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An Optimization Technique for Geothermal Power Plants Using a Binary Fluid Cycle

by
S. W. Wilson
R. A. Walter

October 1976

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SUMMARY

Optimal utilization of geothermal energy is not only desirable but necessary due to the relatively low energy content of geothermal fluids and the subsequent lower efficiencies of energy conversion schemes compared with the efficiencies of conventional power plants.

Although a binary fluid cycle has been recommended as the proper power plant concept for these low temperature heat sources, optimization is needed if economic viability from geothermal power plants is to be demonstrated. Consequently a cost optimization routine was devised and installed as an available option to the GEOCOST code which simulates, among others, binary fluid cycle power plants for use with geothermal energy sources.

The optimization method is based on the hypothesis that a binary fluid cycle which utilizes the available energy in the geothermal fluid most efficiently is also a cycle that will produce power the most inexpensively (at least within only a few percent of the minimum cost cycle as determined by extensive GEOCOST runs). The optimization technique (optimum with respect to minimizing the cost of generating power) was tested over a range of different reservoir conditions (e.g. geothermal fluid temperature at the wellhead, temperature and pressure of the working fluid at the turbine inlet, sink temperatures, etc.). From the results of these tests it was concluded that:

- By specification of the working fluid and the prevailing ambient temperature for heat rejection, a minimum cost binary fluid cycle can be completely designed and specified once the temperature of the geothermal fluid is known.

- With the given input conditions, there exists a geothermal fluid temperature below which a subcritical cycle is always recommended and above which a supercritical cycle is always recommended.
- Minimum cost designs of the binary fluid power plant result in cycles which contain minimum superheat of the working fluid at the turbine inlet.

NOMENCLATURE

APP - approach to the heat exchanger, °F (°K)
 C_p - specific heat of water, Btu/lb_m·°F (J/kg·°K)
 COP - cost of power, mills/kW-hr
 H₁ - enthalpy of working fluid at turbine inlet, Btu/lb_m (J/kg)
 H₄ - enthalpy of saturated liquid working fluid at maximum cycle pressure, Btu/lb_m (J/kg)
 PINCHP - pinch point temperature difference, °F (°K)
 P_{internal} - internal power requirements, kW_e
 P_{net} - net power output, kW_e
 P_{total} - gross power output, kW_e
 QWAT - geothermal fluid flow rate, lb_m/hr (kg/hr)
 QISO - working fluid flow rate, lb_m/hr (kg/hr)
 T_h - source temperature, °R (°K)
 T_o - sink temperature, °R (°K)
 W_{max} - maximum reversible work, Btu/lb_m (J/kg)
 WTI - geothermal fluid temperature entering the power plant, °R (°K)

WT3 - geothermal fluid temperature at heat exchanger
pinch point, °R (°K)

ΔB - change in availability

ΔQ - net heat transfer

ΔS - net entropy change

ΔW - net work produced

η - thermodynamic or Rankine cycle efficiency, %/100

η_u - utilization efficiency, %/100

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AN OPTIMIZATION TECHNIQUE FOR GEOTHERMAL POWER PLANTS USING A BINARY FLUID CYCLE

S. W. Wilson and R. A. Walter

INTRODUCTION

As an available option to the GEOCOST code,* an optimization routine has been written to predict the binary fluid cycle designs which result in minimum cost over a wide range of geothermal fluid temperatures. This procedure is performed without entering into the exhaustive and costly exercises of the alternative trail and error approach.

The optimization routine locates the power cycle which yields the maximum work from the available energy in the geothermal fluid; this cycle has been verified to also be the minimum cost cycle. The theory involved in this development as well as a discussion of the subsequent verification studies are included.

DISCUSSION

Prior to the introduction of this optimization technique, the location of a minimum cost cycle design was obtainable only by an exhaustive search of many GEOCOST case runs. This procedure was costly, time consuming, and often involved an incomplete search matrix. In the process of determining the preferable cost minimizing optimization technique, a number of key system parameters were investigated including the minimum geothermal brine flow rate, the maximum Rankine cycle efficiency, and the maximum utilization efficiency (a reflection of the most efficient use of the geothermal fluid in generating electrical power). As a result of these investigations,

* Computer program developed at BNW under USERDA funding (Bloomster, et al., Feb. 1975)

the maximum utilization efficiency was selected as the best parameter for locating a minimum cost design. The details of this optimization technique, the theory behind it, as well as key results are presented in the following sections.

OPTIMIZATION THEORY

The maximum work that can be produced from any heat source is specified by the second law of thermodynamics. The second law states that the efficiency by which work ΔW can be produced by transferring heat ΔQ from a high temperature source at T_h in a Carnot heat engine and rejecting heat to a low temperature sink at T_o is as follows:

$$\eta = \frac{\Delta W}{\Delta Q} = \frac{T_h - T_o}{T_h} \quad (1)$$

More specifically, the maximum work available can be expressed in an integral form as a summation of heat transferred to an infinite number of small Carnot heat engines between the source temperature T_h and the sink temperature T_o . The net maximum work for a geothermal liquid source at temperatures varying between $T_{h,in}$ and $T_{h,out}$ can be expressed as:

$$W_{\max} = - \int_{T_{h,in}}^{T_{h,out}} \left(\frac{T_h - T_o}{T_h} \right) dQ \quad (2)$$

By expressing dQ as a function of enthalpy, the following equation can be written, assuming constant specific heat.

$$dQ = dH|_p = C_p dT_h \quad (3)$$

Substituting equation (3) into equation (2):

$$W_{\max} = - \int_{T_{h,in}}^{T_{h,out}} \left(\frac{T_h - T_o}{T_h} \right) C_p dT_h \quad (4)$$

$$\text{or } W_{\max} = - \int_{T_{h,in}}^{T_{h,out}} C_p \left[1 - \frac{T_o}{T_h} \right] dT_h$$

$$\text{or } W_{\max} = - \int_{T_{h,in}}^{T_{h,out}} C_p dT_h + C_p T_o \int_{T_{h,in}}^{T_{h,out}} \frac{dT_h}{T_h}$$

At constant pressure, by the definition of enthalpy and entropy, equation (4) can be expressed as:

$$W_{\max} = - \left[\Delta H - T_o \Delta S \right] \Big|_{T_{h,in}}^{T_{h,out}} \quad (5)$$

This quantity represents the maximum reversible work that can be produced, and is commonly referred to as the change in availability, ΔB . Knowing $\Delta H = C_p dT_h$ and $\Delta S = C_p \ln \frac{T_{h,in}}{T_{h,out}}$, equation

(5) can be reduced to

$$W_{\max} = - \left[C_p (T_{h,in} - T_{h,out}) - T_o C_p \ln \frac{T_{h,in}}{T_{h,out}} \right] \quad (6)$$

The term $W_{\max}$ is expressed in Btu/lb_m. The maximum total work that can be produced by a geothermal reservoir is equal to $QWAT \times W_{\max}$. The actual work produced by the conversion process is the gross turbine output minus the internal power consumption, or P_{net} . By dividing P_{net} by $(QWAT \times W_{\max})$ an expression for the efficiency of the geothermal system can be found, and is identified as the utilization efficiency:

$$\eta_u = \frac{P_{\text{net}}}{QWAT \times W_{\max}} = \frac{P_{\text{total}} - P_{\text{internal use}}}{QWAT \times C_p \left[T_{h,in} - T_{h,out} - T_o \ln \frac{T_{h,in}}{T_{h,out}} \right]} \quad (7)$$

This expression is the basis for the optimization study. This efficiency should not be confused with the thermodynamic cycle efficiency, η , defined as:

$$\eta = \frac{P_{net}}{QWAT \times C_p \times (T_{h,in} - T_{h,out})} \quad (8)$$

The cycle efficiency does not accurately reflect the efficient use of the available energy in a geothermal resource because the geothermal fluid which is reinjected after use in the power plant is not considered as a loss in η . Because of the high cost of drilling wells and laying transmission piping to and from the power plant, one cannot afford to disregard the available energy in the rejected geothermal fluid.

The utilization efficiency was used in conjunction with the binary fluid cycle model to determine the optimum cycle for any particular resource temperature, sink temperature, and working fluid.

METHOD OF ANALYSIS

To identify the optimum cycle it is necessary to locate the particular values of the working fluid's pressure and temperature at the turbine inlet which result in the maximum value for utilization efficiency. By examining Figures 1 and 2, one can see how the identification of the turbine inlet conditions (state 1 in Figures 1 and 2) specifies the entire Rankine cycle. This is because states 2SATV and 2SATL are established by the prevailing sink conditions, the type of working fluid used, and the necessary maximum temperature difference in the condenser; state 3 is set by the amount of pump work required to bring the working fluid up to its maximum pressure (the turbine inlet pressure); and state 2 can be

calculated by determining the known deviation from an isentropic expansion of the turbine. Thus by fixing the conditions at state 1 the entire cycle is specified.

FIGURE 1. Subcritical Cycle

FIGURE 2. Supercritical Cycle

In order to determine the turbine inlet state (the conditions at state 1), a number of input parameters are required, including:

- entering geothermal fluid temperature
- prevailing sink conditions
- selection of working fluid
- maximum allowable temperature difference in the condenser
- turbine efficiency.

Using these input parameters, the turbine inlet state was determined by the location of the turbine inlet pressure and the approach temperature of the heat exchanger (approach = entering geothermal fluid temperature - peak working fluid temperature). The location of the optimal turbine inlet state involved a search of all realistic cycles for the turbine inlet state that defined a cycle having the greatest utilization efficiency for the set input parameters.

CYCLE CONSTRAINTS

To insure a reasonable cycle selection, a number of constraints were placed upon the optimization search. These constraints can be listed as follows:

- a. minimum approach = 20°F (266°K)
- b. minimum heat exchanger pinch point temperature* = 15°F (264°K)
- c. minimum working fluid temperature (at state 2SATL) established as the sink temperature + range of the condenser** + 5°F (258°K)
- d. no expansion of the working fluid into the two-phase region was allowed in the turbine
- e. maximum working fluid pressure = 1500 psia (1.034E7 N/m²)
- f. temperature of the geothermal fluid leaving the heat exchangers (WT4) must be equal to or greater than the working fluid temperature at state 3 + 20°F (266°K).

* Lowest allowable temperature difference in a heat exchanger (see Figure 3)

** Temperature rise in the condenser's cooling water

FIGURE 3. Typical Temperature Profiles in Sub-Critical Heat Exchangers

These constraints were established on the basis of thermodynamic limitations and practical design considerations. For the different geothermal fluid temperatures investigated, the minimum pinch point constraint (b), the two-phase constraint (d), the maximum pressure constraint (e), and the exhaust pressure constraint (c) aren't noticeably effected by changing the geothermal fluid temperatures. However, the limiting isotherm constraint (a) and the WT4 constraint (f) do change appreciably with the changes in the geothermal fluid temperatures. As a consequence, it is helpful to view the effect of these constraints and how they limit the general search area for different geothermal fluid temperatures. The geothermal fluid temperatures of 200, 250, 300, 350, 400 and 500°F are schematically presented on isobutane P-H (pressure-enthalpy) diagrams in Figures 4 through 9 as being illustrative of the way these optimization constraints adjust with changes in geothermal fluid temperature. The shaded area in each schematic represents the general search region in which the optimization routine locates the turbine inlet state (i.e., state 1) that yields a cycle of maximum utilization efficiency subject to the additional constraint (b).

FIGURE 4. Geothermal Temperature - 200°F

FIGURE 5. Geothermal Temperature - 250°F

FIGURE 6. Geothermal Temperature - 300°F

FIGURE 7. Geothermal Temperature - 350°F

FIGURE 8. Geothermal Temperature - 400°F

FIGURE 9. Geothermal Temperature - 500°F

OPTIMIZATION RESULTS

The results of the optimization search performed for the different geothermal fluid temperatures within the stated constraints can be displayed on plots of utilization efficiency vs. turbine inlet pressure, with the parametric variables of approach and geothermal fluid temperature. The results of these representative cases can be seen in Figures 10 through 15.

FIGURE 10. Geothermal Temperature - 250°F (394°K) and 200°F (366°K)

FIGURE 11. Geothermal Temperature - 300°F (422°K)

Each approach curve in Figures 10 through 15 behave in accordance with the thermodynamics involved but where the approach curves begin and end are functions of the constraints encountered. Consequently, a discussion of these constraints and their specific role in Figures 10 through 15 is appropriate. This information is presented in tabular form in Table 1.

As an additional point of interest, Figure 11 illustrates some of the interesting thermodynamics that occur in this temperature regime. The curve which represents an "approach" of 20°F decreases after reaching a maximum at around 300 psia, but then begins to increase again at around 500 psia. The

FIGURE 12. Geothermal Temperature - 350°F (450°K)

FIGURE 13. Geothermal Temperature - 400°F (477°K)

FIGURE 14. Geothermal Temperature - 450°F (505°K)

FIGURE 15. Geothermal Temperature - 500°F (533°K)

explanation for this behavior is in the definitions for utilization efficiency and the geothermal flow rate:

$$\eta_u = \frac{P_{net}}{QWAT(\Delta B)} \quad (9)$$

$$QWAT = \frac{Q_{ISO}(H1-H4)}{C_p(IWT1-WT3)} \quad (10)$$

As the turbine inlet pressure increases near the critical pressure (529 psia for isobutane) for a constant approach temperature, the heat of vaporization for isobutane rapidly decreases, causing a reduction in the value of (H1-H4). This decrease lowers the geothermal fluid flow rate, QWAT, and increases the utilization efficiency.

TABLE 1. Comments on Approach Line Limitations

GEOTHERMAL FLUID TEMP. (°F)	APPROACH RANGE (°F)	CONSTRAINT LIMITING THIS APPROACH	COMMENTS
200 (Fig. 10)	20-40	Two-phase	The higher the turbine inlet pressure the smaller the allowable approach range (see Fig. 4).
250 (Fig. 10)	20-60	Two-phase	The WT4 constraint first appears and constrains the higher approaches from the lower turbine pressures (see Fig. 5).
300 (Fig. 11)	20-60	Two-phase	The WT4 constraint limits all approaches at the lower turbine inlet pressures (see Fig. 6).
350 (Fig. 12)	20-45	Two-phase	The minimum approach isotherm is now large enough to make supercritical pressures feasible; the two-phase constraint is an active constraint only for these supercritical pressures (see Fig. 7).
	60	WT4	The approach lines are not only limited at the low turbine inlet pressure side, but on the higher pressure side too (see Fig. 7).
400 (Fig. 13)	20	Max. pressure	If the max. turbine inlet pressure were set higher, the limiting constraint would become the two-phase constraint (see Fig. 8).
	50-100	WT4	See Figure 8.
450 (Fig. 14)	20-40	Max. pressure	
	70	WT4	
500 (Fig. 15)	20-60	Max. pressure	See Figure 9.
	100	WT4	See Figure 9.

An additional comment regarding the 350°F case (Fig. 12) is concerned with the advisability of choosing a sub or supercritical cycle. At these conditions and temperature a subcritical cycle seems more beneficial (from a utilization efficiency point of view) but there is instability (i.e., small changes in pressure result in large changes in efficiency) in the subcritical cycle near the critical pressure. In order to avoid the instability, it is recommended that a supercritical cycle be chosen.

This discontinuity in the "approach lines" at the critical pressure is caused by the higher nonlinearity of the fluid properties near the critical point and the shifting location of the pinch point temperature difference. In the subcritical cycle the pinch point always exists at the point where the working fluid is a saturated liquid. This is indicated in the subcritical heat exchanger temperature profile of Figure 3. The pinch point exists at the preheater exit or vaporizer inlet. As the inlet turbine pressure nears the critical point from below, the pinch point location rapidly moves toward the critical isotherm. Very near the critical point the slope of the vapor pressure curve is close to zero, and the pinch point approaches the critical isotherm. This is depicted on a P-H (pressure vs. enthalpy) diagram for isobutane in Figure 16, where the locus of pinch points are shown for increasing working fluid pressure. As soon as the cycle goes supercritical, the pinch point location tends to follow the critical isotherm. The abrupt shift in direction as pressure is increased through the critical pressure is mainly because the slope of the critical isotherm is near zero around the critical point.

The discussion in this section has presented a scenario on the determination of the most efficient binary fluid power

FIGURE 16. Pressure versus Enthalpy Diagram for Isobutane; Pinch Point Location

plant cycle (with respect to utilizing available energy) that exists for a given set of inlet conditions and geothermal fluid temperature. The demonstration that these maximum utilization efficiency cycles are also minimum cost cycles is discussed in the following section.

COMPARISON OF OPTIMIZATION RESULTS WITH COST OF POWER

The results of the optimization study using the utilization efficiency are based upon thermodynamic reasoning, and a comparison of these results with computed costs is necessary to establish the validity of this analysis. The GEOCOST computer program provides an output value of the cost of power, COP, in mills/kW-hr for each of the cases considered in the optimization study. This economic yardstick was used to compare the minimum cost design with the design that has a maximum utilization efficiency.

For each particular geothermal fluid temperature considered, a matrix of cases was selected with the boundaries defined by the aforementioned constraints. The cost of power was computed by GEOCOST for each case considered in the constrained matrix, and the design that resulted in the minimum cost of power was saved for comparison with the maximum utilization efficiency design selection.

The results of the generation of cost data and the comparison with the optimization study can be shown in Table 2. The following information is provided:

- (a) the optimum turbine inlet conditions for each geothermal fluid temperature considered
- (b) the computed utilization efficiency for each optimum selection
- (c) the cost of power for the optimum design based on (a)

TABLE 2. Optimization Results

Geothermal Fluid Temperature in °F (°K)	Maximum Utilization Efficiency (%/100)	Turbine Inlet Conditions at Max. η_u		Energy Cost at Max. η_u (mills/kW-hr)	Minimum Energy Cost (mills/kW-hr)	Cost Deviation (%)
		Pressure in psia (N/m ²)	Temperature in °F (°K)			
200 (366)	0.100	140 (9.653E5)	159 (343.7)	248.1	234.8	5.7
250 (394)	0.183	200 (1.379E6)	185 (358.0)	87.24	85.06	2.6
300 (422)	0.269	310 (2.137E6)	220 (377.6)	43.99	44.01	0.1
350 (450)	0.390	651 (4.488E6)	305 (424.8)	28.50	28.07	1.5
400 (477)	0.429	980 (6.756E6)	350 (449.7)	23.24	20.98	10.8
450 (505)	0.440	1500 (1.034E7)	410 (483.2)	20.77	18.11	14.7
500 (533)	0.440	1500 (1.034E7)	440 (499.8)	18.26	16.49	10.7

- (d) the cost of power for the design which resulted in the lowest cost, irregardless of the optimum selection in (a)
- (e) the percentage difference between the two values of COP from (c) and (d).

This table indicated the close correlation between the cost of the optimum cycle selected by the utilization efficiency and the cost of the most economic design. These results can also be shown on a pressure vs. enthalpy diagram for isobutane in Figure 17. Notice in this figure the close correlation between optimum turbine inlet state points. The optimum selections based on maximum η_u and minimum COP are shown for each geothermal fluid temperature considered.

Based on the performed analysis and comparison, a number of general statements concerning the optimum cycle selection can be made:

- (a) There exists a close correlation between the maximum utilization efficiency and the minimum cost of power. An optimum cycle can now be selected in GEOCOST without the need for large amount of computer time to search a variety of designs; only the geothermal fluid temperature, sink temperature and type of working fluid are required to make the selection.
- (b) The utilization efficiency increases and the cost of power decreases for an optimum cycle, with increasing geothermal fluid temperature.
- (c) The selected working fluid turbine inlet temperature and pressure increase with increasing geothermal fluid temperature.
- (d) With the given input conditions, there exists a geothermal fluid temperature below which a subcritical cycle is always recommended and above which a supercritical cycle is best.
- (e) The maximization of utilization efficiency results in the conclusion that the working fluid at turbine inlet should contain a minimum amount of superheat. Even though a highly superheated working fluid results in a more efficient extraction of energy in turbine, this increase in energy does not compensate for the increased geothermal fluid flow rate required to raise the working fluid to that high energy state. This conclusion is evident upon examination of Figure 17, where all selected optimum points (that are not constrained to more superheat by the WT4 constraint) lie near the saturated vapor line.
- (f) A low pinch point temperature difference in the heat exchangers is desirable; however, below a value of

about 15°F (264°K) the size of the heat exchangers required increases rapidly due to the low thermal driving force.

- (g) The prevailing sink temperature is fixed for a particular design. The minimum working fluid temperature in the power plant should be as close to this temperature as possible without increasing the condenser surface area dramatically.

CONCLUSIONS

A comprehensive computer model of a binary fluid cycle power plant for use with geothermal resources has been developed. This computer model can be used to assess the technical and economic feasibility of the geothermal resource and to aid in the design of binary power plants. GEOCOST provides information on reservoir and transmission design, major plant components, fluid proper ties throughout the system, type of cycle employed, expected net power output, cost of producing power, and plant performance and design. In addition, an optimization technique has been developed to rapidly select the power plant design with the highest utilization efficiency based upon the maximum use of available energy in the geothermal field. This optimization technique has shown a close correlation with the minimum cost of power and can be used to easily select a minimum cost geothermal plant design knowing the geothermal fluid temperature, the prevailing ambient temperature for heat rejection, and the selected working fluid.

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