CONF-760215--3

LA-UR-76-418

TITLE: GEOTHERMAL ENERGY FOR ELECTRICAL AND NONELECTRICAL APPLICATIONS

AUTHOR(S): Jefferson W. Tester, LASL

Stanley L. Milora, ORNL

MASTER

SUBMITTED TO:

16th Annual Symposium, New Mexico Section, American Society of Mechanical Engineering, Albuquerque, NM, February 26-27, 1976.

los alamos
scientific laboratory
of the University of California
LOS ALAMOS, NEW MEXICO 87544

An Affirmative Action/Equal Opportunity Employer

Form No. 836
St. No. 2629
1/75

UNITED STATES
ENERGY RESEARCH AND
DEVELOPMENT ADMINISTRATION
CONTRACT W-7405-ENG. 36

DISCLAIMER

GEOTHERMAL ENERGY FOR ELECTRICAL AND NONELECTRICAL APPLICATIONS

by

Jefferson W. Tester
University of California
Los Alamos Scientific Laboratory
Los Alamos, New Mexico 87544

Stanley L. Milora
Oak Ridge National Laboratory
Oak Ridge, Tennessee 37830

ABSTRACT

The utilization of geothermal fluids ranging in temperature from 100 to 300°C is discussed from a thermodynamic and an economic viewpoint. Nonaqueous working fluids are evaluated for possible use in sub- and super-critical Rankine power generating cycles, and are compared to more conventional steam flashing cycles. Criteria are presented for determining performance based on the cycle's effectiveness in utilizing the geothermal fluid. Working fluid thermodynamic properties are used to correlate optimum cycle performance at given geothermal fluid temperatures. A generalized method for expressing turbine exhaust end sizes is developed. The geothermal resource potential of the United States for both natural hydrothermal systems and artificially stimulated, dry hot rock systems is discussed. The economics of generating power and of direct utilization for space and process heating applications are compared with fossil fuel and nuclear energy sources.

INTRODUCTION AND SCOPE

The major portion of material presented in this paper has been extracted from a study of geothermal power cycles conducted by Milora and Tester [1]. Readers interested in more detail on the electric power generation aspects of low-temperature resources, like geothermal, solar, or ocean thermal, should consult ref. 1 as well as other numerous papers on this topic [2-9].

Because electric generating efficiencies range from 8 to 20% for geothermal resource temperatures of 100 to 300°C, and because costs associated with producing hot fluid at the surface frequently represent more than 60% of the total capital investment in the power plant, a different set of design criteria is applied than would be to a fossil-fuel fired or nuclear generating cycle. When geothermal fluid production costs are high relative to surface equipment costs, a premium is placed on designing and operating power conversion systems near their thermodynamic limiting efficiencies.

Nonelectric utilization of geothermal energy is frequently more attractive than producing power, particularly where a specific resource temperature can be matched to a user need for process or space heating [10-13]. In this case, the tradeoff between costs associated with producing the geothermal fluid at the surface and costs of heat exchange and fluid transportation are critical.

Electric, nonelectric, and combined or hybrid systems will be discussed with respect to the types of geothermal resources as well as the criteria necessary for effective utilization in a thermodynamic and economic sense.

GEOTHERMAL RESOURCES

The total geothermal resource base can be divided into two major categories, natural hydrothermal and artificially stimulated, dry hot rock systems. Natural hydrothermal systems can be further subdivided into four subsystems: (1) vapor-dominated (dry steam), (2) liquid-dominated (super-heated water), (3) geopressured, and (4) lavas and magmas [3,9]. White and Williams [9] of the U.S. Geological Survey recently published estimates of the total geothermal resource potential for the United States. The combined recoverable energy from vapor- and liquid-dominated systems is $\sim 3 \times 10^{21}$ J and from the dry hot rock resource including all heat above 15°C from 0-10 km in the continental U.S. is $\sim 335 \times 10^{21}$ J (assuming 1% recovery) [9,1].

If all of this energy were converted into electricity this would amount to $\sim$ 8,550,000 MW(e)-centuries. In comparison, the total annual energy consumption for the U.S is presently $\sim 0.1 \times 10^{21}$ J with $\sim$ 390,000 MW(e) as electricity. Clearly, the potential geothermal resource warrants serious consideration as an energy alternative.

Engineering difficulties and high costs primarily caused by the inaccessibility of geothermal heat have limited exploitation of the resource to a few areas of the world where natural systems exist at or near the surface. An additional problem arises for hot dry rock systems in that water is not indigenous to the reservoir, rock permeability and porosity are low, and therefore artificial fluid circulation paths must be created to expose hot rock to the fluid. Because heat transfer rates in dry hot rock systems are controlled by the low thermal conductivity of the rock itself, large surface areas are required [3,14]. One concept being developed by the Los Alamos Scientific Laboratory [14] is to create a large fracture by hydraulic fracturing with injected water under pressure. Elastic theory suggests that a thin, vertically-oriented, disc-shaped fracture of elliptical cross section will be formed at depths over 1 km. A two-hole system connected to the fracture would be used to supply and return water to and from the fractured region to remove heat.

ENERGY UTILIZATION SYSTEMS

Generating electricity from vapor-dominated systems, such as those at The Geysers in California or at Larderello, Italy, is usually accomplished by direct injection of treated geothermal steam into a low-pressure steam turbogenerator. For liquid-dominated or dry hot rock systems, steam vapor can be created by flashing the geothermal fluid to a lower pressure. Then, the saturated steam phase is used to drive a turbogenerator unit, with the unflashed liquid phase either reinjected or discarded (see Fig. 1, B). Binary-fluid Rankine cycles employing nonaqueous working fluids are alternatives to single- and multiple-flashing systems currently in use in various parts of the world (for example Cerro Prieto, Mexico and Wairakei, New Zealand [3]). Binary-fluid cycles involve a primary heat exchange step where heat from the geothermal fluid is transferred to another working fluid which expands through a turbogenerator and then passes to a condenser/desuperheater for heat rejection to the environment. The cycle is completed by pumping the fluid up to the maximum cycle operating pressure (see Fig. 1, A). Dual and topping/bottoming cycles (Fig. 1, C and D) are combinations of two different binary-fluid cycles arranged to optimize power production for each working fluid.

Nonaqueous working fluids with large, low-temperature vapor densities would require smaller, less costly turbines than the low-pressure steam turbines employed in the flashing systems of the same power output. Flashing cycles are, of course, simpler in that they do not require a primary heat exchanger. Models used in characterizing power cycle performance, in a thermodynamic and an economic sense, are developed in a later section and optimization procedures are illustrated for a number of cases.

Nonelectric applications are spread over a wide area. In addition to domestic and industrial space heating, geothermal energy is already used for a number of process heat applications. In New Zealand, geothermal fluids are used to generate high-quality steam for pulp and paper processing and in Iceland geothermal steam is used for mining diatomite [3].

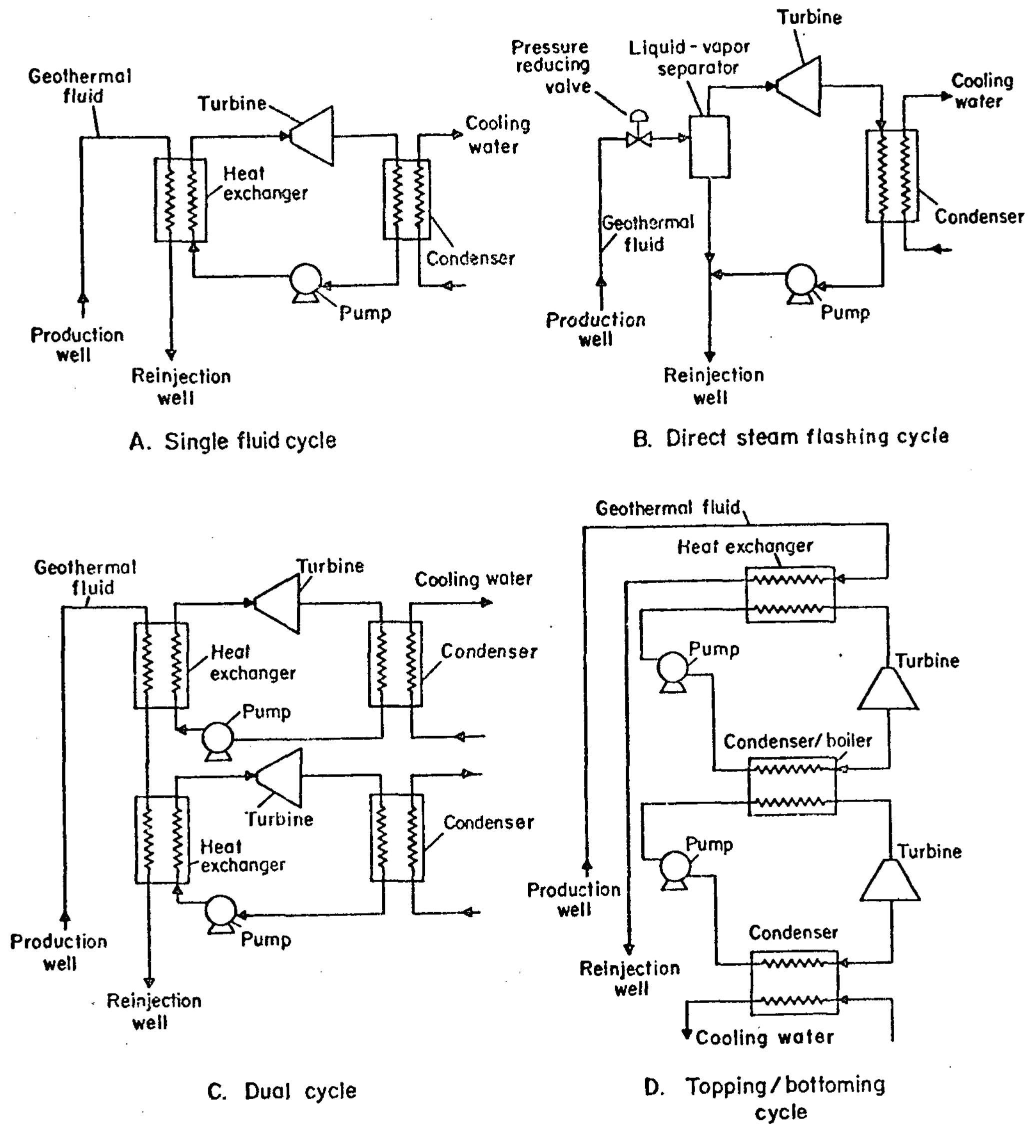

Fig. 1 Power cycle schematics for liquid-dominated or dry hot rock geothermal resources.

As Reistad [13] points out, the critical characteristic of the geothermal resource is its temperature. It is important to match the resource to the needs of the potential user. This effectively defines a lower bound of temperatures in excess of prevailing ambient conditions throughout the world. For natural hydrothermal and for first-generation dry hot rock systems, a 250°C resource temperature is probably an upper limit [13]. Dry hot rock systems are potentially more flexible in that fluid temperatures can be specified by design.

Reistad [13] describes a number of potentially large consumers of geothermal energy at temperatures up to 250°C. In addition to space and water heating, air conditioning and refrigeration, and clothes drying, he examines the temperature-energy spectrum for the manufacturing industries in detail. For example, in the chemical industry, chlorine-caustic soda, synthetic soda ash, aluminum oxide, and Frasch sulfur processes consume large amounts of low-temperature energy. Centrally-located chemical complexes employing many smaller-scale processes also have substantial low-temperature steam requirements which might be ideally suited for a geothermal heat supply system.

Because many processes require significant quantities of electricity and steam, combined or hybrid energy supply systems currently are used. Frequently, electricity is generated in a steam Rankine cycle with low-temperature steam, exhausted from the turbogenerator, used directly in the process. For example, consider the possible use of a proposed dual cycle for a dry hot rock resource at 280°C (see Fig. 2). In the case shown, a saturated steam cycle extracts heat from the high temperature end (280 to 173°C) of the geothermal water loop, and a supercritical isobutane cycle removes heat from 173°C to a reinjection temperature of 55°C.

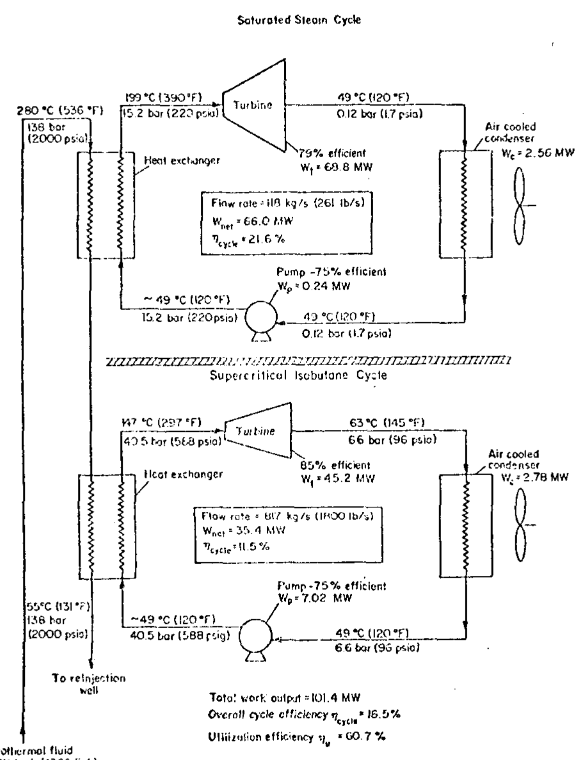

Fig. 2. A saturated steam/isobutane dual Rankine cycle schematic for a 280°C dry hot geothermal resource. Thermodynamic optimum performance conditions shown. (from ref. 1)

Both cycles reject heat to the environment at a condensing temperature of 49°C. Alternatively, direct use might be made of the energy from 173°C to the 55°C reinjection temperature rather than generating electricity in the isobutane cycle. Clearly, there are a variety of options that can be pursued depending on the energy and temperature requirements of the process as well as the cost effectiveness of electricity generation versus direct utilization.

POWER CYCLE PERFORMANCE OPTIMIZATION

Accurate data on the thermodynamic properties of proposed fluids are required to calculate cycle performance. Heat capacity at constant pressure in the ideal gas state C_p^*, vapor pressure P^{sat}, pressure-volume-temperature (PvT) behavior, and liquid density at saturation ρ_ℓ^{sat} are expressed with semi-empirical equations written in reduced form. For example, a modified form of the Martin-Hou equation of state with 21 parameters is used for calculating PvT properties [1]. This equation is accurate for densities above the critical point, a requirement for supercritical Rankine cycle calculations. Derived properties such as entropy and enthalpy are obtained by suitable differentiation and integration of the semi-empirical equations for C_p^*, P^{sat}, $P = f(T,v)$, and ρ_ℓ^{sat}.

Seven working fluids in addition to water were examined. Refrigerants, R-22 ($CHClF_2$), R-600a (isobutane, $i\text{-}C_4H_{10}$), R-32 (CH_2F_2), R-717 (ammonia, NH_3), RC-318 (C_4F_8), R-114 ($C_2Cl_2F_4$) and R-115 (C_2ClF_5) were selected because they provided a range of properties including, critical temperature and pressure, and molecular weight. All of these compounds have relatively high vapor densities at heat rejection temperatures as low as 20°C.

Detailed cycle calculations were performed to examine the effects of cycle operating pressure, heat rejection temperature, temperature differences in the primary heat exchanger, turbine and pump efficiencies, and geothermal fluid temperature. In each case a utilization efficiency η_u was determined which related the actual electrical work produced by the cycle to the maximum work (or availability) possible with specified geothermal source, and heat rejection temperatures. Parametric effects were explored using computer computational techniques [1].

A specific example is included here for a 150°C liquid-dominated resource with heat rejection at 26.7°C and R-115 as the working fluid. Temperature-enthalpy diagrams for four different operating pressures are presented in Fig. 3 to illustrate the dramatic effect that pressure has on cycle performance. In going from a subcritical cycle at 27.5 bars (Case A, P_r = 0.87) to a supercritical cycle at 80.1 bars (Case C, P_r = 2.54), the utilization efficiency η_u increases from 46.5 to 63.2%. This improvement is due primarily to a more uniform heat capacity at the higher pressure and a reduction in the amount of sensible heat rejection (desuperheat). At supercritical pressures there is no phase change and the working fluid heating path can be maintained almost parallel to the geothermal fluid cooling path. As the pressure is increased to 114.4 bars (Case D) cycle performance declines to an η_u of 54.6%. This is caused by the less than ideal efficiencies of the turbine and pump components (85 and 80% respectively) and the larger component work requirements associated with higher pressure operation.

For any given working fluid, there is an optimum set of operating conditions yielding a maximum η_u for particular geothermal fluid and heat rejection temperatures. In screening potential working fluids, some knowledge of the magnitude of η_u and how it changes would be useful. Computer optimizations for the seven working fluids studied were conducted for geothermal fluid temperatures ranging from 100 to 300°C are shown in Fig. 4 (see Ref. 1 for details). At each point, cycle pressures were

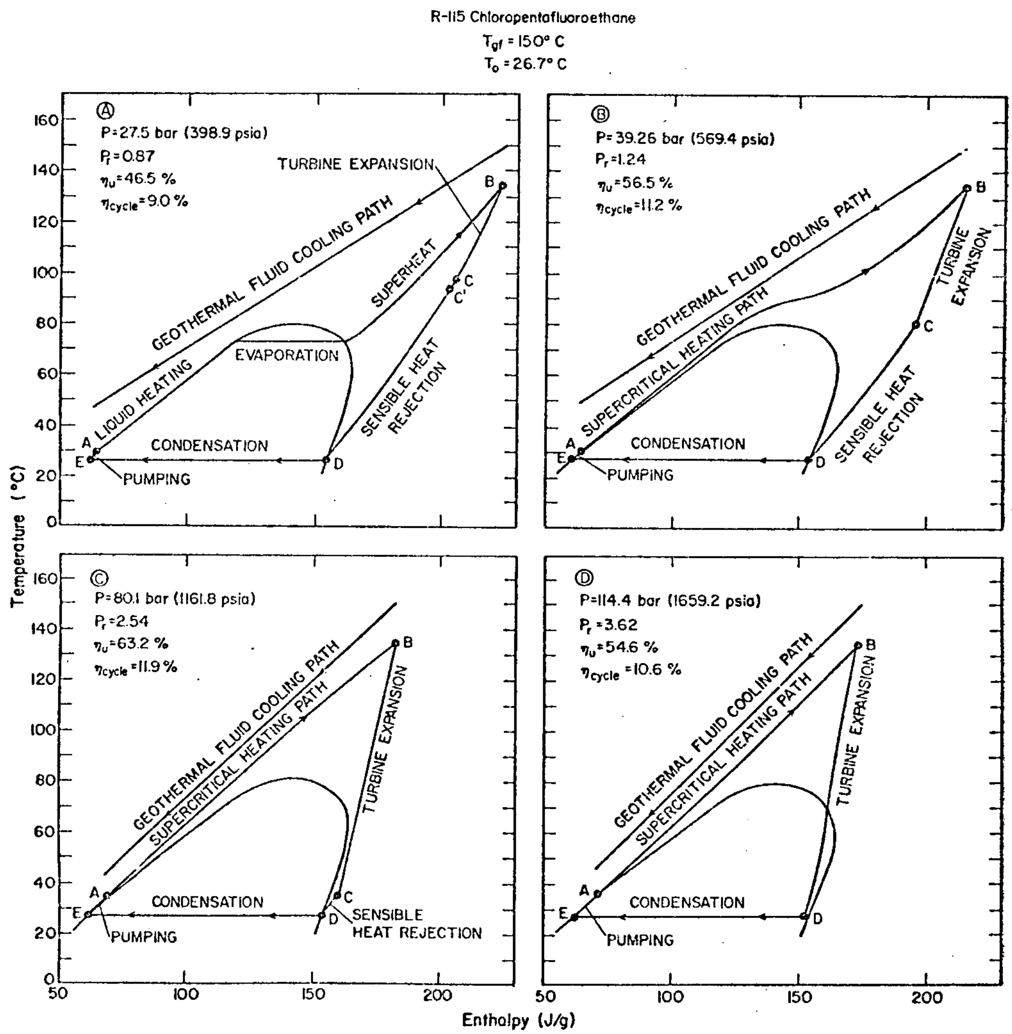

Fig. 3. Approach to a thermodynamically optimized Rankine cycle for R-115 with the effect of cycle pressure shown. (from ref. 1)

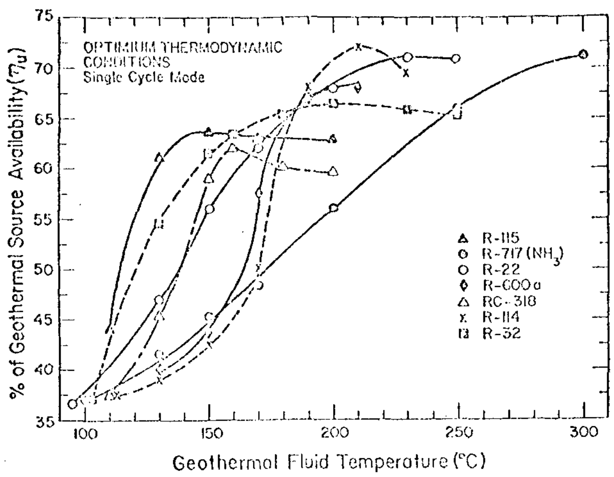

Fig. 4. Geothermal utilization efficiency η_u as a function of geothermal fluid temperature for optimum thermodynamic operating conditions. (from ref. 1)

varied until an optimum was determined at that temperature. One observes a characteristic maximum η_u at a particular resource temperature which is different for each fluid but generally in the range of 60 to 70%.

Because the calculations leading to Fig. 4 are complex and time consuming, a simpler technique for preliminary fluid evaluation was obtained by plotting the temperature of maximum η_u (T^*) minus the fluid's critical temperature (T_c) as a function of the reduced, ideal gas state heat capacity ($C_p^*/R = (\gamma/\gamma-1)$ where $\gamma = C_p/C_v$ and R is the gas constant). The data for the seven fluids studied fit the empirical equation:

$$T^* - T_c = 790/(C_p^*/R) \qquad (1)$$

The quantity ($T^* - T_c$) is effectively the degree of superheat above the critical temperature for optimum performance and can be explained by changes in properties associated with changes in molecular weight [1].

POWER GENERATION ECONOMICS

Generating costs were determined using a factored estimate model based on major component costs, including geothermal wells, heat exchangers, condensers/desuperheaters, turbines and pumps. The total fixed capital investment ϕ for a completed system is expressed as a function of the total equipment ϕ_E and geothermal well cost ϕ_W:

$$\phi = f_I[\phi_E + \Sigma f_i \phi_E] + f_I^*[\phi_W + f_W\phi_W]. \qquad (2)$$

The fractions f_i of ϕ_E represent direct costs associated with the constructional aspects of equipment installation. f_I covers the indirect costs such as engineering fees, contingency, and escalation during construction, f_W the direct costs for piping from the wellhead to the power plant, and f_I^* the indirect costs associated with discovery of the geothermal field including land acquisition and surface exploration. Separate cost correlations were used for each component. Well costs were based on depth, diameter, and type of rock formation. Heat exchanger and condenser/desuperheater costs were calculated from required surface areas and existing manufacturing cost estimates. Turbine costs were based on a model developed by the Barber-Nichols Company of Arvada, Colorado [1]. Turbine design parameters, blade pitch diameter, number of stages and exhaust ends in tandem units, blade tip speed, and stage pressure were used in the turbine cost equation. Pump costs were determined from manufacturers' estimates based on power rating and casing pressure.

In selecting nonaqueous working fluids for power cycle applications, turbine sizes should be small to reduce costs because of the economic tradeoff between the additional heat exchange surface area required for binary-fluid cycles and the much larger and more costly turbines required in flashing systems.

It is important to operate turbines at high efficiency. A similarity analysis of performance shows that turbine efficiency is controlled by two dimensionless numbers involving four parameters: (1) blate pitch diameter; (2) rotational speed; (3) stage enthalpy drop; and (4) volumetric gas flow rate [1]. For operation at maximum turbine efficiency the relationship among these parameters is specified; therefore, turbine sizes and operating conditions and consequently costs can be estimated. For fluid screening purposes, a generalized figure of merit ξ which scales directly with turbine size was developed (see Table I). ξ is expressed as an explicit function of the fluid's molecular weight M, critical pressure P_c, and reduced vapor energy density $h_{fg})_r/v_g)_r^{sat}$. $h_{fg})_r = h_{fg}/RT_c$

is the reduced latent heat and $v_g)_r = v_g/v_c$ the reduced gas specific volume evaluated at the heat rejection temperature T_0.

TABLE I

TURBINE SIZE FIGURE OF MERIT
(taken from ref. 1)
T_0 = 26.7°C

Compound	Formula	$\xi = \frac{\sqrt{M}}{P_c}\left[\frac{v_g)_r^{sat}}{h_{fg})_r}\right]_{T_0}$
R-717	NH_3	0.177
R-32	CH_2F_2	0.223
R-22	$CHClF_2$	0.411
R-115	C_2ClF_5	0.649
R-600a	C_4H_{10}	0.881
RC-318	C_4F_8	1.628
R-114	$C_2Cl_2F_4$	2.246
Water	H_2O	30.71

Economic optima were determined by varying the operating pressures of binary-fluid and flashing cycles for specified geothermal resource temperatures, turbine staging, heat rejection and heat exchange conditions (ΔT's). A 150°C liquid-dominated resource with an R-32 binary-fluid cycle and with a two-stage flashing cycle, and a 250°C dry hot rock resource with an R-717 (ammonia) binary-fluid cycle were considered initially. Well flow rates (45 kg/sec for the 150°C resource and 136 kg/sec for the 250°C resource), geothermal temperature gradients (50-60°C/km), well depths, and reservoir lifetime ($\geq$ 20 yr) were specified. Both production and reinjection well costs were included. Figure 5 presents the results for the cases described above.

Assuming that the power plant design has been optimized, generating costs are primarily controlled by three main variables: (1) fluid temperatures, which affect cycle efficiency and thus equipment costs; (2) well flow rates, which affect well efficiency (kW/well); and (3) geothermal temperature gradients, which affect well depth and cost for a given geothermal fluid temperature.

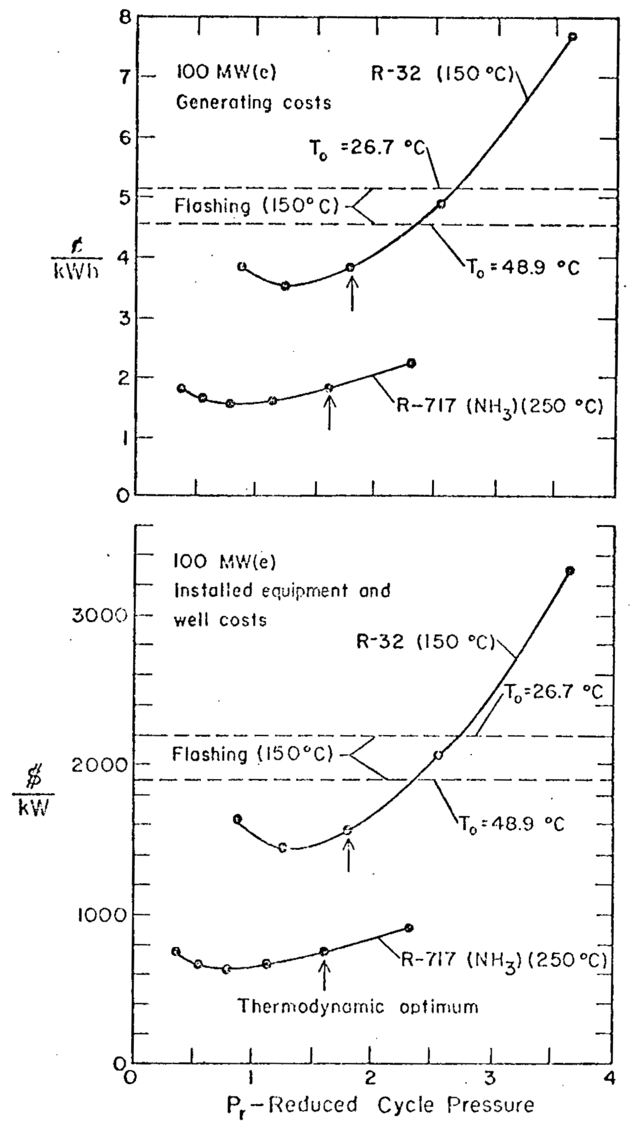

Fig. 5. Approach to economic optimum cycle conditions. Cost per kWh and cost per installed kW as a function of reduced cycle pressure P_r. A 250°C dry hot rock geothermal resource with an ammonia Rankine cycle and a 150°C liquid-dominated resource with a R-32 Rankine cycle and a flashing cycle were studied. (from ref. 1)

For a defined set of resource and power plant conditions there should exist an optimum depth (or temperature) for drilling.

POWER PRODUCTION COST COMPARISON

In the United States only a small fraction (~0.1%) of total generating capacity is currently produced from geothermal resources. One major factor that will influence the rate of geothermal power development is the cost of more conventional energy sources such as oil, coal, gas, and nuclear. Table 2 presents estimates for geothermal, fossil-fuel, and nuclear electric generating costs. Capital equipment, fuel, and operating and maintenance costs are included. Although potentially competitive with fossil-fuel fired or nuclear plants, geothermal installations are very capital intensive because a total investment in both surface equipment and wells (the "fuel" source) is required. Geothermal energy for nonelectric applications might might even be more economically competitive because the heat is being used directly, thereby avoiding conversion efficiency losses. For example, a 200°C natural hydrothermal resource with a geothermal gradient of ∿50°C/km could produce heat at costs of 0.1 - 0.48¢/kWh (as heat) versus ∿0.7¢/kwh from oil at $12 bbl (assuming minimal fluid transportation costs).

TABLE II

Comparison of Geothermal, Nuclear, and Fossil-Fuel Estimated Generating Costs (from ref. 1)
(1976 Dollars)

Resource Type	Installed Equipment Costs ($/kW)	Equipment Cost (¢/kWh)[a]	Operating and Maintenance (¢/kWh)	Well or Fuel Cost (¢/kWh)	Total Generating Cost (¢/kWh)
Direct flashing[b]	300-600	0.68-1.37	0.13	0.80-2.80	1.61-4.30
Binary-fluid cycles[b]	400-700	0.91-1.60	0.13	0.53-2.45	1.57-4.18
Nuclear[d]	≥800	≥1.83	0.13	0.30	≥2.26
Fossil fuel--oil[e]	400-600[c]	0.91-1.37	0.13	2.0($12/bbl)	3.04-3.50
Fossil fuel--coal[e]	400-600[c]	0.91-1.37	0.13	1.0($25/ton)	2.04-2.50

[a]17% annual fixed charge rate.
85% (7446 hr/yr) load factor.

[b]150-200°C resources.
$\dot{m}_w$ = 100-300 lb/sec.

[c]Higher costs reflect stringent environmental control systems.

[d]Plant startup in 1984.

[e]Plant startup in 1980.

REFERENCES

1. Milora, S. L., and Tester, J. W., Geothermal Energy as a Source of Electric Power--Thermodynamic and Economic Design Criteria, MIT Press, Cambridge, Massachusetts, 1976.

2. Tester, J. W., "Geothermal Energy from Dry Hot Rock Reservoirs," to be presented at the AIChE 81st National Meeting, Kansas City, Missouri, April 11-14, 1976.

3. Kruger, P., and Otte, C., eds., Geothermal Energy, Stanford University Press, Stanford, California, 1973.

4. Anderson, J. H., "The Vapor-Turbine Cycle for Geothermal Power Production," Geothermal Energy, Stanford University Press, Stanford, California, 1973.

5. Aronson, D., "Binary Cycle for Power Generation," Proceedings of American Power Conference, pg. 23, 1961.

6. Chou, J. C. S., "Regenerative Vapor-Turbine Cycle for Geothermal Power Plant," Geothermal Energy, Vol. 2, pg. 21, 1973.

7. Cortez, D. H., Holt, B., and Hutchinson, A. J. L., "Advanced Binary Cycles for Geothermal Power Generation," Energy Sources, Vol. 1, No.1, pg. 74, 1973.

8. Jonsson, V. K., Taylor, A. J., and Charmichael, A. D., "Optimization of Geothermal Power Plant by Use of Freon Vapour Cycle," Timarit-VFI, pg. 2, 1969.

9. White, D. E., and Williams, D. L., eds., "Assessment of Geothermal Resources of the United States-1975," Geological Survey Circular, No. 726, U. S. Geological Survey, Reston, Virginia, 1975.

10. Beall, S. E., and Samuels, G., "The Use of Warm Water for Heating and Cooling Plant and Animal Enclosures," Oak Ridge National Laboratory Report ORNL-TM-3381, Oak Ridge, Tennessee, 1971.

11. Beall, S. E., "Waste Heat Uses Cut Thermal Pollution," Mechanical Engineering, Vol. 93, pg. 15, 1971.

12. Bodvarsson, G., Boersma, L., et al, "Systems Study for the Use of Geothermal Energies in the Pacific Northwest," Oregon State University Report RLO-2227-T19-1, Corvallis, Oregon, 1974.

13. Reistad, G. M., "Analysis of Potential Nonelectrical Applications of Geothermal Energy and Their Place in the National Economy," Lawrence Livermore Laboratory Report UCRL-51747, Livermore, California, Feb. 1975.

14. Smith, M. C., Aamodt, R. L., Potter, R. M., and Brown, D. W., "Man-Made Geothermal Reservoirs," Second United Nations Geothermal Energy Symposium, San Francisco, California, May, 1975.