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A Comparison of Methods for Electric Power Generation from Geothermal Hot Water Deposits

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A Comparison of Methods for Electric Power Generation from Geothermal Hot Water Deposits

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ABSTRACT

A comparison is made of three energy conversion concepts: the Flashed Steam System, the Binary Cycle System, and the Total Flow System. These systems are analyzed for typical wellhead characteristics found in hot-water geothermal resources. The Total Flow concept passes the steam and water mixture directly through convergent-divergent nozzles and an impulse turbine. The Total Flow concept has the potential for 60% greater efficiency than either of the other systems.

I. INTRODUCTION

The increasing need for clean, non-fossil energy sources is focusing attention on use of geothermal energy for electrical power generation. Geothermal energy is defined here as thermal energy stored in underground deposits of superheated steam, hot water, and hot dry rock. Although dry steam deposits, such as those at The Geysers, California, are technologically the easiest to exploit since the steam can be used directly to drive a steam turbine, their occurrence is rare. They are estimated,[1] to be only one-twentieth as common as hot water deposits. On the other hand, there is an immense amount of energy stored in deep, dry hot rock, but no technology yet exists for recovery of this energy.

It is generally believed, however, that the water-dominated geothermal systems constitute an energy resource that could be developed with moderate extensions of existing technologies to produce a significant supply of clean, low-cost electrical energy. Temperatures of the water deposits of primary interest vary from 300°F to more than 600°F with dissolved solids content ranging from less than 0.1% to over 25%, respectively. The magnitudes and locations of these deposits are not yet well known, but they generally occur in the western states, and are at depths varying from about 3,000 to 10,000 feet.

Several factors contribute to the current interest in conversion machines or systems designed uniquely for the utilization of hot-water geothermal resources.

1 Hot-water resources are more abundant than steam resources.

2 Economic considerations encourage development of energy conversion machines with highest efficiencies in order to minimize the number of wells per unit of electrical power output.

3 The range of chemical conditions of the hot-water resources require special systems. In some cases, utilization of hot brines is not feasible by present conversion methods.

In the following discussion we will compare three conversion concepts: the Flashed Steam System, the Binary Cycle System, and the Total Flow System. Fundamental reasons for interest in the Total Flow concept will be discussed from a thermodynamic point of view. No attempt will be made to present detailed designs since neither the Binary Cycle nor the Total Flow method have yet evolved to the commercial stage of development.

II. WELLHEAD CHARACTERISTICS

It is useful to first briefly describe the characteristics of geothermal

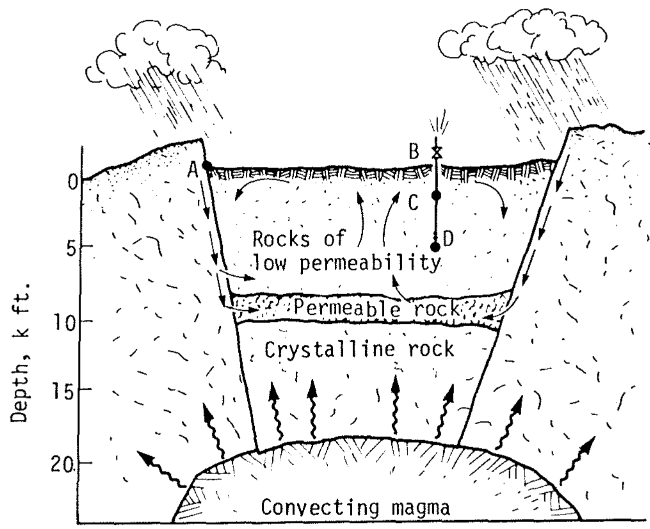


Figure 1

hot-water systems. Figure 1 shows one model of a hot-water geothermal system. Cold water is supplied by surface runoff, such as at point A, where it percolates downward through pervious rock to be heated by conduction from deep impervious hot rock overlying areas of magmatic intrusions. The hot water rises, and convection cells are set up giving rise to slow circulation of the water as shown. (A detailed discussion of these systems is provided by Muffler and White[2] and White.[3]) In some cases surface manifestations such as hot springs result from fissures which allow the water to escape upward.

The water in these reservoirs is subject to its own hydrostatic pressure which is greater than the vapor pressure corresponding to the temperature at depth. If a well is drilled into this system as shown by BCD, once initiated, it will flow unaided as the rising fluid partially vaporizes, reducing the average density and thus the pressure in the well. Vaporization begins at point C where the pressure is slightly less than the vapor pressure of the hot water. The flow up a geothermal hot-water well, then, consists of liquid flow from D to C, and flashing two-phase flow, vapor (steam) and water mixture, from C to the wellhead B.

The flow rate is governed by well depth, diameter, pipe friction, reservoir characteristics, in-situ water temperature, and wellhead pressure. An approximate evaluation of well flow is relatively straightforward by numerical solution of the momentum and continuity equations for steady flow with the assumptions of no lateral heat transfer, and an isentropic expansion of fluid from C to B, using the thermodynamic properties of pure water, i.e., Steam Table data. Since the detailed methodology of this calculation is already given,[4] we will report only the results of these calculations of flow from hot-water reservoirs with different in-situ temperatures as shown in Table I. The wellhead characteristics are temperature, pressure, quality (mass of vapor/total mass), enthalpy and flow rates for two different friction factors; all for

Table I. Calculated geothermal well flow characteristics for self pumping mode of production.[1]

Reservoir Condition Temp. °F	Reservoir Condition Enthalpy Btu/lb	Wellhead temperature °F	Wellhead pressure psia	Vapor fraction %	Wellhead enthalpy Btu/lb	Flow rate[2] 10^6 lb/hr/well f=0.02	Flow rate[2] 10^6 lb/hr/well f=0.04	Thermal energy extraction rate[2] MW_t/well
350	321.8	281	50	7.4	318.2	0.32	0.26	17.5
400	375.0	326	98	8.3	371.0	.42	.35	29.0
500	488.0	390	220	13.7	478.7	.56	.44	50.3
572	577.2	434	360	18.9	562.5	0.63	0.47	65.4

NOTE:

1. Well depth = 5000 ft
   Production Casing = 7-5/8 in. o.d., 6.77 in. i.d.; Area = 0.25 ft^2
   f = Moody friction factor
   Calculations based on thermodynamic properties of pure water
2. With reference to saturated liquid at 120°F, f = 0.04
   Flow rate is for maximum thermal energy rate

the maximum thermal energy extraction rate with respect to a sink temperature of 120°F.

Note that only 1-3% of the fluid energy content is lost in the vertical flow from the flashpoint to the surface. Also, the flow rate is strongly influenced by pipe friction. Actual friction values are not known, but we have compared these calculated conditions with flow data from existing wells, and found that reasonable agreement is obtained for f = .02 to f = .04. Hence, the calculated wellhead conditions given in Table I will be used here for purposes of comparison of the performance of conversion systems.

III. CONVERSION SYSTEMS

1. The Flashed Steam System (Fig. 2A)

In the early 1960's New Zealand pioneered the recovery and conversion of energy in hot-water deposits by using the so-called "Flashed Steam Method." The wellhead product is fed into a flash separator where the vapor fraction is increased by an isenthalpic expansion to a lower pressure. The steam is then used to drive a standard axial-flow multistage steam turbine for electric power generation. Overall efficiency is increased by the use of condensing turbines wherever the proportion of noncondensible gases is sufficiently low. The separated brine fraction is discarded. The overall thermal efficiency is low since, at best, only about 10% of the thermal energy at the wellhead is converted in a single-flash system. The efficiency can be increased somewhat, by using additional stages of separation, but this introduces additional complexity in system design.

The Flashed Steam System is simple, it utilizes standard machinery, and is workable if the salinity of the water is low ($\lesssim$3% dissolved solids) since carryover of salts into the vapor can cause corrosion, erosion, or scaling of turbine components. Such plants are currently operational in New Zealand, Japan, and Mexico.

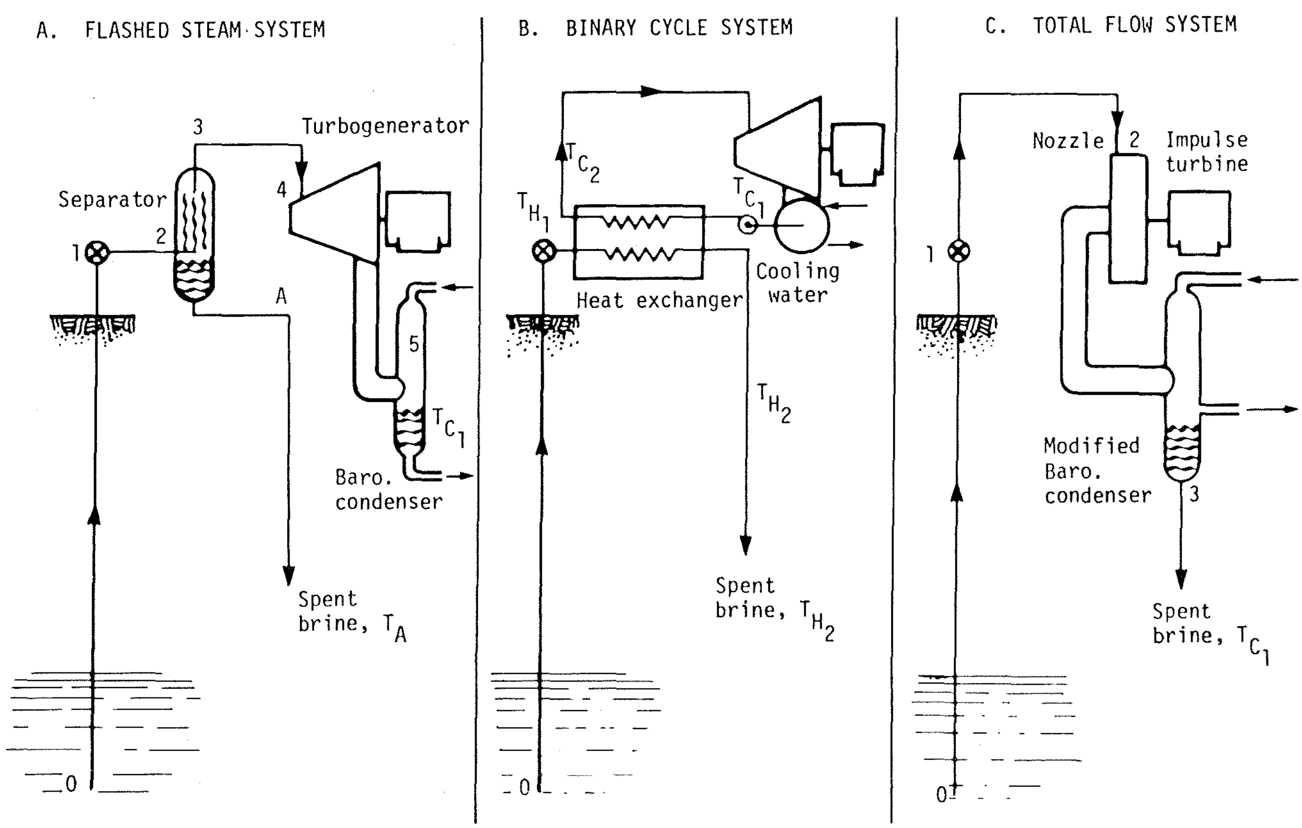


Figure 2

2. The Binary Cycle Concept (Fig. 2B)

In an attempt to utilize brines containing larger amounts of dissolved solids or non-condensible gases, or brines at relatively lower temperatures, the Binary Cycle concept has recently been introduced as an alternate scheme for conversion of hot-water geothermal energy. This method requires use of a heat exchanger at the wellhead to transfer the internal energy of the brine to a clean secondary fluid (e.g., isobutane, Freon*). The vaporized secondary fluid operates a turbogenerator then is condensed and pumped back through the heat exchanger. The use of a secondary fluid protects the turbine from the brine, but scale formation in the brine side of the exchanger will be a major problem. Further, as will be shown later, the brine outlet temperature must be high to optimize energy transfer per unit of exchanger area. Hence, this results in about as much energy discarded with the spent brine as in the Flashed Steam System giving no significant advantages in thermal efficiency. To our knowledge, there is only one such plant operational. It is located on the Kamchatka Peninsula in Russia and produces 680 kW from a Freon binary system operating from shallow wells producing low-saline water at about 180°F. (For a complete summary of worldwide use of geothermal energy the reader is referred to an excellent paper by Koenig.[5])

3. The Total Flow Concept (Fig. 2C)

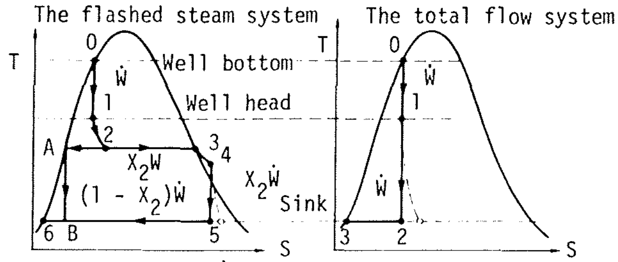
To date, electric generation from geothermal energy almost exclusively involves low-pressure, axial-flow multistage steam turbines which evolved from the requirements of fossil-fueled power plants. Generally, the adaptations of these devices to the geothermal steam environment has been successful, but application is limited to relatively clean, high quality, steam. The major considerations are materials, scale deposition, and efficiency.

In order to introduce greater flexibility in the utilization of geothermal fluids of widely varying temperatures and salinities, the Total Flow method has been proposed and is under development at the Lawrence Livermore Laboratory (LLL). In this concept, the entire wellhead product is fed directly into an impulse turbine. This involves expansion of the fluid through converging-diverging nozzles to convert the enthalpy of the high-temperature fluid into kinetic energy in the form of low-temperature, high-velocity streams of fluid. These jets are used to drive a corrosion-resistant impulse turbine of special design, but similar in principle to hydroelectric devices. The advantages are that since the entire pressure (and temperature) drop occurs in the nozzles, scaling may be easier to control; the turbine wheel operating at relatively low temperature can more easily be made resistant to corrosion and erosion. Along with the inherent mechanical and thermodynamic simplicity, this system has the potential for achieving higher thermal efficiency than either the Flashed Steam or Binary methods.

IV. COMPARISON OF CONVERSION SYSTEMS

The above three systems are shown schematically in Fig. 2 and a cycle diagram comparing the Flashed Steam and Total Flow methods on the Temperature-Entropy plane is shown in Fig. 3. From Fig. 3 it is immediately obvious that direct expansion of the total wellhead product is thermodynamically the simplest and provides an upper bound on cycle efficiency regardless of the number of stages of separation used in the Flashed Steam System. There will always be some useful energy discarded with the separated liquid as shown, for example, by the path A-B. For the case of the single-flash system shown, the power output given in Table II was calculated for different separation pressures until a maximum power output was obtained. (In all calculations a sink temperature of 120°F was used.) The calculation was repeated for each case shown in Table I where columns 3, 4, and 5 identify the wellhead state point in Fig. 3.

* Reference to a company or product name does not imply approval or recommendation of the product by the University of California or the U.S. Atomic Energy Commission to the exclusion of others that may be suitable.



Ideal power out = $X_2\dot{W}(h_4-h_5)$

Power lost in brine = $(1-X_2)\dot{W}(h_A-h_5)$

Ideal cycle efficiency = $X_2(h_4-h_5)/h_1-h_6$

Ideal power out = $\dot{W}(h_1-h_2)$

Power lost = 0

Ideal cycle efficiency = h_1-h_2/h_1-h_3

Figure 3

The calculation for the Total Flow method is also done for each case. In both systems, it is assumed that the engine efficiencies are 70%, i.e., the result of turbine inefficiencies represented by the dashed expansion line. The "actual" power outputs from these two systems are plotted in Fig. 4. It should be noted that we have deliberately neglected pumping requirements, and other peripheral energy costs associated with plant operation. We have, however, estimated the losses resulting from the pressure drop to the Flashed Steam Turbine (path 3-4) by charging a 20% pressure drop along the saturated vapor line.

In both systems condensing is accomplished with a barometric direct-contact condenser. For high-salinity waters, however, the condenser for the Total Flow System will require modification to separate the brine fraction from the condensed vapor fraction in order to prevent fouling of the cooling system. In either system, there is generally sufficient steam condensed to make up for evaporation from a cooling tower, hence, no continual source of cooling water is required.

The Binary Cycle System calculation requires special treatment. Rather than carry out a detailed cycle analysis by assuming a variety of secondary fluids, we will instead take a more generalized approach developed by Higgins[6] which will yield an upper bound for plant performance. This will give an optimistic result, but will be of sufficient value for purposes of comparison relative to the Total Flow method.

For any heat exchanger, the energy transferred, q, to the secondary fluid is given by[7]

$$q = \dot{W}F_G\ UA\Delta T_m \tag{1}$$

where U is the overall heat transfer coefficient, assumed here to be constant.

A is the heat exchanger area.

F_G is the correction factor relating the actual mean temperature drop to the logarithmic value. It will be assumed here to be unity.

ΔT_m is the logarithmic mean temperature drop.

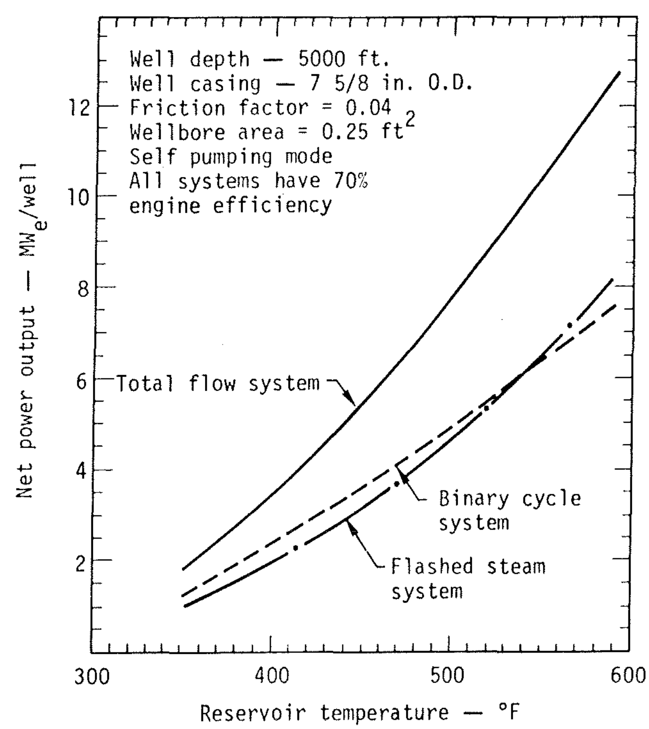


Figure 4

$$\Delta T_m = \frac{\left(T_{H_1} - T_{C_2}\right) - \left(T_{H_2} - T_{C_1}\right)}{\ln\left(\frac{T_{H_1} - T_{C_2}}{T_{H_2} - T_{C_1}}\right)} \qquad (2)$$

where T_{H_1}, T_{H_2}, T_{C_2}, T_{C_1}, are the terminal temperatures on the exchanger. (See Fig. 2B.)

The exchanger can be designed to make T_{C_2}, the secondary fluid peak temperature, as close to T_{H_1} as desired, but only at the expense of increasing the exchanger area, A. Since heat exchanger costs vary nearly linearly with A, then the cost of energy transfer varies with q/A. Since q varies with T_{H_2}, and T_{C_1} is fixed by ambient conditions and is here chosen arbitrarily as 120°F, a trial and error solution for exchanger operation is required in which T_{H_2} is chosen and T_{C_2} calculated. Further, the optimum power output from the secondary cycle occurs when the Carnot efficiency, C, is maximized. Hence, the entire system is optimized when the function

$$\frac{q}{Q} \cdot \frac{q}{A} \cdot e_c = \left(\frac{h_1 - h_2}{h_1 - h_f}\right) U \, \Delta T_m \left(\frac{T_{C_2} - T_{C_1}}{T_{C_2}}\right) \qquad (3)$$

is maximized, where q is the energy transferred from the brine, $h_1 - h_2$ and $Q = h_1 - h_f$ is the energy at the wellhead relative to the liquid at the sink temperature, 120°F.

This optimization assumes the cost of energy to be linearly related to the area of the heat exchanger. A more realistic analysis requires detailed information about heat exchanger systems suitable for geothermal applications.

Using the thermodynamic properties of pure water (Steam Table data), trial and error solution of (3) was carried out to give T_{C_2} and T_{H_2} for each wellhead condition given in Table I. The power output was then obtained from

$$P = \dot{w}(h_1 - h_2)e \qquad (4)$$

where e is the engine efficiency of the secondary loop. This was taken arbitrarily at 70% which is optimistic since it implies that a plant can operate at 70% of the Carnot efficiency for the secondary system. Nevertheless, the calculations of (4) for each wellhead condition are sufficiently valid for our purposes here, and are plotted in Fig. 4 which gives the comparison of systems.

These curves, of course, represent a comparison made by simple cycle analysis, and do not include effects of energy costs of plant operation such as pumping, leakages, etc. which are probably comparable for each system. The advantage of the Total Flow method is clearly seen since these results show that up to 60% more power output may be possible. This is especially important for the lower temperature reservoirs since the capital investment for the larger number of wells required per unit of electrical power is a larger fraction of total initial costs.

It is of interest to present further details for one case — namely, the 572°F reservoir temperature case. Table II gives additional information relative to the energy balance for each system operating from a single well. Here we have computed the amounts of power discarded in the spent brine by assuming the fluid from the separator or exchanger can be isentropically expanded in a Total Flow device. Note that the Binary Cycle System discards less available energy since the spent brine leaves the exchanger at 279°F as compared to 338°F from the separator. The losses in the Flashed Steam System, are due to the irreversibility in the separation process, 1-2, and the 20% pressure drop from 3-4. For the Total Flow System, the spent brine is rejected at the sink temperature (120°F) and no inherent system losses occur in the cycle. As mentioned earlier, each system is assumed to have an engine efficiency of 70% in order to calculate the actual power output. The cooling duty is less for the Total Flow System because of the higher efficiency, and because only the vapor fraction requires condensation. The Binary Cycle System, however, requires a separate continual source of cooling water for condensation of the secondary working fluid. This may be a severe disadvantage in arid regions. For further information regarding the Binary Cycle System the reader is referred to Refs. 8 and 9.

V. THE TOTAL FLOW SYSTEM

1. Turbine Requirements

Starting with the thermodynamic principles discussed above, we will discuss the turbine characteristics needed for effective operation in geothermal applications. Several factors have combined to lead to an impulse turbine utilizing a converging-diverging nozzle for conversion of the fluid enthalpy to kinetic energy, and possibly either a radial-inflow or tangential flow wheel to convert this energy to shaft work. The geothermal environment encourages design simplicity to allow use of corrosion and erosion resistant materials, flexibility for incorporation of scale control features, and a means to minimize capital costs. Hence, a single stage impulse turbine appears to be most adaptable to the wide variety of thermodynamic and physical conditions inherent in geothermal fluids. Currently, the tangential-flow turbine appears most promising. It has the potential for high efficiency since it is relatively insensitive to the effects of slip (velocity difference between liquid and vapor phases), and offers considerable design flexibility. One such concept currently under study is shown in Fig. 5 which is a schematic of a tangential flow turbine being designed for testing.

The efficiency of such devices is dependent upon many factors which interact in a very complex manner. We can, however, define the Total Flow turbine requirements in a general way. With reference to Fig. 3 the ideal output for the Total Flow system is obtained from the isentropic enthalpy drop, $(h_1 - h_2)_s$. Since this occurs entirely in the nozzle, the nozzle exit

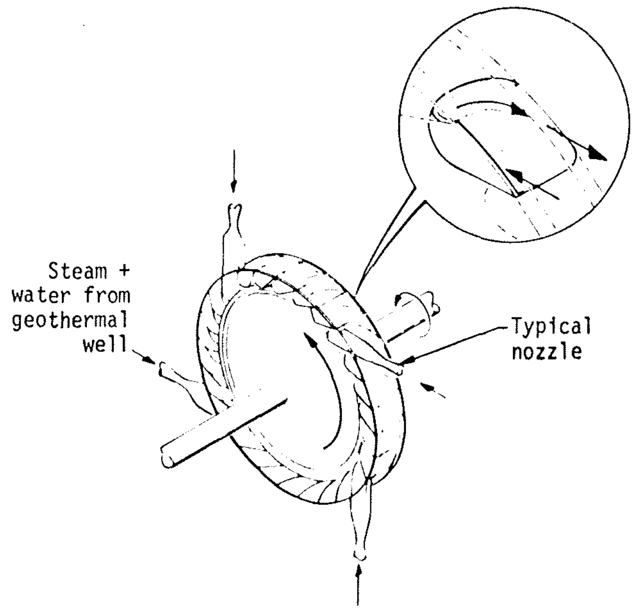


Figure 5

velocity is given by (neglecting inlet velocity)

$$V^2 = 2g_c J \eta^2 (h_1 - h_2)_s \qquad (5)$$

where, η, is the nozzle velocity coefficient. Hence, η^2 is the nozzle efficiency for conversion of fluid enthalpy to kinetic energy. If e_w is the wheel efficiency, which is a composite measure of the effects of blading geometry, turbulence, fluid friction, entrance and exit losses, fanning losses, etc., then the actual power output from the Total Flow turbine is

$$P = \dot{w}(h_1 - h_2)_s \eta^2 e_w \qquad (6)$$

This can be compared with the power output from the Flashed Steam system (see Fig. 3) by forming the ratio

$$\frac{P_{TF}}{P_{FS}} = \frac{(h_1 - h_2)_s \eta^2 e_w}{x_2 (h_4 - h_5)_s e} \qquad (7)$$

Table II. Comparison of systems for conversion of energy from a 572°F reservoir flowing at 0.47×10^6 lb/hr up a 7-5/8 in. well.

	Flashed Steam System	Binary Cycle System	Total Flow System
Thermal energy extraction rate, MW_t	65.4	65.4	65.4
Ideal power output, MW_e	10.5	10.2	16.5
Discarded power in spent brine, MW_e	3.3	2.6	0
Irreversibility losses, MW_e	2.7	3.7	0
Actual power output (@ 70%), MW_e	7.4	7.1	11.6
Overall efficiency,* %	11.3	10.9	17.7
Cooling duty - 10^6 Btu/hr/100 MW_e	1700	1600	548
Min. No. of prod. wells/100 MW_e	14	14	9

*Separator conditions: p = 115 psia, (T = 338°F)
Exchanger conditions: T_{H_1} = 434°F; T_{H_2} = 279°F; T_{C_2} = 313°F

Sink temperature: T_{C_1} = 120°F

where e is the engine efficiency of the Flashed Steam system conversion machinery. Using the data for the special case shown in Table II, this ratio reduces to

$$\frac{P_{TF}}{P_{FS}} = \frac{1.6\ \eta^2 e_w}{e} \qquad (8)$$

This is plotted in Fig. 6 with e taken as 70%.

The results of this calculation show that for 60% more power output, the nozzle coefficient must be approximately 0.9 with a wheel efficiency of about 90%. Hence, the research goal for development of a Total Flow turbine is to achieve these results. For hydraulic operation, impulse turbines typically operate with 90% efficiencies. Although very little work has been done to design such devices to operate efficiently from two-phase flow, we believe that, in principle, high wheel efficiencies can be achieved.

The expansion of fluids and gases through nozzles has been the subject of intensive study with literally hundreds of papers written on the subject. This work

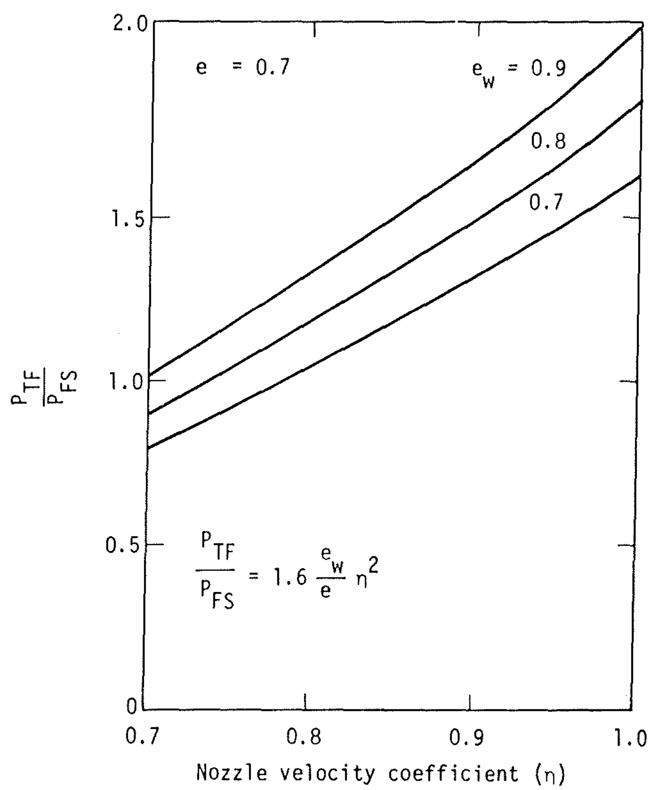


Fig. 6. Comparison of the actual power output of the total flow system with the Flashed Steam system.

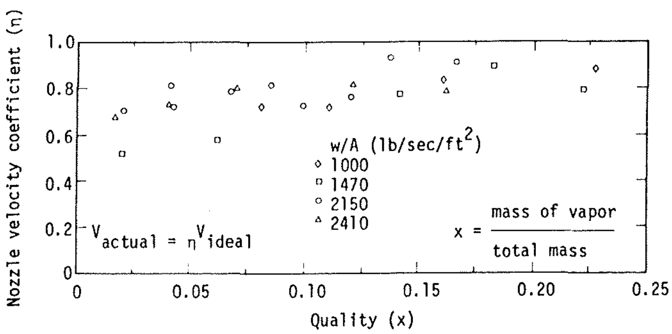


Fig. 7. Performance of two-phase flow through a converging-diverging nozzle.

almost exclusively involved the expansion of superheated and high-quality steam, liquids, and gases, with very little effort devoted to low quality (<20%) steam, or two-phase flow. One of the most widely quoted studies is that of Goodenough[10] in 1927 who concluded that nozzle coefficients dropped to 0.9 when the quality dropped to 80%, i.e., 20% moisture. He presented data to substantiate this, and constructed an analytical expression which predicts a continually decreasing nozzle coefficient with decreasing quality. This work has now become the "conventional wisdom" and is used in many texts and reference works to conclude that two-phase flow of low-quality steam through nozzles will not efficiently convert thermal energy to kinetic energy. Based on this evidence alone, it would appear that the Total Flow system for geothermal applications will not function efficiently.

Perhaps because of the trend toward higher and higher temperatures of the working fluids for steam power systems, very little further work was done beyond that of Goodenough to investigate the details of flow of low-quality steam through nozzles. In 1962, however, additional data was gathered[11-13] for nozzle coefficients with two-phase steam flow from 0-20% quality. These data are shown in Fig. 7 which indicates that nozzle coefficients above 0.8 were obtained for flow rates, qualities and nozzle inlet pressures typical of conditions expected in geothermal applications. The backpressure, however, was atmospheric. While these are the only such data available, the results are encouraging since those authors emphasized that no attempts were made to optimize the nozzle design. Even though the test conditions were grossly different, the results of Goodenough[10] and those of Maneely[12] are plotted on the same graph as shown in Fig. 8. The purpose here is to illustrate that extrapolation of Goodenough's result to low quality conditions gives a result which differs markedly from the recent data, and that conclusions based solely on the "conventional wisdom" could be misleading.

Obviously, additional work is needed to completely understand nozzle performance. It will be important to carry out extensive tests to determine nozzle coefficients as a function of inlet quality with backpressures varying from atmospheric down to typical condenser pressures in the region of 3 in. Hg. We plan to do this work along with the analytical work needed to develop methods for optimization of nozzle geometries. For the present, however, we are left with the existing evidence to support our conclusion that achievement of nozzle coefficients of 0.9 is within the realm of possibility and that the Total Flow system, as we have defined it, can become a significant advance in geothermal technology in terms of increased conversion efficiencies.

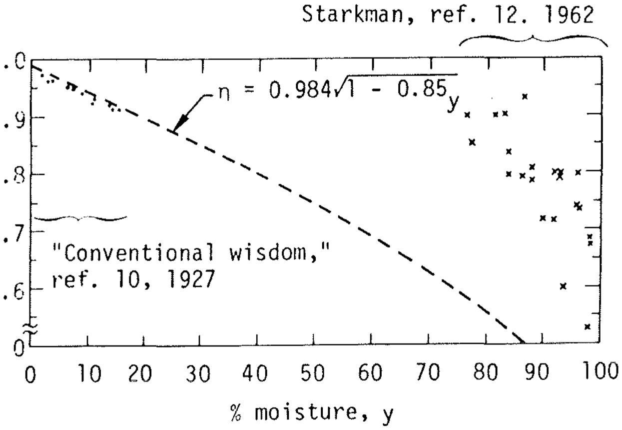
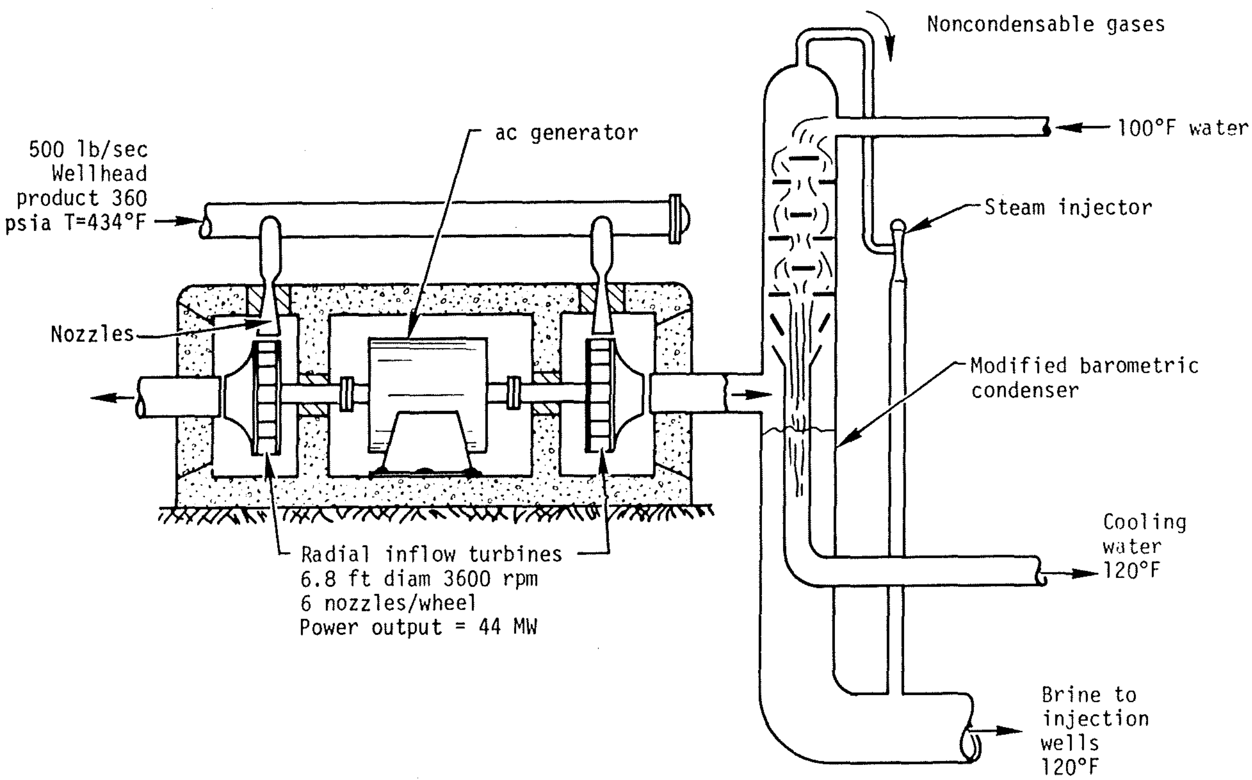


Figure 8



Schematic of a total flow system.

Figure 9

2. A System Concept

For sake of completeness, it is important to close with a brief discussion of the nature of a complete system for electrical power generation. One possible configuration is shown schematically in Fig. 9. This is intended only for illustrative purposes and is presented only as a concept designed around the parameters listed in Table II. The system chosen here consists of a single generator operated by two radial inflow turbines with a wheel efficiency, e_w, of 90% driven by six nozzles per wheel, each with a nozzle coefficient η of 0.9. The fluid flow of 500 lb/sec will require four 7-5/8 in. wells with flow rates shown in Table I. The power output, with $\eta^2 e_w = 0.72$ will be 44 MW_e. The entire system is referenced to a condensing temperature of 120°F or a turbine backpressure of about 3.5 in. Hg. The quality of the two-phase flow at the turbine outlet is about 35%, and condensation of the vapor fraction can be accomplished with barometric condensers.

For high-salinity fluids, however, the condenser can be modified as shown. The turbine outflow can be introduced tangentially to the standpipe, separating the vapor to be condensed by the flow of cooling water in the loop shown. The separated brine flows down through one of the two barometric legs for disposal. As mentioned earlier, calculations indicate that the amount of water evaporated in the cooling system is very nearly equal to the condensed vapor. Depending on atmospheric conditions, there will either be a small net gain or loss in cooling water inventory. Hence, the system, like the Flashed Steam case, does not require a continual flow of cooling water as does the Binary Cycle system. A comparison of the cooling duties is given in Table II which illustrates that cooling requirements are less for the Total Flow system than for either of the other systems per unit of electrical power output.

For the hypersaline brines containing up to 26% of dissolved solids, the flow through the turbine will likely be a slurry of scale particles. Consequently, an erosion-resistant turbine will be the primary problem in terms of materials technology. A wide variety of materials can be considered because the turbine is operating at relatively low temperatures. A major difficulty, however, in the system is the management of the solids in disposal of the brine. Although only the mechanical aspects of the Total Flow System are emphasized herein, considerable effort is underway at the Lawrence Livermore Laboratory to address all of the technical problems related to utilization of geothermal energy.

VI. CONCLUSIONS

It is clear that, in principle, the Total Flow System represents the thermodynamic upper bound on efficiency for direct conversion of the energy in the hot water deposits. Consequently, if developed successfully, it will be a significant advance in geothermal technology. This has particular economic significance since the number of wells per unit of electrical power output can be greatly reduced. This is illustrated by the calculations shown in Table II where the number of wells is reduced directly by the ratio of overall efficiencies, i.e., 60% more power output from the Total Flow System. Figure 4 illustrates that this factor applies over a range of reservoir temperatures. Since the thermal energy extraction rate from the wells drops drastically for the lower temperature reservoirs (Table I), this saving in capital investment becomes a very important factor when considering development of the low temperature water deposits. Even though the Total Flow System was originally conceived as an attractive method for utilization of the high temperature/high saline brines, it will become a very important method for all types of liquid-dominated geothermal systems. From a purely mechanical engineering point of view, the Total Flow System is, in our opinion, the most promising method for development of these resources.

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