indirect Capital ΔCOE/COE	0.001	0.002	0.004	0.005	0.007	0.004	0.005	0.006	0.007
Total ΔCOE/COE	0.019	-0.006	-0.067	-0.074	-0.058	-0.050	-0.062	-0.066	-0.065

pressure drops in the heater for the smallest pinch points is shown in Figure 3 to increase the magnitude of the pinch point corresponding to minimum cost of power from 4° to near 5°F.

For the plants whose outlet geofluid temperatures were constrained at 160°F (for which recuperators were used) the heater pinch point for minimum power cost was found to be 7 or 8°F. The major reason for this "optimum" pinch point being larger than that for the cases without recuperators is that the required recuperator size and cost increased markedly for the lower heater pinch points, resulting in higher total capital costs and costs of power. This effect can be seen in Table 7.

The last point of interest shown by Figure 3 is that use of a recuperator to prevent silica precipitation (by keeping the geofluid outlet temperature above 160°F) is predicted to "pay for itself" in terms of producing power at an equal or even a lower cost than the "best" unrecuperated plant because of the reduction in required plant heat rejection.

2. Effects of Evaporative Condenser Approach to Wet Bulb Temperature

2.1 Summary of Cycles Analysed. The second part of the study includes comparisons of geofluid effectiveness and cost of electricity calculated for a number of selected cycles in which the final condensing (or bubble point) temperature was varied between 70 and 105°F while assuming a constant ambient wet-bulb temperature of 60°F. Supercritical cycles were selected which vaporized isobutane at a pressure of 1000 psia, and 0.96 isobutane/0.04 heptane at a pressure of 600 psia; plant inlet geofluid temperature was assumed to be 360°F. All of the plants considered in this part of the study were assumed to have an outlet geofluid temperature constraint of 160°F; each of the plants with the 0.96 iC_4 /0.04 C_7 working fluid employed a turbine exhaust recuperator (the isobutane plants did not).

2.2 Presentation of Results. Results of the cycle and value analyses for this part of the study are shown in Figure 4 as fractional increments (relative to the reference plant defined in Table 1) in energy produced, $\frac{\Delta \text{kWh}}{\text{kWh}}$, (corresponds to increments in geofluid effectiveness) and in cost of electricity, $\frac{\Delta \text{COE}}{\text{COE}}$, plotted as functions of condensing (bubble point) temperature. Also shown is the auxiliary scale "approach to wet-bulb temperature". Curves are shown in Figure 4 for both isobutane plants and plants using the 0.96 iC_4 /0.04 C_7 mixed-hydrocarbon working fluid.

In order to maintain constant heater outlet conditions at the fixed geofluid flow assumed, major changes in heater pinch point and in component sizes and costs were required as the condensing temperature was changed. These effects are shown in Table 8 along with the contributions to fractional changes in cost of electricity ($\Delta \text{COE}/\text{COE}$) associated with changes in performance and component capital costs.

Plant power balances are shown in Figure 5 as functions of condensing temperature for both working fluids. The figure shows how the turbine power and the working-fluid-pumping and evaporative-condenser heat rejection parasitics vary with condensing temperature at the fixed wet bulb temperature of 60°F. Also shown in Figure 5 for comparison, are estimates of net plant geofluid effectiveness for both working fluids at a condensing temperature of 85°F when a shell-and-tube condenser plus a wet cooling tower were substituted for the evaporative-condenser in each of the two plants. These estimates were made following the calculation procedures discussed in Reference 1.

2.3 Interpretation of Results. The same general trends of power produced and cost of power with approach of condensing temperature to wet bulb temperature are shown in Figure 4 for the mixed hydrocarbon working fluid as for isobutane (using the evaporative condenser heat-rejection system). However, the best values of geofluid effectiveness and cost of electricity are about 11% and 6% better

Figure 4. Effects of Condensing Temperature on Electrical Energy Production and Cost When Using Evaporative Condenser Heat Rejection

**TABLE 8. EFFECTS OF APPROACH OF CONDENSING TEMPERATURE
TO AMBIENT WET BULB ON ENERGY PRODUCED AND COST OF ELECTRICITY**

Working Fluid	Isobutane			0.96 Isobutane/0.04 Heptane			
Condensing Temperature, °F	105	90	75	105	90	75	70
Approach to Wet Bulb, °F	45	30	15	45	30	15	10
Heater Pinch Point, °F	11	14	16	4	9	12	14
Recuperator Pinch Point, °F	No Recuperator			8	9	8	9
$\Delta kWh/kWh$	0	0.048	0.018	0.051	0.121	0.160	0.133
Energy Output $\Delta COE/COE$	0	-0.046	-0.018	-0.048	-0.108	-0.138	-0.117
Heater $\Delta COE/COE$	0	-0.008	-0.013	0.005	-0.009	-0.013	-0.018
Evaporative Condenser $\Delta COE/COE$	0	0.037	0.151	0.003	0.028	0.086	0.142
Recuperator $\Delta COE/COE$	No Recuperator			0.008	0.007	0.008	0.007
Turbine $\Delta COE/COE$	0	0.004	0.011	0.003	0.007	0.014	0.018
Turbine Piping $\Delta COE/COE$	0	0.002	0.004	0.001	0.003	0.005	0.006
Generator $\Delta COE/COE$	0	0.001	0.002	-0.001	0.001	0.002	0.002
Misc. W.F. Components $\Delta COE/COE$	0	-0.002	-0.003	0.002	0.000	-0.001	-0.001
Cooling Water $\Delta COE/COE$	0	-0.000	-0.001	-0.000	-0.000	-0.001	-0.001
Indirect Capital $\Delta COE/COE$	0	0.003	0.015	0.002	0.004	0.010	0.016
Total $\Delta COE/COE$	0	-0.009	0.148	-0.025	-0.067	-0.028	0.054

Inlet geofluid temperature = 360°F
 Outlet geofluid temperature = 160°F
 Wet bulb temperature = 60°F

Figure 5. Effects of Working Fluid on Plant Power Balance
 When Using Evaporative Condenser Heat Rejection

for the $0.96 \text{ iC}_4/0.04 \text{ C}_7$ working fluid than for isobutane. As would be expected, the optimum approach of condensing temperature to ambient wet bulb temperature (a type of pinch point) is smaller when maximizing geofluid effectiveness than when minimizing cost of power; at the lowest condensing temperatures the size and cost of the required evaporative condenser units more than offset the increased turbine power produced. An approach of condensing temperature to wet bulb of about 30°F appears to be near optimum for minimum cost of power for both working fluids studied. For maximizing geofluid effectiveness the optimum approach temperature was found to be about 15°F for the isobutane/heptane mixture.

The power balances presented in Figure 5 show that the turbine work per pound geofluid and evaporative-condenser parasitic losses are somewhat comparable for the two working fluids at the different condensing temperatures. The significant difference shown in net plant work per pound geofluid (net plant geofluid effectiveness) lies in the much reduced working-fluid pumping power required for the lower heater pressure of the isobutane/heptane cycle.

It can be seen from Figure 5 that the plant improvements predicted to result from use of the $0.96 \text{ iC}_4/0.04 \text{ C}_7$ working fluid, rather than isobutane, are smaller when heat rejection is accomplished with the evaporative condenser system than when a shell-and-tube condenser and wet cooling tower are used. The mixed hydrocarbon working fluid shows about a 13% increment in geofluid effectiveness, relative to isobutane using the shell-and-tube condenser/wet-tower combination; when evaporative-condenser heat rejection is used, the mixed hydrocarbon retains a performance advantage (relative to isobutane), but the increment is reduced to about 10%. The mixed hydrocarbon working fluid requires a counterflow heat exchange process to minimize thermodynamic irreversibilities occurring during condensation at constant pressure because of the temperature range over which condensation occurs. The evaporative condenser

does not provide this counterflow heat exchange to the extent it is provided by the shell-and-tube condenser; hence, a fraction of the geofluid effectiveness advantage of the mixed hydrocarbon is lost when using the evaporative condenser. The evaporative condenser used in combination with the mixed-hydrocarbon working fluid at the 360°F resource temperature appears from these estimates, however, to show an overall performance advantage relative to plants using either a pure isobutane working fluid or shell-and-tube condenser/wet-tower heat rejection. Cost of power comparisons cannot be made, at this time, between the evaporative-condenser and shell-and-tube condenser/wet-tower systems because detailed capital costs of those components, on a comparable basis, have not been assembled.

3. Market Penetration/Resource Utilization. As indicated in the Introduction, one of the major purposes of the present evaluation was to determine the approximate values of minimum approach temperature differences, or pinch points, for working-fluid vaporizers which will result in minimum cost of electricity. It is perhaps of interest to estimate how the production of electricity by investors from geothermal energy might be influenced as the actual pinch point deviates from the optimum. To illustrate this sensitivity and the importance of selecting a near-optimum pinch point, use has been made of some market penetration analyses conducted by Technecon Analytic Research, Inc. using investment-decision models and methods described in Reference 5. Technecon's analyses indicated that a 10% increase in plant geofluid effectiveness (at constant capital cost) should result in a 33% increase in the amount of electric power which will be produced in a "free market" from hydrothermal resources in the temperature range between 350 and 400°F. Reducing plant capital cost 10% was projected to increase power production by 13%.

As an example of the sensitivity of the amount of geothermal/electric power produced in the year 2000 (resource utilization) to changes in the vaporizer pinch point, unrecuperated plants having no outlet

geofluid temperature constraints were compared for vaporizer pinch points of 5° and 15°F. From Figure 3 the fractional difference in geofluid effectiveness between these cases is $0.148 - 0.043 = 0.105$, with the smaller pinch point providing the higher geofluid effectiveness. These two plants were estimated to have about the same capital cost in mills/kWh. Therefore, for this example about $33\% \times 1.05 = 35\%$ more power would be predicted to be produced in the year 2000 from resources in the 350 to 400°F temperature range by plants having optimized heater/vaporizer pinch points (near 5°F) than by those having 15°F pinch points.

A similar example can be illustrated for the effect of selecting an "optimized" evaporative condenser size. For the 60°F wet bulb temperature assumed, the fractional difference in geofluid effectiveness (Figure 4) between a plant having a condensing temperature of 90°F relative to one having a condensing temperature of 105°F is $0.121 - 0.051 = 0.070$. The corresponding fractional change in capital cost was estimated to be -0.022. About $33 \times 0.70 + 13 \times 0.22 = 26\%$ more produced power would be predicted in the year 2000 from the 350 to 400°F resource subset with plants having evaporative condensers of a near "optimum" size (resulting in a 30°F approach of condensing temperature to wet bulb), than with plants in which the condensate has a 45°F approach to wet bulb temperature.

V. CONCLUSIONS

The overall conclusions reached from the study are first, that plant performance and cost of electricity are moderately sensitive to the selection of heater-vaporizer and heat-rejection system approach temperature differences (pinch points).

Second, the magnitudes of pinch points for maximum geofluid effectiveness are not the same as those corresponding to minimum cost of electricity. The approach of condensing temperature to wet-bulb temperature at maximum geofluid effectiveness for evaporative-condenser heat rejection was predicted to be about 15°F for the 0.96 iC_4 /0.04 C_7 plant; the magnitude of the minimum approach temperature difference corresponding to minimum cost of electricity was estimated as 30°F.

For unrecuperated 0.96 iC_4 /0.04 C_7 plants the heater/vaporizer pinch point corresponding to minimum cost of electricity was predicted to be about 5°F, which is lower than the value of 10°F commonly assumed in many previous analyses conducted for the Conversion Technology Program.

Finally, for a 360°F resource the mixed hydrocarbon working fluid 0.96 iC_4 /0.04 C_7 was shown to retain a geofluid-effectiveness advantage (about 10%), relative to isobutane for plants utilizing evaporative-condenser heat rejection.

A somewhat more detailed list of results and conclusions is presented in the Summary, Section I, and is not repeated here. Included, there, is a brief summary of the effect of changes of heater/vaporizer and evaporative condenser sizes (and pinch points) on market penetration of geothermal/electric power production.

V. REFERENCES

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