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Electrical Generating Capacities of Geothermal Slim Holes

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J. W. Pritchett

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Work Performed Under Contract No. DE-FG07-96ID13455

**For
U.S. Department of Energy
Assistant Secretary for
Energy Efficiency and Renewable Energy
Washington, DC**

**By
Maxwell Technologies Systems Division, Inc.
San Diego, CA**

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Table of Contents

Section	Page
Executive Summary	iii
1. Introduction.....	1-1
1.1 Background	1-1
1.2 Free Parameters	1-4
1.3 Depth of the Piezometric Surface.....	1-5
1.4 Practical Considerations Regarding Well Diameter	1-11
2. Characteristics of Self-Discharging Geothermal Wells	2-1
2.1 Modeling Flow in the Reservoir	2-1
2.2 Modeling Fluid and Heat flow Up the Well	2-4
2.3 Modeling Wellhead Discharge Characteristics	2-9
3. Characteristics of Geothermal Wells With Downhole Pumps	3-1
3.1 Application of Downhole Pumps to Geothermal Wells	3-1
3.2 Modeling Downhole Pump Performance	3-4
3.3 Approximate Analysis of Design Constraint Effects	3-13
3.4 Downhole Pump Electrical Power Consumption	3-15
3.5 Modeling Well Performance Using Downhole Pumps	3-15
3.6 Performance of Geothermal Wells Equipped With Downhole Pumps.....	3-16
4. Single-Flash Steam Plants	4-1
4.1 Utility of Single-Flash Steam Plants	4-1
4.2 Steam Expansion Through the Turbine	4-6
4.3 Plant Performance	4-8
5. Double-Flash Steam Plants	5-1
5.1 Types of Double-Flash Plants	5-1
5.2 Modeling Double-Flash Steam Plants	5-6

Section		Page
6.	Binary Plants and Steam/Binary Hybrids	6-1
	6.1 Binary Geothermal Plants Using Downhole Pumps and Recuperation	6-1
	6.2 Performance of Single-Hydrocarbon Off-Grid Binary Plants	6-4
	6.3 Self-Discharging Wells — Binary Plants With Cold-Water Recirculation	6-5
	6.4 Flash Steam/Binary Cycle Hybrid Plants	6-9
7.	Optimizing System Performance	7-1
	7.1 Wells With Downhole Pumps and Binary Plants	7-1
	7.2 Wells With Downhole Pumps and Wellhead Separators	7-1
	7.3 Self-Discharging Wells	7-28
8.	Electrical Capacities of Geothermal Wells With Wellhead Generators	8-1
	8.1 Contents of the Appendices	8-1
	8.2 Influence of Well Diameter on Electrical Capacity	8-1
	8.3 Effect of Power Plant Design on Electrical Capacity	8-2
	8.4 Utility of Downhole Pumps for Various Plant Designs	8-13
	8.5 Influence of Parameter Variations on Optimum Electrical Capacity	8-18
9.	Conclusions and Recommendations	9-1
	9.1 Optimization Incorporating Project Economics	9-1
	9.2 Optimization Incorporating Long-Term Effects on the Reservoir	9-3
	9.3 Proof-of-Concept Experiments	9-3
10.	References	10-1

Executive Summary

Theoretical calculations are presented to estimate the electrical generating capacity of the hot fluids discharged from individual geothermal wells using small wellhead generating equipment over a wide range of reservoir and operating conditions. The purpose is to appraise the possibility of employing slim holes (instead of conventional production-size wells) to power such generators for remote off-grid applications such as rural electrification in developing countries. Frequently, the generating capacity desired is less than one megawatt, and can be as low as 100 kilowatts; if slim holes can be usefully employed, overall project costs will be significantly reduced.

This report presents the final results of the study. Both self-discharging wells and wells equipped with downhole pumps (either of the "lineshaft" or the "submersible" type) are examined. Several power plant designs are considered, including conventional single-flash backpressure and condensing steam turbines, binary plants, double-flash steam plants, and steam turbine / binary hybrid designs. Well inside diameters from 75 mm to 300 mm are considered; well depths vary from 300 to 1200 meters. Reservoir temperatures from 100°C to 240°C are examined, as are a variety of reservoir pressures and CO₂ contents and well productivity index values. Detailed results are presented graphically in Appendices A-J (Vols. 2-4), which constitute the bulk of the report.

Among the conclusions of the study is the observation that, if a downhole pump is needed to cause the well to discharge, "lineshaft" pumps provide superior performance for large-diameter (*i.e.* 300 mm) geothermal wells whereas for smaller wells the newer "submersible" pumps deliver more fluid to the wellhead and have a broader range of permissible operating conditions (at the present-day state of the art). Also, as well diameter decreases, the range of conditions narrows under which downhole pumps are likely to provide advantages over self-discharge. Downhole pumps of the "lineshaft" design cannot be practically applied to wells less than 150 mm inside diameter. Even "submersible" pumps are only marginally useful below this diameter.

The major conclusion is that slim holes can in fact provide useful amounts of electrical power (> 100 kWe) using self-discharging wells as small as 100 mm inside diameter from geothermal reservoirs at temperatures as low as 150°C. If the reservoir temperature is higher, electrical output exceeding 1000 kWe is possible. Even 75 mm ID wells can provide more than 100 kWe if the reservoir temperature exceeds 170°C. A self-discharging 150 mm slim hole can provide 1000 kWe at a reservoir temperature of 180°C and 3000 kWe at 240°C. If the reservoir temperature is lower, however, it will be necessary to resort to downhole pumps, due mainly to inadequate gas-lift to maintain the flow up the well. This means, in turn, that wells of at least 150 mm ID will be needed to accommodate the pumps. A 150 mm ID slim hole equipped with a downhole pump can produce 100 kWe if the reservoir temperature is 105°C, 430 kWe at 150°C reservoir temperature, and 1100 kWe at 200°C.

Although a broad spectrum of power plant designs was considered in the study, a few relatively simple plant/well/pump combinations appear to be most practical. For high-temperature reservoirs, a conventional single-flash condensing steam turbine powered by a self-discharging well seems to be the most promising candidate. Single-flash backpressure steam turbines can also be employed if the reservoir temperature is high enough - backpressure turbines offer mechanical simplicity, small size and low capital cost, but suffer from low overall efficiency and air-pollution problems. Although more elaborate plant designs (double-flash plants or flash/binary hybrids) can deliver more electricity from the same self-discharging well than a single-flash condensing plant, the additional mechanical complexity, cost, and maintenance problems probably will not justify the relatively marginal performance advantages for most off-grid applications.

For lower temperature resources, it will probably be necessary to apply a downhole pump. It has been traditional practice in the geothermal industry to choose a binary plant under these circumstances, and small-scale binary plants for projects of this type appear to be feasible for off-grid applications. As an alternative, single-flash condensing steam turbines can be used which obtain steam by expanding the compressed liquid delivered to the wellhead by the downhole pump in a sub-atmospheric separator. Under some conditions, such a sub-atmospheric flash plant may actually outperform a binary plant.

1

Introduction

1.1 Background

A growing market exists for off-grid electrification projects in the developing countries of the world. Small geothermal generating plants (from 100 to 1000 kilowatts capacity) offer much promise in various areas around the Pacific Rim, particularly Latin America, Indonesia, the Philippines, and isolated islands. Typically, these remote-area installations would consist of single production wells driving generating equipment located immediately adjacent to the wellhead. Entingh, *et al.*, (1994) have examined the economic feasibility of such projects and have concluded that small plants of this type can compete favorably with existing off-grid electrification techniques in this capacity range (mainly diesel generators), and also with other proposed renewable energy sources (photovoltaic solar, wind, hybrids).

It is already reasonably clear that small geothermal power plants in the 5 to 10 megawatt range are feasible, using one or two standard size production wells connected to a small wellhead generator. Conventional practice for projects of this type involves using steam turbines driving electrical generators for relatively high-temperature systems (reservoir temperature over 200°C or so). For lower-temperature systems, binary plants are usually recommended, and a downhole pump will be required in the production well(s) to maintain discharge and sufficient pressure to assure single-phase flow in the heat exchanger.

For geothermal projects in the 100–1000 kWe range, the costs of drilling and completing the production well dominate the economics. Entingh *et al.* assumed that the well would be drilled using conventional techniques (13-3/8 inch casing to 750 feet, 9-5/8 inch casing below that depth). If a slim hole could be used instead, considerable savings in drilling costs and final electricity price would be realized. Slim-hole drilling costs are typically only about one-third (per foot) of those of conventional drilling practice in the geothermal industry (Combs and Dunn, 1992). For remote applications, additional advantage would probably be realized by the relative portability of the small rigs required (it is even practical to transport slimhole rigs by freight helicopter, if needed). Furthermore, the environmental impact and land-use requirements of slim-hole drilling operations are generally much more acceptable than conventional drilling.

The principal shortcomings of slim holes for electrical power production are (1) the capacity of a slim hole to produce fluid is substantially less than that of a conventional production well (Pritchett, 1993), and (2) if a downhole pump is required, small-diameter downhole pumps suitable for high-temperature operation are not yet available on a commercial basis. The absence of suitable downhole pumps mostly reflects lack of demand; existing components are available to construct both “lineshaft” and “submersible” downhole pumps suitable for geothermal service in wells as small as six inches inside diameter, and perhaps smaller (Pritchett, 1997).

The remaining problem is to establish whether or not slim holes can deliver enough hot geothermal fluid to the wellhead to be of practical interest for off-grid electrification projects. Several years ago, the author undertook a preliminary study of this question (Pritchett, 1995a), using theoretical techniques to mathematically model the

fluid flow within geothermal wells of various diameters and to forecast their probable electrical generating capacity. This study was far from exhaustive, and employed several simplifying assumptions to render the problem tractable with minimum effort. Two generating schemes were considered: binary-type generating equipment supplied by wells containing downhole lineshaft pumps, and single-flash steam turbogenerators (both condensing and backpressure types) fueled by self-discharging wells.

The preliminary study indicated that, under reasonable conditions, slim holes can deliver sufficient fluid to be of considerable practical utility using either technique. Electrical generating capacities over 100 kWe are possible with binary generators using wells as small as four inches inside diameter so long as the reservoir temperature exceeds 130°C or so. At 200°C reservoir temperature, a 6" I.D. well could supply a one-megawatt binary generator. For the steam turbogenerator cases, a fixed (and non-optimum) steam turbine inlet pressure of 4.46 bars (50 psig) was selected. Using self-discharging slim holes, a 4" I.D. well can produce as much as 800 kWe from a reservoir at 240°C, and a 6" I.D. hole can provide enough steam to generate two megawatts of electricity. Owing mainly to the fixed turbine inlet pressure considered, however, the study indicated a minimum practical reservoir temperature around 180°C for flash-steam generators.

These results encouraged the U. S. Department of Energy (DOE) to provide subsequent support for elaboration and extension of the feasibility study, relaxing a number of the more restrictive assumptions and approximations. The purpose is to provide more quantitative results concerning the probable generating capacities of individual geothermal wells, and how these capacities depend upon various pertinent parameters including well diameter, feedpoint depth, productivity index, reservoir pressure, reservoir temperature, and reservoir fluid composition. A preliminary report concerning results for self-discharging wells driving single-flash backpressure and condensing wellhead steam turbines was prepared three years ago (Pritchett, 1995b), which largely confirmed the results of the original preliminary study and indicated that, if anything, the original study had probably underestimated the potential of slim-hole electricity generation.

Next, an examination was made of the current capabilities of downhole geothermal pumps and their potential applicability to slim holes (Pritchett, 1997). Two different types of downhole pumps are available—"lineshaft" pumps in which a wellhead motor drives the downhole pump body using a long concentric rotating driveshaft, and "submersible" pumps in which the electric motor is also located downhole (see Figure 1.1). If the well is self-discharging, the fluid which emerges from the wellhead is a mixture of water and steam; the evolution and expansion of the steam provides the lift to raise the boiling mixture to the surface. If a downhole pump is installed (either "lineshaft" or "submersible"), compressed liquid water is driven upward, through a sealed pipe (the "column") which fits within the well casing, under sufficient pressure that the fluid emerging from the wellhead remains in a single-phase-liquid condition. This is necessary to avoid scaling within the column. The annular region between the column pipe and the well casing is filled with stationary saturated steam under pressure, and sealed at the wellhead. For large-diameter wells (300 mm; 12-inch), "lineshaft" pumps provide better performance than "submersible" pumps, due mainly to limitations on submersible motor power. For smaller wells, however, "submersible" pumps can deliver more fluid to the surface under a wider range of conditions.

Figure 1.1. Self-discharging wells compared to wells with downhole pumps.

In the present report, calculations are reported of the electrical generating capacities of wells of various diameters ranging between 75 mm (~3") and 300 mm (~12"). All wells treated are considered to be drilled vertically, with uniform inside diameter from wellhead to feedpoint. The calculations represent stable discharge conditions. Both self-discharging wells (using the same techniques described by Pritchett, 1995b) and wells with lineshaft and submersible downhole pumps (Pritchett, 1997) are considered. The fluids from these wells are then employed to generate electricity using various techniques. These include single-flash backpressure and condensing steam turbines (using a slightly improved power plant model similar to that of Pritchett, 1995b), double-flash plants with both backpressure and condensing high-pressure stages, binary plants, and both backpressure and condensing steam/binary hybrid plants (with the steam from the separator driving a steam turbine and the separated liquid driving a binary-cycle plant).

1.2 Free Parameters

Combining the three well production modes (self-discharging, lineshaft pump or submersible pump) with the seven different power plant designs considered (single-flash backpressure turbine, single-flash condensing turbine, double-flash backpressure/condensing turbine, double-flash condensing/condensing turbine, binary plant, backpressure/binary hybrid or condensing/binary hybrid) yields a total of twenty-one combinations (not all of which are practical). For each of the twenty-one combinations, the optimum performance (defined as maximum net electrical generation capacity) which may be obtained by optimizing the well/plant design depends on several free parameters, including:

- (1) Well diameter (d)—the (uniform) inside diameter of the well. For self-discharging wells, the values selected are 75 mm, 100 mm, 125 mm, 150 mm, 200 mm and 300 mm. For wells with “lineshaft” downhole pumps, diameters of 150 mm, 200 mm, 250 mm and 300 mm are examined (smaller diameter lineshaft pumps are impractical - see Pritchett, 1997). Diameters of 100 mm, 125 mm, 150 mm, 200 mm, 250 mm and 300 mm are considered for wells with “submersible” pumps.
- (2) Feedpoint depth (D_p)—the depth below the wellhead at which reservoir fluid enters the well. It is assumed that shut-in stable reservoir fluid conditions at the feedpoint depth are single-phase liquid. Feedpoint depths of 300 meters, 600 m, 900 m and 1200 m are considered.
- (3) Reservoir permeability, or well “productivity index” (I). Three different values are considered: 2 kilograms per second per bar (representative of “low permeability” resources), 4 kg/s/bar (moderate permeability) and infinity (good permeability).
- (4) Reservoir gas content (C_p)—the mass fraction of noncondensable gases dissolved in the reservoir liquid. It is assumed that the noncondensable gas may be regarded as consisting exclusively of carbon dioxide. Liquid-phase CO_2 mass fractions of zero, 0.25, 0.5, 0.75 and 1 percent are considered.
- (5) Reservoir temperature (T_p) under shut-in conditions at the feedpoint depth. Numerous values are considered, ranging from 100°C to 240°C, subject to the restriction that the temperature must lie below the boiling point for the particular reservoir pressure and CO_2 content involved. For cases involving self-discharging wells, the temperature interval considered is 5°C (100°C, 105°C, 110°C, ..., 240°C). For wells with downhole pumps, the interval is 1°C (100°C, 101°C, 102°C, ..., 240°C).
- (6) Reservoir pressure or “piezometric surface depth” (Z). The piezometric surface depth, together with the reservoir temperature and the feedpoint depth, establishes the stable shut-in reservoir pressure at the feedpoint depth, as discussed below. Values for Z ranging from zero to 250 meters are considered. For self-discharging wells, at 50-meter sampling interval is used ($Z = 0, 50 \text{ m}, 100 \text{ m}, 150 \text{ m}, 200 \text{ m}$ or 250 m), whereas for pumped wells a 10-meter interval is employed ($Z = 0, 10 \text{ m}, 20 \text{ m}, \dots, 240 \text{ m}, 250 \text{ m}$).

As can easily be appreciated, the total number of cases to be considered is potentially astronomical. To reduce the scope of the problem, as in previously-reported results (Pritchett, 1995b) the calculations were subdivided into two "series":

Series 1: CO₂ content = 0; all other parameters vary over entire range outlined above.

Series 2: CO₂ content = 0.25, 0.50, 0.75 or 1.00 percent; feedpoint depth (D_f) and productivity index (I) restricted to single values (600 m and 4 kg/s/bar respectively); remaining parameters vary over entire range.

In other words, "Series 1" considers the entire range of parameter variation for reservoirs with little or no noncondensibles—"Series 2" considers the effects of CO₂ content upon results for a somewhat more restricted parameter range. The number of combinations considered is still extremely large, however; for Series 1, the total number of combinations of plant types, well types (self-discharging, lineshaft pump or submersible pump), feedpoint depths, productivity index values, CO₂ content values, reservoir temperature values and piezometric surface depth values adds up to 3,167,136. Series 2 is somewhat smaller (1,055,712); the total number of cases considered thus totals 4,222,848. Not all of these combinations are practical, of course—some yield reservoirs which are above the boiling point or boiling conditions at the maximum possible pump depth, for example—but a major fraction of these combinations were actually calculated.

1.3 Depth of the Piezometric Surface

The "piezometric surface depth" is herein defined as follows:

$$Z = D_f - \frac{(P_r - P_a)}{\nabla P(T_r)} \quad (1.1)$$

where

- Z = piezometric surface depth
- D_f = well feedpoint depth
- T_r = reservoir temperature at feedpoint depth
- P_r = stable reservoir pressure at feedpoint depth
- P_a = one standard atmosphere = 1.01325 bars
- $\nabla P(T)$ = hydrostatic gradient for saturated liquid H₂O at temperature T.

In other words, the piezometric surface lies at the depth at which a reservoir-temperature hydrostatic pressure gradient, projected upward from the stable shut-in feedpoint pressure, would reach one atmosphere. The piezometric surface is usually slightly shallower than the stable standing water level in the well after a long period of shut-in. Table 1.1 lists values of ΔP for various temperatures. Figure 1.2 shows the relationships among piezometric surface depth (Z), reservoir temperature (T_r) and reservoir pressure (P_r) for well feedpoint depth (D_f) = 300 meters. Similarly, Figures 1.3, 1.4 and 1.5 show the same relationships for feedpoint depths (D_f) of 600, 900 and 1200 meters respectively.

Table 1.1. Values of the Hydrostatic Gradient ∇P for Various Temperatures .

Temperature (°C)	∇P (bars/meter)
90	0.0946
100	0.0939
110	0.0932
120	0.0924
130	0.0916
140	0.0908
150	0.0898
160	0.0889
170	0.0879
180	0.0869
190	0.0858
200	0.0847
210	0.0836
220	0.0824
230	0.0811
240	0.0797
250	0.0783

Both the original feasibility study (Pritchett, 1995a) and the present work consider only single-phase-liquid reservoirs. Certain of the above combinations of (feedpoint depth / reservoir temperature / reservoir CO_2 content / piezometric surface depth) result in reservoir pressures which are below the bubble-point pressure for the local conditions. For the CO_2 -free Series 1 calculations, these combinations fall into the shaded area ("CO₂ mass fraction = zero") in Figures 1.2–1.5. For Series 2, they fall to the right of the bubble-point pressure curve for the pertinent reservoir CO_2 concentration in Figure 1.3. These combinations were eliminated from the list of cases to be examined.

Figure 1.2. Relationships among reservoir feedpoint temperature, piezometric surface depth, and reservoir stable feedpoint pressure for well feedpoint depth = 300 meters.

Figure 1.3. Relationships among reservoir feedpoint temperature, piezometric surface depth, and reservoir stable feedpoint pressure for well feedpoint depth = 600 meters.

Figure 1.4. Relationships among reservoir feedpoint temperature, piezometric surface depth, and reservoir stable feedpoint pressure for well feedpoint depth = 900 meters.

Figure 1.5. Relationships among reservoir feedpoint temperature, piezometric surface depth, and reservoir stable feedpoint pressure for well feedpoint depth = 1200 meters.

1.4 Practical Considerations Regarding Well Diameter

For self-discharging wells, the performance of a particular production well may be described by the “wellhead characteristics”; the relationships among flowing wellhead pressure and water and steam discharge rates under stable flowing conditions. Figure 1.6 shows a typical “characteristics curve” for a particular geothermal well. The total discharge rate and the flowing wellhead pressure are related by this curve. The maximum flow rate obtainable from the well $\dot{M}_{\max}$ and the maximum attainable wellhead pressure $P_{\max}$ (usually realized at a flow rate substantially smaller than $\dot{M}_{\max}$) are functions of the properties of both the well and the reservoir. Usually, both $P_{\max}$ and $\dot{M}_{\max}$ increase with (1) increasing reservoir pressure, (2) increasing reservoir temperature, (3) increasing reservoir CO_2 content, (4) increasing well diameter, (5) increasing reservoir permeability (or well productivity index), and (6) decreasing feedpoint depth.

Figure 1.6. Typical performance characteristics relationship for a self-discharging geothermal well.

Figure 1.7 shows how the maximum attainable flow rate $\dot{M}_{\max}$ varies with well inside diameter according to a series of numerical calculations made using the techniques discussed in Section 2 of this report. In these calculations, the well feedpoint depth is 600 meters, the reservoir temperature is 200°C, the piezometric surface depth is 100 meters (which yields a stable shut-in reservoir pressure of 43.4 bars at 600 meters depth) and no CO_2 is present. For sufficiently high reservoir permeability (infinite productivity index), the attainable discharge rate increases approximately with diameter (d) to the power 2.5, as indicated (see, for example, Pritchett, 1993). For lower productivity indices, flow rates are reduced (particularly for large-diameter wells).

Figure 1.7. Influences of well diameter and well productivity index upon maximum attainable discharge rate for self-discharging geothermal wells.

Since, all else being equal, the cost of drilling a geothermal well in a particular location to a particular depth increases approximately in proportion to the well diameter, it follows that slim holes will not be cost-effective production wells for large geothermal development projects in high-permeability geothermal fields. The economic value of a geothermal well is in direct proportion to the rate at which it can deliver hot fluid to the wellhead. Since, for high-permeability systems, this rate increases approximately as diameter^{2.5}, it follows that the drilling cost per kilowatt of electrical capacity will vary roughly as diameter^{-1.5}; using 300 mm ID production wells will therefore be approximately three times more cost-effective than using 150 mm ID slim holes. Under these circumstances (high reservoir permeability and large geothermal projects with power demand in excess of the capacity of a single well), it seems reasonably clear that the best strategy is to drill the largest diameter production wells feasible.

Even for large projects, however, if the reservoir permeability is limited, smaller wells will sometimes be advantageous. In the example shown in Figure 1.7, for a productivity index of 4 kg/s/bar the discharge rate is still increasing in proportion to diameter^{1.6} at a diameter of 300 mm, indicating that optimum performance will be

found at even higher diameters. For productivity index = 2 kg/s/bar, however, the slope of the curve appears to pass through diameter^{1.0} at a well diameter around 250 mm, indicating that 250 mm ID is about the optimum well diameter under these conditions.

For small off-grid projects, on the other hand, the total electricity demand is limited. For purposes of this study, it was assumed that the market was typically for between 100 and 1000 kWe electrical capacity. The fluid discharge rates shown in Figure 1.7, if supplied to a conventional single-flash condensing steam turbine, are calculated (using the methods described later in this report) to be capable of electrical capacities as indicated in Table 1.2. These results clearly show, for this case at least, that wells over 150 mm inside diameter discharge far too much fluid (and cost far too much to drill) for applications in this range.

Table 1.2. Calculated Electrical Capacities of Individual Self-discharging Geothermal Wells Using Wellhead Single-Flash Condensing Steam Turbines.

Well Inside Diameter (mm)	2 kg/s/bar Productivity Index (kWe)	4 kg/s/bar Productivity Index (kWe)	Infinite Productivity Index (kWe)
75	229	241	256
100	461	504	560
125	748	848	1005
150	1072	1261	1597
200	1772	2217	3267
300	2595	4344	8735

Feedpoint depth = 600 meters
 Piezometric surface depth = 100 meters
 Reservoir temperature = 200°C
 No reservoir CO₂

Another advantage of slim holes (both for small-scale off-grid electrical generation projects and for geothermal exploration generally) pertains to the risk of "dry holes". Essentially, a "dry hole" is a well which cannot be made to spontaneously self-discharge. In other words, the performance characteristics of the well (Figure 1.6) involve a "maximum attainable flowing wellhead pressure" which is around one atmosphere or less. This can happen due to a variety of causes (reservoir pressure or temperature too low, for example), but in a well-established geothermal area where other wells have demonstrated production capacity, the cause is usually a low productivity index (resulting from low local permeability near the well feedpoint). Figure 1.8 shows the results of a series of calculations (again using the methods outlined in Section 2) of the maximum attainable wellhead pressure as a function of productivity index for wells of 75 mm, 150 mm and 300 mm inside diameter. Other parameters are the same as in Figure 1.7.

Figure 1.8. Influences of well diameter and well productivity index upon maximum attainable flowing wellhead pressure for self-discharging geothermal wells.

For high values of the productivity index, larger wells tend to be characterized by higher wellhead pressures. As productivity index (reservoir permeability) decreases, however, the larger diameter wells suffer more in performance from increased reservoir flow resistance than the slim holes. In this particular case, it is unlikely that the 300 mm ID well could be induced to discharge if its productivity index is less than 0.15 kg/s/bar or so - for this level of reservoir permeability, a 300 mm ID well would be a "dry hole". For the same level of reservoir permeability, on the other hand, the slim holes should still be capable of sustained discharge. Thus, using slim holes for reservoir exploration both (1) reduces the costs of exploration drilling and (2) reduces the "dry hole" risk. In typical off-grid rural electrification projects, extensive exploration programs will usually be too expensive because of the limited revenue stream to be obtained from the project; considerable "wildcat" drilling is likely to be involved. Slim holes, in addition to being cheaper to drill and more in line with the electrical power demands of the project, offer reduced risk of "dry hole" failures.

2

Characteristics of Self-Discharging Geothermal Wells

The mathematical representation used to obtain the results for geothermal well electrical capacity consists of three essential elements; the description of events taking place in the reservoir as fluid flows toward the well, the flow of fluid mass and heat up the well from the feedpoint to the wellhead, and the generation of electricity from the fluids discharged at the wellhead. The bulk of the computational effort involved the characterization of the flow up self-discharging wells. To efficiently accomplish the enormous number of wellbore simulations required, the essential computational modules of the WELBOR borehole-flow simulator (Pritchett, 1985) were incorporated within a master software package which operates in "batch" mode.

2.1 Modeling Flow in the Reservoir

Earlier preliminary studies (Pritchett, 1995a) clearly suggest (1) that the relative effect of reservoir flow resistance increases with increasing well diameter, and (2) that for slim holes, changing the well productivity index assumed over a very wide range (from a relatively low value - 2 kg/s/bar - to infinity) has a surprisingly small effect upon the well's electrical capacity. For six-inch holes, varying the productivity index over this range changes the electrical output by only about 25 percent, and for four-inch holes the electrical output variability is less than 15 percent.

It is important to recognize that the values of the productivity index for different individual wells in a particular geothermal field are likely to exhibit a considerable amount of variability. The productivity index depends on both the permeability of the reservoir itself (which usually varies significantly from point to point), and also upon the details of how the well happens to become connected to the reservoir's fracture network. Accordingly, a significant component of random chance is present; adjacent wells will often have important differences in productivity index. Garg and Combs (1997), however, have examined a substantial body of field data from several Japanese geothermal fields together with some U.S. data concerning the performance of wells of various diameters drilled into the same geothermal area. Most of these data sets exhibit little or no correlation between productivity index and well diameter. Therefore, it seems most reasonable to regard the productivity index as a "reservoir property" similar to permeability, rather than a "well property".

For the present study, the "productivity index" approach was therefore retained as a reasonable way to represent reservoir flow resistance, and in fact the same three values as used in the feasibility study were used: 2 kg/s/bar, 4 kg/s/bar, and infinity. These represent a reasonable range of values for typical geothermal fields. For present purposes, the "productivity index" is formally defined as follows:

$$I = \lim_{P_r \rightarrow P_f} \left(\frac{\dot{M}}{P_r - P_f} \right) \quad (2.1)$$

where "I" is the productivity index, " $\dot{M}$ " is the total fluid flow rate from the formation into the well at the feedpoint, " P_r " is the stable reservoir pressure at the feedpoint depth, and " P_f " is the flowing pressure within the well at the feedpoint (with $P_f < P_r$). Under most circumstances, the relationship between the stable pressure within the flowing well at the feedpoint depth and the total fluid discharge rate (water plus steam) at the wellhead is therefore given simply by:

$$P_f = P_r - (\dot{M} / I) \quad (2.2)$$

To compute conditions within the well at the feedpoint depth, the procedure is as follows. Starting with the reservoir pressure P_r , reservoir temperature T_r and the reservoir CO_2 mass fraction C_r , the corresponding fluid enthalpy (H_r) is determined. Then, the condition of the fluid within the well at the feedpoint is determined by imposing the same fluid enthalpy and CO_2 mass fraction (C_r), but reducing the pressure from the reservoir value to the feedpoint value P_f .

In most cases, the resulting state of the fluid at the feedpoint will be single-phase liquid, at pressure P_f and a slightly elevated temperature (about 0.01 degrees Celsius per bar of pressure decline from reservoir conditions; essentially negligible). So long as the feedpoint state is single-phase liquid, no further attention need be paid to the flow in the reservoir, and it is permissible to proceed directly to calculation of the flow up the well towards the wellhead.

Under certain circumstances, however, (combinations of low productivity index, high reservoir temperature, low reservoir pressure, shallow feedpoint depth, large well diameter, and/or high reservoir CO_2 content), the resulting predicted state at the feedpoint will not be single-phase liquid, but will also contain a gaseous component. This means that the water is actually boiling within the formation, in an annular region surrounding the well. Since the resistance of the reservoir rock to the flow of a two-phase mixture will generally be substantially greater than to pure liquid, the use of Eq. 2.2 above to represent the flow resistance is too optimistic in these cases.

The degree to which the evolution of steam within the formation around the well is likely to impede the fluid flow towards the borehole is extremely site-specific, and no general treatment of this question is available. The approach taken in the present study is as follows. Let P_b represent the "bubble-point pressure" associated with fluid enthalpy H_r and CO_2 content C_r , and define a "critical flow rate" $\dot{M}_c$ by:

$$\dot{M}_c = I(P_r - P_b) \quad (2.3)$$

So long as the well discharge rate $\dot{M}$ is less than the critical rate $\dot{M}_c$, the reservoir will remain single-phase liquid and Eq. 2.2 may be used to obtain the feedpoint pressure. If, however, $\dot{M} > \dot{M}_c$, the following alternate form is adopted:

$$P_f = P_b + F[(P_r - P_b) - (\dot{M} / I)] \quad (2.4)$$

Here, "F" represents the factor by which flow resistance in the two-phase region exceeds that of the corresponding single-phase liquid situation. Note that if "F" is set to unity, Eq. 2.2 is recovered. The proper value for "F" is likely to be site-specific, and to depend upon the relative permeability functions appropriate for the reservoir and the detailed geometry of the fracture network supplying the well. Theoretical considerations and field experience (see, for example, Garg and Combs, 1997; Garg and Combs, 1998) suggest that a suitable "typical" value for F probably lies somewhere between 5 and 10. Somewhat arbitrarily, therefore, the value:

$$F = 8 \quad (2.5)$$

was selected for the present study. This value is probably conservative.

Figure 2.1 illustrates an extreme case of the results of Eqs. 2.4 and 2.5. Here, the reservoir pressure is 35 bars and the feedpoint pressure is fixed at 20 bars; the productivity index is 4 kg/second/bar, so that under single-phase conditions the fluid supply rate from the formation to the well at the feedpoint is 60 kilograms per second (all liquid). The boiling point for water at 20 bars absolute pressure is 212.3°C, so that if the reservoir temperature exceeds this value, two-phase flow will occur within the reservoir in the vicinity of the well feedpoint. Therefore, as Figure 3.1 shows, for higher reservoir temperatures steam enters the well at the feedpoint and the total flow rate is less than 60 kg/s.

Figure 2.1. Effects of in-formation boiling upon downhole productivity.

It should also be noted that the present model, being steady, does not take into account the so-called "excess enthalpy" which can occur in a transient fashion during the time while the relatively impermeable country rock between the fractures cools down conductively to a state of temperature equilibrium with the two-phase fluid flowing in the fractures around the well. This transient period (which can last weeks, months, and in some cases years) is characterized by augmented steam supply to the well and, therefore, higher generating capacity than the present steady calculations would indicate. Thus, predicted results which involve two-phase flow in the reservoir should be regarded as less reliable than those for which in-situ boiling does not occur. Furthermore, results under these conditions are most likely to err on the conservative side.

Fortunately, of the numerous combinations of conditions (reservoir pressure, reservoir temperature, reservoir CO_2 content, productivity index, feedpoint depth and borehole diameter) employed for calculations of the behavior of self-discharging wells for this study, only 7.2 percent result in predictions of two-phase flow in the reservoir, even at the highest discharge rate considered. Of these, more than half are members of "Series 2" (with substantial reservoir CO_2); 16 percent of the high- CO_2 "Series 2" cases result in two-phase flow, whereas only 4.4 percent of the zero- CO_2 "Series 1" cases were affected. Furthermore, in Series 1, more than half of the two-phase cases were for the shallowest feedpoint depth considered (300 m). Only 7 percent of these two-phase cases involve feedpoint depths exceeding 600 meters. More than half of the two-phase cases involved reservoir temperatures of 210°C or higher. Finally, most of the two-phase cases involved the largest-diameter wells (with their higher fluid-carrying capacity); the slim holes of primary interest in this study were only rarely involved. Wells of 300 mm diameter account for 42 percent of the cases and those of 200 mm diameter represent another 24 percent; by contrast the slim holes of 150, 125, 100 and 75 mm diameter account for only 15, 10, 6 and 3 percent, respectively, of these cases.

2.2 Modeling Fluid and Heat Flow Up the Well

Figure 2.2 illustrates one of the computed profiles of conditions within discharging wells which were generated in the course of this project using the computational modules of the WELBOR simulator (Pritchett, 1985). In this particular case, the inside diameter of the well is 150 mm and the feedpoint depth is 600 meters. No CO_2 is present and, at 600 meters depth, the reservoir temperature is 200°C and the stable shut-in pressure is 43.39 bars (corresponding to a piezometric surface depth of 100 meters). This particular profile corresponds to a flowing wellhead pressure of 3.00 bars, which results in wellhead discharges of 23.72 kilograms per second of liquid and 3.50 kg/s of steam (27.22 kg/s total wellhead discharge).

With a total discharge rate $\dot{M}$ of 27.22 kg/s and a productivity index (I) of 4 kg/s/bar, Eq. 2.2 yields a value of 36.58 bars pressure within the well at the 600-meter feedpoint depth (P_f). As Figure 2.2A indicates, the piezometric gradient is about 0.085 bars/meter; the average flowing pressure gradient within the well below 353 meters depth is only slightly greater (about 0.086 bars/meter). The pressure gradient within the well consists of the sum of three terms (Pritchett, 1985); hydrostatic pressure, pipe friction, and a "fluid acceleration" term. The WELBOR code represents pipe friction (for both single-phase and two-phase flow) using the method of Dukler, Wicks and Cleveland (1964). In the single-phase region below 353 meters depth in the well, hydrostatic effects dominate, frictional effects play a very minor role, and the acceleration term is negligible.

Figure 2.2. Computed self-discharging mass and heat flow in the well for a typical case.

At 353 meters depth, the rising fluid reaches its bubble-point pressure and begins to boil. Within the two-phase region, all three terms in the momentum equation (hydrostatic pressure, pipe friction, and fluid acceleration) play significant roles. On the average, the pressure gradient within the upper part of the flowing well is significantly less than in the deeper single-phase zone (0.035 bars/m compared to 0.086), but also varies significantly with depth. In the upper part of the well, most of the wellbore volume is filled with steam (as contrasted to liquid water); at the wellhead, the liquid phase only occupies 7.44 percent of the well's cross-section area. As Figure 2.2B shows, the upflow rate of the liquid water below 353 meters depth is essentially constant and modest (1.78 meters per second), but once boiling begins both phases accelerate, with the upward velocity of the vapor phase exceeding that of the liquid phase due to buoyancy. At the wellhead, the liquid phase is rising at 19.4 meters per second, whereas the steam phase velocity is 129 m/s. In the WELBOR code, interphase "slip" (the velocity difference between the phases) is treated using the empirical correlation of Hughmark (1962), which has been successfully matched to measured downhole flowing data in numerous geothermal wells.

In general, the specific enthalpy of an element of fluid entering the well at the feedpoint and flowing upward within the hole will decrease due to three causes: (1) increase in the fluid kinetic energy (Figure 2.2B), which takes place at the expense of the fluid enthalpy, (2) work done against the force of gravity, and (3) lateral heat losses through the casing to the surrounding reservoir. Lateral heat losses from the well to the surrounding formation through the casing were taken into account in these calculations by (1) assuming a particular distribution of temperature in the rock formation adjacent to, but a few meters away from, the well, and (2) employing a heat transfer coefficient derivable from the thermal boundary layer analysis presented by Minkowycz and Cheng (1976). As shown by Pritchett (1981; 1993), the effective heat transfer coefficient, based on the Minkowycz-Cheng results, may be expressed by:

$$U = 0.6 K / d \quad (2.6)$$

where "d" is the borehole diameter and "K" is the "effective formation thermal conductivity". The outward-directed heat flux from the fluid in the well to the formation outside is then:

$$\text{Heat flux} = U (T - T_o) \quad (2.7)$$

where T is the temperature within the well at a particular depth and T_o is the "formation temperature" at the same depth. Thermal conductivities of geothermal reservoir rocks typically vary between about 2 watts/meter/°C and 4 watts/meter/°C. For the present study, an "average" value was assumed for all calculations:

$$K = 3 \text{ watts/meter/}^\circ\text{C} \quad (2.8)$$

The distribution of formation temperature between the surface and the feedpoint depth will obviously vary from site to site. For the present calculations, it was assumed that the formation temperature distribution $T_o(D)$ is given by the following quadratic fit:

$$T_o = T_a + (T_r - T_a) \frac{D}{2D_f} \left(3 - \frac{D}{D_f} \right) \quad (2.9)$$

where D is depth (which varies from zero to D_f , the feedpoint depth), T_a is air temperature (taken as 10°C), and T_f is the reservoir temperature at the feedpoint depth (between 100°C and 240°C). The various formation temperature distributions (for the reservoir temperatures and feedpoint depths considered in this study) are illustrated in Figure 2.3.

For the particular case depicted in Figure 2.2, Figure 2.2C shows that (as is usually the case), the formation temperature is significantly less than the temperature of the upflowing fluid within the well; thus, the net lateral heat flow through the casing will be outward. Within the single-phase flow region below 353 meters depth, the fluid is almost isothermal (the temperature actually decreases slightly, from 200.0°C to 199.6°C , between the feedpoint and the boiling interface). Above the boiling level, the temperature declines with increasing elevation, following the saturation pressure/temperature relation for two-phase water/steam mixtures.

As discussed above, the flowing enthalpy of the fluid within the well changes with elevation because of three effects: change in fluid kinetic energy, work done against gravity, and lateral heat losses. Figure 2.2D shows all three effects for this particular case. The flowing enthalpy at the feedpoint is 853.3 Joules per gram; at the wellhead, the flowing enthalpy has declined by 14.1 J/g to 839.2 J/g. The work done against gravity is given simply by:

$$dH_g = g dz \quad (2.10)$$

where “ g ” is the acceleration of gravity and “ dz ” is the change in elevation. Thus, the effect upon the wellhead enthalpy is just:

$$\Delta H_g = g D_f \quad (2.11)$$

which depends only on feedpoint depth. For this particular case ($D_f = 600$ meters), this amounts to 5.9 J/g; for the various feedpoint depths considered in this study, the gravity-work enthalpy effect is:

D_f (m)	ΔH_g (J/g)
300	2.94
600	5.88
900	8.82
1200	11.76

The effect of the change in kinetic energy is very small; for the case illustrated in Figure 2.2D, the contribution is negligible for most of the well’s length and only becomes noticeable at shallow depths, reaching a maximum value of $\Delta H_{ke} = 1.2$ Joules/gram at the wellhead.

Figure 2.3. Profiles of formation temperature used in the study.

The largest contribution to the enthalpy decline (7.1 Joules/gram in this particular case) is the effect of lateral heat losses. Using Eqs. (2.6) and (2.7) above, it may easily be shown that the reduction in wellhead enthalpy due to this effect may be written:

$$\Delta H_1 = 0.6 \pi K(\bar{T} - \bar{T}_0) D_f / \dot{M} \quad (2.12)$$

where $\bar{T}$ is the average value of the temperature in the well (averaged over the length of the well) and $\bar{T}_0$ is the similarly-averaged formation temperature. Note that ΔH_1 increases with increasing feedpoint depth (D_f) and decreases with increasing flow rate $\dot{M}$. Note also that well diameter does not appear explicitly in Eq. 2.12. Since slim holes are restricted in their total carrying capacity compared to large-diameter wells, these lateral heat losses are more important for slim holes than for production-size wells.

2.3 Modeling Wellhead Discharge Characteristics

As discussed previously, the motivation for calculating the fluid flow from the reservoir into the well at the feedpoint and then up the borehole to the wellhead is to establish the flowing conditions at the wellhead. The relationships among wellhead pressure and discharge rates of water and steam are essential for determining the electrical generating capacity of the well. These "wellhead characteristics" may be expressed by how (1) wellhead pressure and (2) flowing wellhead enthalpy depend on total (liquid plus vapor) mass discharge rate (for cases with reservoir CO_2 , the mass fraction of CO_2 in the liquid and vapor phases is also needed). These parameters permit the determination of the separate flow rates of water and of steam at the wellhead using the steam tables.

To illustrate, consider the following typical set of parameters (which represent mainly near-median values in the study range):

Casing inside diameter:	150 mm
Feedpoint depth:	600 m
Reservoir CO_2 content:	none
Reservoir temperature:	200°C
Piezometric surface depth:	100 m
Productivity index:	4 kg/s/bar

The computed discharge characteristics for this case are shown in Figure 2.4. Calculation of these relationships involved numerous integrations up the borehole using the WELBOR code, each for a different value of the total discharge rate. In this case, the maximum attainable wellhead pressure is 6.9 bars (realized at a total discharge rate of 9.7 kg/s) and the maximum attainable discharge rate is 28 kg/s (for wellhead pressures less than 2.4 bars). As flow rate decreases, the wellhead discharge enthalpy also decreases, reflecting the growing relative influence of lateral heat losses to the cooler formation rocks surrounding the well. The heat losses are greater for lower flow rates due to the longer residence time of each parcel of fluid within the borehole.

Figure 2.4. Stable wellhead discharge characteristics for a typical case.

This increased heat loss is also responsible for the decrease in flowing wellhead pressure with decreasing flow rate below 9.7 kg/s. As the heat content of the flowing fluid declines with decreasing flow rate, the bubble-point pressure also declines, so that the transition from single-phase to two-phase flow occurs at a shallower depth in the well. The downhole pressure gradient in the single-phase region substantially exceeds that in the two-phase region (Figure 2.2A), which means that the average value (from feedpoint to wellhead) of the pressure gradient increases with decreasing flow rate. Since, at low flow rates, the flowing feedpoint pressure is nearly constant, this means that the wellhead pressure will decrease with decreasing flow rate for sufficiently low flow rate values.

Apart from well diameter itself, the temperature of the reservoir has the greatest effect upon wellhead discharge characteristics. Figure 2.5 illustrates the effects of perturbing the reservoir temperature to both higher and lower values as compared to the original "base" case discussed above. Increasing the reservoir temperature causes the maximum attainable wellhead pressure to increase, and also tends to increase the maximum total possible discharge. Note, however, that for the highest temperature considered (240°C), if the flowing wellhead pressure is less than about 12 bars, feedpoint pressures within the well will become so low that boiling will begin in the formation outside the well. The resulting increase in reservoir flow resistance (see Section 2.1) adversely influences the discharge characteristics in this range.

Next, we revert to a fixed reservoir temperature of 200°C but examine the effect of varying the stable shut-in reservoir pressure by considering piezometric surface depths other than the $Z = 100$ meter "base case" value. Figure 2.6 shows how changing the reservoir pressure influences the relationships among wellhead pressure, wellhead enthalpy, and total discharge rate. Increases in reservoir pressure (decreases in piezometric surface depth) have the effect of increasing both the maximum attainable reservoir pressure and the maximum possible discharge rate. The wellhead enthalpy, by contrast, is insensitive to changes in reservoir pressure.

Returning to $Z = 100$ meters, we next examine the effect of non-zero dissolved reservoir CO_2 . Figure 2.7 shows that, not surprisingly, the effect of raising the reservoir CO_2 content is to increase both the maximum attainable wellhead pressure and the maximum total discharge rate. The reason is simply that raising the reservoir CO_2 content increases the bubble-point pressure; the region occupied by vapor within the wellbore increases, the average flowing pressure gradient within the well decreases, and the wellhead pressure rises. Note, however, that the curve labeled "1 percent CO_2 " exhibits markedly degraded performance for flow rates above about 11 kg/second. The reason is that, for this CO_2 content, production at higher flow rates causes the bubble-point within the well to reach the feedpoint depth, and boiling begins within the formation. As a result, the reservoir flow resistance increases markedly (see Section 2.1). As Figure 2.7 also shows, the flowing wellhead enthalpy is only slightly influenced by CO_2 content within this range. The slightly lower enthalpies at high CO_2 mass fractions mainly reflect the lower heat capacity of CO_2 compared to water.

Well performance is obviously very sensitive to diameter. Figure 2.8 shows how the wellhead flowing characteristics change if the well diameter varies from the "base case" value of 150 mm. The maximum attainable wellhead pressures are all about the same (between 6 and 7 bars); the wellhead enthalpy as a function of discharge rate is virtually independent of borehole diameter, as suggested by Eq. 2.12. The attainable discharge rates, of course, are strongly dependent upon well diameter. As noted previously, Pritchett (1993) showed that the maximum attainable discharge rate will increase in proportion to (diameter^{2.5}) so long as the reservoir permeability is sufficient.

As already observed, variations in the productivity index (reservoir permeability) have much greater effects on large-diameter wells than they do on slim holes. Figure 2.9 illustrates the stable self-discharge wellhead characteristics for a very small-diameter well (75 mm I.D.) as a function of total discharge rate for productivity index (I) equal to 2 kg/s/bar, 4 kg/s/bar, and infinity. As before, the feedpoint depth is 600 meters, no CO₂ is present, the reservoir temperature is 200°C and the piezometric surface depth is 100 meters. Although performance improves slightly with increase in productivity index, the range of variation is quite small. Figures 2.10 and 2.11 are similar, but for 150 mm and 300 mm I.D. wells respectively. As well diameter increases, the frictional pressure drop in the formation comes to represent a larger and larger fraction of the total pressure decline, so that the effect of productivity index variation is very important for large-diameter production wells.

The final parameter to be considered is the depth of the feedpoint (in this study, 300, 600, 900 or 1200 meters; the base case is 600 m). As in the case of the productivity index, the relative importance of feedpoint depth variation depends markedly upon borehole diameter. Figure 2.12 illustrates the wellhead characteristics for 75 mm I.D. wells (with no CO₂, reservoir temperature = 200°C, piezometric surface depth = 100 m, and productivity index = 4 kg/s/bar) for each of these four feedpoint depth values. Figures 2.13 and 2.14 are similar, but for 150 mm and 300 mm I.D. wells respectively. Increasing the depth of the feedpoint has a general tendency to degrade performance. This degradation is most important for small-diameter wells, owing mainly to the relatively greater importance of lateral heat losses to the formation for small diameter wells and to the increase in lateral heat losses with feedpoint depth noted earlier (see Eq. 2.12). Larger-diameter wells are less sensitive to variations in feedpoint depth. An important exception is represented by the results for the shallowest depth considered (300 m). In this case, both for the 150 mm and 300 mm well diameters (particularly the latter), the relatively low reservoir pressure results in in-formation boiling at this depth, and significantly degraded performance.

Figure 2.5. Influence of reservoir temperature upon wellhead discharge characteristics.

Figure 2.6. Influence of reservoir pressure (depth of piezometric surface) upon wellhead discharge characteristics.

Figure 2.7. Influence of CO₂ mass fraction dissolved in reservoir liquid upon wellhead discharge characteristics.

Figure 2.8. Influence of well inside diameter upon wellhead discharge characteristics.

CASING INSIDE DIAMETER = 75 MM
 FEEDPOINT DEPTH = 600 M
 NO CO₂ PRESENT IN RESERVOIR
 RESERVOIR TEMPERATURE = 200°C
 PIEZOMETRIC SURFACE DEPTH = 100 M

Figure 2.9. Influence of reservoir permeability (productivity index) upon wellhead discharge characteristics for a minimum-diameter slim hole (75 mm ID).

Figure 2.10. Influence of reservoir permeability upon wellhead discharge characteristics for a relatively large slim hole (150 mm ID).

Figure 2.11. Influence of reservoir permeability upon wellhead discharge characteristics for a conventional production well (300 mm ID).

Figure 2.12. Influence of well feedpoint depth upon wellhead discharge characteristics for a minimum-diameter slim hole (75 mm ID).

Figure 2.13. Influence of well feedpoint depth upon wellhead discharge characteristics for a relatively large slim hole (150 mm ID).

Figure 2.14. Influence of well feedpoint depth upon wellhead discharge characteristics for a conventional production well (75 mm ID).

3

Characteristics of Geothermal Wells With Downhole Pumps

3.1 Application of Downhole Pumps to Geothermal Wells

If the reservoir temperature is low, it will frequently be necessary to install a downhole pump in the production well. Experience with downhole geothermal pumps in slim holes is lacking, so a separate study was carried out to estimate the characteristics of downhole pumps that could be constructed using existing technology for slim-hole geothermal applications (Pritchett, 1997). Mathematical models were developed to describe the performance of downhole geothermal pumps under a variety of conditions. The models are based on (1) performance characteristics for existing downhole pumps, as described in the catalogues of various downhole pump suppliers (including Johnston Pump, Gould Pump, Stang Hydronautics, Reda Pump, etc.), (2) consultation with geothermal developers who rely on downhole pumps for the success of their projects, and (3) advice from acknowledged experts in downhole pump design (Frost, 1997; Wilson, 1997). In this section, the results of that study are summarized and the implications for wellhead fluid delivery capacity (and parasitic electrical load to drive the pump) are explored.

Two different types of downhole pump have been used in geothermal wells (see Figure 1.1). Most of the practical experience is with "lineshaft" pumps, in which an electric motor is mounted vertically at the wellhead, and power is transmitted down to the submerged pump body by means of a long rotating driveshaft. More recently, "submersible" pumps have also been applied to geothermal wells - in a submersible pump, the electrical motor is also located downhole, and is mounted coaxially below the pump body itself.

In either case, the pump assembly consists of (1) a number of "bowl/impeller" assemblies arranged in series (each provides a particular pressure increment to the flowing fluid) located downhole, above which is (2) a long vertical "column" (consisting of sections of pipe which carry the upflowing brine to the surface) which fits inside the well casing. Typically, the annular space between the exterior of the column and the well casing (which carries no flow, and will be mainly filled with steam and gas) is a centimeter or two wide, and a substantial pressure difference will be present between the inside and the outside of the "column".

For submersible pumps, the "column" consists of a simple cylindrical pipe; for lineshaft pumps, the "column" contains a concentric coaxial driveshaft to propel the pump body, which is surrounded by bearings and a stationary "lube string" (another pipe which carries oil for the shaft and for the pump itself), so that in lineshaft pumps the brine travels upward in the annular region between the outer column and the inner lube string. Horizontal cross-sections through geothermal wells containing each type of downhole pump (above the bowl/impeller assemblies) are illustrated in Figure 3.1.

Figure 3-1. Horizontal cross-sections above the bowl/impeller assemblies through geothermal wells containing downhole pumps.

In the lineshaft pump design, there is no problem installing a large electric motor at the wellhead to drive the pump, but under practical conditions the motor will usually be of the four-pole type, and rotate the lineshaft at a speed in the neighborhood of 1800 RPM. For submersible pumps, in which the motor is directly connected to the bowl/impeller assembly, it is feasible to use higher-speed (~3600 RPM) two-pole motors, but available motor power is limited due to the confined space within the well casing and the high temperatures of the geothermal brines in which the motor is immersed. Thus, lineshaft pumps usually rotate at ~1800 RPM whereas submersible pumps usually operate at ~3600 RPM.

Field experience with downhole pumps of either variety in small-diameter geothermal wells is very sparse. To date, almost all geothermal downhole pump applications have involved large-diameter wells and lineshaft pumps (typically "12-inch" pumps; in the pump industry, pumps are usually designated by the size of well into which they will fit - the pump itself is somewhat smaller). Accordingly, Pritchett (1997) tried to estimate what the characteristics of various small-diameter downhole geothermal pumps would be, if a market for such a pump were to develop in the future, based upon reasonable inferences from existing technology, theoretical design studies, and practical experience. For slim-hole applications, interest extends clear down to approxi-

mately three-inch (inside diameter) completions, but it is unlikely ever to be practical to apply downhole pumping technology (at least in anything like the current form) to such small geothermal wells. Downhole pumps as small as six inches have been built in the past for various applications.

Practical experience clearly indicates that it is essential that the reservoir fluid be delivered to the wellhead in a compressed-liquid condition. Although it is in principle possible to permit two-phase flow in the upper part of the column (resulting in gas-lift which would help the pump deliver the fluid to the surface, and in lower pumping pressure requirements), such downhole boiling will almost certainly result in serious scaling problems within the pump assembly. Particularly in view of the remote locations usually involved in off-grid rural electrification projects, the resulting maintenance problems are believed to be unacceptable.

Consequently, for both submersible and lineshaft pumps, the wellhead pressure must be sufficient to maintain single-phase liquid conditions. This means that:

$$P_{wh} = P_b + 1 \text{ bar} \quad (3.1)$$

where P_{wh} is wellhead pressure and P_b is bubble-point pressure, which depends on wellhead temperature and CO_2 content in the liquid. The one-bar safety factor is added to allow for minor temporal fluctuations in wellhead temperature and CO_2 content. Note that the wellhead temperature will in general differ somewhat from the reservoir temperature due to various effects including lateral heat losses from the well to the formation, pressure-work, and power dissipation. Over the range of interest, the bubble-point pressure varies with wellhead temperature and fluid CO_2 content as shown in Figure 3.2.

Figure 3.2. Bubble-point pressure for CO_2 solutions.

Similarly, it is essential that the fluid passing through the bowl/impeller assembly be single phase liquid to avoid pump cavitation, loss of head, and damage. Therefore, the pump inlet must be located below the depth within the well where the flowing pressure would otherwise reach boiling conditions. We require that the pump inlet pressure be at least one bar in excess of the bubble-point pressure for the local temperature and CO_2 content (Figure 3.2).

3.2 Modeling Downhole Pump Performance

Whether of the lineshaft or submersible type, a downhole geothermal pump will consist of a series of “pump stages”, each consisting of an impeller/bowl assembly. Water enters the pump body at the bottom of the first stage, which imparts a particular pressure increment to the flowing fluid. The water proceeds directly from the first stage outlet into the second stage inlet, receiving another increment of pressure from the second impeller. Thus, the fluid pressure rises by a fixed amount as it passes through each stage. Finally, the water discharged from the final stage flows into the “column”, which is either a simple pipe (submersible pump) or an annular region between the column wall and the lube string/lineshaft assembly (lineshaft pump). Usually, a lineshaft geothermal pump will consist of anywhere from ten to fifty stages; a submersible pump (which operates at higher speed and therefore produces more pressure increase per stage) might consist of only a few stages. Typically, the vertical height of each stage is comparable to (or slightly less than) the diameter of the well in which it is installed, so even a large fifty-stage twelve-inch downhole pump assembly is only around fifteen meters long. These dimensions are negligible compared to typical depths of emplacement of the downhole pump for geothermal applications, so that for mathematical purposes it suffices to regard the total pressure increase imparted by the entire multi-stage assembly as occurring at a single depth in the well (the pump inlet depth).

Figure 3.3 illustrates the distributions with depth of the pressure in (1) the reservoir, (2) the well below the pump inlet, (3) the upflowing water in the column, and (4) the “annulus” between the outer column pipe and the well casing, under conditions of steady pump operation in a geothermal well. The reservoir pressure (shown in red) extrapolates to zero gauge pressure at depth = Z (the piezometric surface depth). Within the well at the feedpoint depth, the flowing pressure (green) is lower than the reservoir pressure by an amount $\dot{M} / I$ - see Equation 2.2 - and has a vertical gradient which is only slightly greater than that in the reservoir (owing to pipe friction). The pump inlet is located slightly below the depth at which the water would otherwise reach the bubble-point pressure. In the annulus between the pump assembly and the well casing, the pressure (shown in blue) declines hydrostatically above the pump inlet depth; where this hydrostatic pressure distribution reaches the local bubble-point pressure, a water surface forms with liquid water below and saturated steam above. The pressure in the annulus above the water level is almost constant with depth, owing to the low density of steam as compared with water. The effect of the operation of the downhole pump is to impart an increment of pressure (the “pump head”) to the liquid (green) flowing through the various pump stages. This “pump head” must be sufficient to maintain the wellhead pressure (P_{wh}) above the local bubble-point pressure (Eq. 3.1), taking into consideration the hydrostatic gradient in the long vertical column pipe and the effects of frictional pressure loss against the column walls and interior assembly.

Figure 3.3. Distributions of pressure in and around a geothermal well equipped with an operating downhole pump.

In order to assemble a downhole pump to impart a particular pressure increase (dictated by the necessity of maintaining the wellhead pressure above bubble point conditions), the design engineer simply combines a sufficient number of stages together to obtain the desired pumping pressure. Thus, the problem of understanding the flow rate/pressure/energy properties of the pump as a whole amounts to that of understanding the behavior of a single stage. At a fixed shaft speed (either ~1800 RPM or ~3600 RPM), a relationship exists between the volumetric flow rate passing through the pump and the pumping pressure head increase induced

by the stage - as flow rate increases, the pressure difference decreases. The “bowl efficiency” is defined as the ratio of the “hydraulic power” applied by the pump to the fluid passing through it (equal, numerically, to the product of the volumetric flow rate and the pressure rise imparted by the impeller) to the mechanical power consumed by the pump shaft (per stage). This “bowl efficiency” is low for both very low and very high flow rates, and passes through a maximum value at the optimum design conditions for the particular pump body in question. This “optimum volumetric flow rate” at which the bowl efficiency is maximized depends on the design of the bowls and the impellers, and varies considerably among the various pump models available.

Many different designs for vertical turbine downhole pumps exist. Applications for these pumps include agricultural irrigation, fire-suppression, municipal water supplies, oil production, etc. For geothermal applications, emphasis is placed on robust performance, high flow rates, high bowl pressures, and reliability. Downhole pumps have been in common use in the geothermal industry for about thirty years. Until quite recently, virtually all geothermal applications of downhole pumps have involved lineshaft pumps, typically in twelve-inch sizes. In just the last few years, the possibilities inherent in submersible pumps have started to be explored by the geothermal industry.

Figure 3.4 illustrates the performance characteristics (as functions of volumetric flow rate) for a single stage of a typical twelve-inch geothermal lineshaft pump driven at ~1800 RPM by a four-pole electric motor. This particular pump is unremarkable, lying in the middle of the performance range of similar pumps installed in geothermal wells all around the world today. The bowl efficiency attains its maximum value (83 percent) at this pump’s optimum volumetric flow rate (110 liters per second). The pumping head at these optimum design conditions is 1.4 bars per stage, and the shaft mechanical power required per stage is 19 kilowatts. If thirty of these stages were assembled to construct a lineshaft downhole geothermal pump, the pump’s design operating conditions would be a flow rate of 110 liters per second with a total pumping head of 42 bars ($= 1.4 \times 30$). Total mechanical shaft power required to drive the pump would be 570 kW ($= 19 \times 30$).

The performance data illustrated in Figure 3.4 are for four-pole motor drive (~1800 RPM; typical for lineshaft pump installations). If this same pump stage body were instead incorporated in a submersible pump assembly and driven by a two-pole (~3600 RPM) downhole motor, volumetric flow rates through the pump will double. As a consequence, the maximum attainable bowl efficiency will be the same as at the lower speed (83 percent), but the optimum flow rate will double (to 220 l/s) and the pumping head per stage will increase by a factor of four (to 5.6 bars). Both the hydraulic power per stage and the shaft power requirement per stage will increase by a factor of eight, to 125 kW and 152 kW respectively. These scaling rules are verified by considerable manufacturers’ test data.

The “optimum flow rate” for a particular downhole pump will depend upon the intended purpose for the pump as well as upon diameter. As noted above, high flow rates are important for geothermal production pumps. Figure 3.5 shows the value of the “optimum flow rate” for a variety of existing commercially-available downhole pumps (designed for various purposes) intended to fit inside wells between 6" and 12" diameter. For purposes of this plot, flow rates for pumps driven by 2-pole motors (3600 RPM) were reduced by a factor of two in accordance with the above scaling rules. The following algebraic expression (Pritchett, 1997) appears to adequately express the “design envelope” (maximum flow rate as a function of well inside diameter), as illustrated in Figure 3.5:

Figure 3.4. Single-stage operating characteristics of a typical twelve-inch lineshaft-type downhole geothermal pump driven by a four-pole electric motor (~1800 RPM).

Figure 3.5. Comparison between actual optimum flowrate data from downhole vertical turbine pumps (flowrate for maximum hydraulic efficiency) and the “design envelope” upper bound employed in the mathematical model.

$$\begin{aligned}\dot{V}(1/s) &= 4.03 \times 10^{-8} d^{3.85} \text{ (for lineshaft pumps)} \\ &= 8.06 \times 10^{-8} d^{3.85} \text{ (for submersible pumps)}\end{aligned}\quad (3.2)$$

where "d" is the inside diameter of the well in millimeters.

Actual flow rates through downhole pumps may, of course, be less than the "design envelope" limit shown in Figure 3.5. For geothermal applications, the highest possible flow rate is usually desired, but under certain conditions limitations on total power deliverable to the pump (together with minimum total pumping head requirements imposed by Eq. 3.1) may further limit pump performance. If the pump is "flow rate limited", the volumetric flow rate through the pump will be given by Eq. 3.2 (Figure 3.5). If the pump is "power limited", the optimum design flow rate will be reduced to a value less than that given by Eq. 3.2, such that the volumetric flow rate through the pump multiplied by the required pumping head will not exceed the available hydraulic power.

The pump head (per stage) at the optimum flow rate also depends on well diameter. Figure 3.6 shows values obtained from the data supplied by the various pump manufacturers, along with a mathematical representation (Pritchett, 1997):

$$\begin{aligned}\text{Pump Head per Stage (bars)} &= 0.3375 \times 10^{-5} d^{2.3} \text{ (lineshaft pumps)} \\ &= 1.3500 \times 10^{-5} d^{2.3} \text{ (submersible pumps)}\end{aligned}\quad (3.3)$$

Although the scatter in the measurements is considerable, the present model appears to adequately represent per-stage pump head for both lineshaft and submersible pumps.

The maximum attainable bowl efficiency also increases with well (or pump) diameter, albeit rather weakly. Pritchett (1997) found that bowl efficiency can be adequately expressed by:

$$\text{Bowl efficiency} = [1 - (50/d)] \times 100\% \quad (3.4)$$

where "d" is again well inside diameter in millimeters. Figure 3.7 compares this fit to the (rather scattered) data available from the various downhole pump suppliers.

The mathematical model for downhole geothermal pump performance (Pritchett, 1997) also incorporates a number of practical design constraints, listed in Table 3-1. These include limitations imposed by the mechanical strength of the lineshaft, limitations on downhole motor size, and similar issues (also see Figure 3.8). In addition to these constraints, the downhole motors and electrical cables employed in "submersible" pump designs impose temperature limitations. Current technology limits pump temperatures to about 205°C, according to manufacturers of downhole motors. Higher temperatures are likely to cause degradation of performance and premature failures. For purposes of the present calculations, this limitation was disregarded; computations were performed for reservoir temperatures up to 240°C since the state-of-the-art in downhole motors is advancing at a healthy pace at present. The reader is cautioned, however, that operation of submersible pumps above 205°C is not presently practical.

Figure 3.6. Pumping head per stage at optimum (maximum efficiency) flowrate for downhole vertical turbine pumps of various sizes at ~3600 RPM motor speed (two-pole motors - submersible pumps) and at ~1800 RPM motor speed (four-pole motors - lineshaft pumps) compared to mathematical model.

Figure 3.7. Comparison of mathematical model with measured values of maximum bowl efficiency (realized at optimum flow rate) for downhole vertical turbine pumps of various sizes.

Figure 3.8. Some practical design constraints for downhole geothermal pumps which depend on well diameter.

Table 3-1. Practical Design Constraints on Downhole Geothermal Pumps.

Constraint 1	Upper limit on output shaft power from the downhole electric motor for submersible pumps (a function of well diameter - see Figure 3.8).
Constraint 2	Maximum power transmittable by lineshaft for lineshaft pumps based on limitations in lineshaft strength (depends on the well diameter - Figure 3.8).
Constraint 3	Limitations on the total length of the lineshaft based on dynamic loading considerations (depends on the well diameter - Figure 3.8).
Constraint 4	Pump inlet depth must be shallower than well feedpoint depth.
Constraint 5	Maximum pressure difference between the inside and the outside of the column pipe (38 bars for cast-iron fabrication; 75 bars for stainless steel. Stainless steel assumed available for present calculations).
Constraint 6	Practical assembly limitations to fifty stages total (lineshaft pumps) or 65 stages (submersible pumps).

3.3 Approximate Analysis of Design Constraint Effects

To appraise the role of the various design constraints (Table 3-1) in evaluation of downhole pump performance, it is useful to perform a preliminary approximate analysis of the relations among various downhole pump parameters under operating conditions. For this purpose, for the moment we make the following simplifying assumptions:

- The fluid flowing up the well and through the column to the wellhead is an isothermal incompressible liquid.
- All frictional effects are neglected - pipe friction in the well and column, and friction in the formation (the productivity index of the well is infinite).

Under these conditions, it may be shown (see Figure 3.3) that the pump head required (ΔP) will be approximately equal to:

$$\Delta P \approx P_b + GZ \quad (3.5)$$

The approximate pump inlet depth will be given by:

$$D_i \approx Z + (P_b/G) \quad (3.6)$$

and the hydraulic power required of the pump will be approximately:

$$\dot{E}_h \approx \dot{V}(P_b + GZ) \quad (3.7)$$

In these expressions,

P_b = bubble-point pressure (a function of temperature and CO_2 content - see Figure 3.2).

G = hydrostatic pressure gradient (positive); for temperatures between 100°C and 240°C , will range from 0.094 bars/meter to 0.080 bars/meter.

Z = depth of piezometric surface.

$\dot{V}$ = volumetric flow rate through the pump.

For practical purposes, the performance limitations listed as “constraints 1 and 2” in Table 3-1 (maximum power for downhole motor; lineshaft strength limitation) constitute limitations on the maximum value which the pump hydraulic power $\dot{E}_h$ may reach. If this limiting value is denoted by $\dot{E}_{\max}$, then it follows that an upper bound is imposed on the volumetric flow rate through the pump, given by:

$$\dot{V}_{\max} \approx \frac{\dot{E}_{\max}}{P_b + GZ} \quad (3.8)$$

If this value exceeds that given by the “design envelope” (Eq. 3.2; Fig. 3.5), then the pump is “flow rate limited” and can operate at the “design envelope” value. Otherwise, however, the pump is “power limited” and should be designed for a lower optimum volumetric flow rate ($\dot{V}_{\max}$ above).

Performance constraint 3 (Table 3-1) imposes an upper bound on the pump inlet depth, as does constraint 4. Let $D_{\max}$ represent the more restrictive of these two limitations; it follows that the maximum value permissible for the bubble-point pressure (P_b) is:

$$P_{\max_1} \approx G(D_{\max} - Z) \quad (3.9)$$

In other words, for a particular reservoir CO_2 content, the pump cannot be operated for reservoir temperatures that would yield $P_b > P_{\max_1}$ (see Fig. 3.2). Otherwise, cavitation and two-phase flow will take place within the pump’s bowl/impeller assemblies, reducing pump effectiveness and probably destroying the pump within a short period of time.

Finally, performance limitations (5) and (6) in Table 3-1 (maximum pressure difference across column wall; maximum number of pump stages) amount to limitations on the total pumping head (ΔP) which the pump can impart to the upflowing liquid passing through it. Denoting this limiting value by $\Delta P_{\max}$, it follows that an additional constraint is imposed on the bubble-point pressure:

$$P_{\max_2} \approx \Delta P_{\max} - GZ \quad (3.10)$$

which constrains the operational range of the pump (in terms of CO₂ content and reservoir temperature) in the same way as Eq. 3.9 above.

3.4 Downhole Pump Electrical Power Consumption

The parasitic electrical load imposed by the operation of a downhole pump must be deducted from the power plant electrical generating capacity. The relationship between the electrical power required to operate the downhole pump and the "hydraulic power" produced by the pump (equal to the pressure increase imparted to the flowing fluid by the pump, multiplied by the volumetric flow rate through the pump) involves power losses of various kinds, including:

- *Bowl efficiency* (efficiency of conversion of mechanical power to hydraulic power by the submerged bowl/impeller assemblies; discussed above),
- *Shaft efficiency* (losses to friction in thrust bearings and lineshaft bearings),
- *Motor efficiency* (efficiency of conversion from electrical power to mechanical power), and
- *Transmission efficiency* (electrical losses between the generator and the pump motor - can be important for submersible pumps).

Mathematical representations for these processes have been incorporated in the model as discussed by Pritchett (1997).

3.5 Modeling Well Performance Using Downhole Pumps

The next task is to calculate the performance of a geothermal well (characterized by particular values of inside diameter, feedpoint depth, piezometric surface depth, productivity index, reservoir temperature and reservoir CO₂ content) in which a high-performance downhole pump (of either the submersible or lineshaft design) has been installed. The simplifying approximations imposed above for the purposes of Section 3.3 (neglect of friction and restriction to incompressible isothermal systems) are now relaxed. Quantities of interest include the wellhead pressure, wellhead temperature, and wellhead discharge rate, as well as the pump inlet depth required and the parasitic electrical load required for pump operation.

Despite the additional complications imposed by the pump, the calculation of well performance is actually much easier in the "pumped" case than in the "self-discharging" case, since two-phase (water/steam) flow is not involved. The fluid which emerges from the wellhead consists of compressed liquid water containing dissolved CO₂ gas at the same concentration as the reservoir fluid. The wellhead pressure will be given by Equation 3.1 (wellhead pressure = local bubble-point pressure plus one bar) and the bubble-point pressure depends on the CO₂ content and the wellhead temperature (Figure 3.2).

Mass and heat flow in the reservoir and in the portion of the well below the pump inlet depth are treated in exactly the same way as in the self-discharging case, with the simplification that the flow will always be single-phase liquid. In the column pipe, the momentum equation similarly provides the vertical flowing pressure gradient as the sum of (1) hydrostatic gradient (which depends on temperature), (2) friction (treated as described by Pritchett, 1997), and a “fluid acceleration” term which may be neglected owing to the absence of two-phase effects. Of course, the fluid pressure jumps discontinuously at the pump inlet depth, by the amount required to maintain the flowing wellhead pressure at the required value (Eq. 3.1).

The energy balance, as before, contains three terms: a “kinetic energy change” term (which may be neglected since two-phase flow does not occur), work done against gravity, and lateral heat losses to the formation. Lateral heat losses will still occur even from the rising liquid within the column. The reason is that the annular space between the column pipe and the well casing is sealed at the surface, and contains saturated steam at high pressure (Figure 3.3). Consequently, steam will tend to condense along the inner casing surface and to evaporate along the exterior of the column. The resulting “heat pipe” should be very efficient at facilitating heat transfer across the annular space into the rock formations outside. Additional sources of heat are also incorporated in the mathematical model (both as point-sources near the pump inlet depth and distributed vertically between the inlet depth and the wellhead) which tend to increase the enthalpy of the upflowing liquid in the column. These represent the pressure-work done on the fluid by the impellers (equal to the pumping head multiplied by the volumetric flow rate through the pump), frictional heating in the thrust bearing and in the lineshaft bearings (if present), and electrical resistive heating in the motor and in the power transmission cable (for submersible pumps).

3.6 Performance of Geothermal Wells Equipped With Downhole Pumps

Extensive detailed numerical calculations for both lineshaft and submersible pumps have been carried out as part of this project. Results of these calculations of pump performance are presented graphically in Appendices C and D (for lineshaft pumps) and Appendices G and H (for submersible pumps). Results are presented for the following quantities:

- Mass discharge rate (kilograms per second),
- Pump inlet depth (meters),
- Flowing wellhead pressure (bars),
- Temperature loss (Celsius) between feedpoint and wellhead,
- Total pumping pressure head imparted by impellers (bars),
- Number of bowl/impeller stages,
- Total parasitic electrical load required to drive pump (kilowatts), and
- Overall electric/hydraulic efficiency for the downhole pump (percent).

Calculations were carried out in all cases for piezometric surface depths ranging from zero to 250 meters and for reservoir temperatures from 100°C to 240°C. Appendices C and G provide results for “Series 1” calculations, with well feedpoint depth = 300, 600, 900 or 1200 meters, productivity index = 2 kg/s/bar, 4 kg/s/bar or infinity, and no reservoir CO₂. Appendices D and H display “Series 2” results which incorporate the effects

of noncondensable gas (reservoir CO_2 contents of zero, 0.25, 0.50, 0.75 and 1.00 percent by mass) with well feedpoint depth = 600 meters and productivity index = 4 kg/s/bar. Well inside diameter values considered for Appendices C and D (lineshaft pumps) are 150 mm, 200 mm, 250 mm and 300 mm; for submersible pumps (Appendices G and H) diameters of 100 mm, 125 mm, 150 mm, 200 mm, 250 mm and 300 mm are included. It should be pointed out that the performance results for submersible pumps for diameters less than 150 mm are somewhat speculative, since no actual test data were available for pumps smaller than six-inch size.

It is important to emphasize that these computed results pertain to "design limitations", not to the effects of varying operating conditions upon a particular downhole pump. In other words, for each combination of pertinent parameters (feedpoint depth, reservoir temperature, etc.), the computations provide the estimated performance of a downhole pump for which the design was optimized for that particular set of conditions. If the conditions are different, a different pump design will result.

In this section, a selection of computed results is presented to illustrate the general capabilities of wells with downhole pumps and the influence of the various key free parameters upon performance. Figure 3.9 shows, as functions of reservoir temperature, the net electrical generating capacity of a wellhead binary plant (which will be described in Section 6) for production wells of diameter 150 mm, 200 mm, 250 mm and 300 mm equipped with both lineshaft and submersible downhole pumps. For these curves, the piezometric surface depth (100 m), productivity index (4 kg/s/bar) and well feedpoint depth (600 meters) are all fixed, and no reservoir CO_2 is present. These results clearly show that, while lineshaft-type downhole pumps are slightly advantageous for large-diameter production wells (up to 25 percent advantage for 300 mm well diameter), for smaller well diameters the submersible pumps are to be preferred, with net power output exceeding that available from lineshaft pumps by a factor of two or so for diameters of 200 mm or less. Also, the maximum reservoir temperature for which lineshaft pumps can be used declines with decreasing well diameter, and is only around 135°C for 150 mm diameter slim holes. At higher temperatures, the dynamic-loading limitation on total lineshaft length (Table 3-1, constraint 3) comes into play restricting the well inlet depth (see Equation 3.9) to shallow values above the boiling level in the well. The performance of all submersible pumps for diameters of 200 mm, 250 mm and 300 mm is "motor power limited" under these conditions; only for the smallest (150 mm diameter) well shown in Figure 3.9 can the "design envelope" flow rate be attained (Eq. 3.2; Fig. 3.5). For the lineshaft pumps, for those temperature conditions under which such a pump can be used, the pumps are all "flow rate limited" at the "design envelope" flow rate, but of course this limiting flow rate for a lineshaft pump is only one-half of that for a submersible pump of the same diameter.

Figure 3.10 illustrates the performance characteristics of lineshaft pumps as functions of diameter in a somewhat different way. The same conditions apply (feedpoint depth, 600 m; piezometric surface depth, 100 m; productivity index, 4 kg/s/bar; no CO_2). The curves show, as functions of reservoir temperature, (A) the wellhead discharge rate, (B) the electrical requirements for the wellhead motor, (C) the pump inlet depth, and (D) the wellhead temperature decrease (relative to feedpoint temperature) for well ID values of 150, 200, 250 and 300 mm. The upper limit on the feasible temperature range in each case is imposed by a "maximum inlet depth" constraint, as noted above; for the 150, 200 and 250 mm diameters this limiting depth is imposed by the lineshaft dynamic loading constraint (Table 3-1, constraint 3). For the largest diameter (300 mm), the limiting depth is the well feedpoint depth in this case (Table 3-1, constraint 4); the dynamic loading constraint would permit emplacement of a 300 mm lineshaft pump as deep as 715 meters.

BINARY PLANT DRIVEN BY WELL EQUIPPED WITH DOWNHOLE PUMP

PIEZOMETRIC SURFACE DEPTH: 100 m

WELL FEEDPOINT DEPTH: 600 m

WELL PRODUCTIVITY INDEX: 4 kg/s/bar

RESERVOIR CO₂ CONTENT: None

Figure 3.9. Effect of well diameter and downhole pump design on net power available from wellhead binary geothermal generating plants.

Figure 3.10. Influence of well inside diameter upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for wells equipped with lineshaft-type downhole pumps.

Figure 3.11 is similar, but pertains to wells equipped with submersible pumps. Also, results are presented for 100 mm well diameter as well as the larger values. The dotted lines indicate the region in which temperatures exceed 205°C, where state-of-the-art advance will be required in the temperature tolerance of downhole motors and cables for reliable operation. For the three largest diameters (200, 250, 300 mm) the temperature range is limited by inlet depth limitations (feedpoint depth), although in each case the limiting temperature imposed by this constraint exceeds 205°C. Performance is “motor power limited” for these same three largest diameter pumps and is “flow rate limited” for the smaller pumps (although the 150 mm pump becomes “motor power limited” for reservoir temperatures exceeding 225°C or so).

It should be noted, in both cases (Figures 3.10 and 3.11), that the amount of temperature decline between feedpoint and wellhead increases with decreasing well diameter. This occurs for the same reason as for self-discharging wells - the surface-area to volume ratio increases as well diameter decreases, making lateral heat loss to the surrounding cooler rocks more effective.

Figures 3.12 and 3.13 (for well diameters of 300 mm and 150 mm respectively) show the same information in such a way as to facilitate the comparison of lineshaft and submersible pump performance in (3.12) conventional production wells and (3.13) slim holes. Clearly, existing lineshaft pump technology outperforms existing submersible pump technology for 300 mm diameters. Equally clearly, the reverse is true for slim holes. It was largely for this reason that detailed calculations concerning lineshaft pumps in holes of less than 150 mm diameter were not carried out during this project.

Next, consider the effect of varying the depth of the piezometric surface (reservoir pressure; Z) from the above “base case” value of 100 meters. Figure 3.14 illustrates the performance of standard production wells (300 mm ID) equipped with lineshaft pumps. Other parameters are the same as before (feedpoint depth = 600 m, productivity index = 4 kg/s/bar, no CO_2). In these cases, pump performance is “flow rate limited” for $Z = 0, 50$ m, 100 m, and (except for temperatures $> 168^\circ\text{C}$), 150 m. For lower reservoir pressures ($Z = 200$ and 250 m), performance is limited by lineshaft strength considerations (Table 3-1, constraint 2; Figure 3.8). The temperature range is limited by the feedpoint depth. Similarly, Figure 3.15 shows the performance of 300 mm ID standard production wells equipped with submersible pumps. In all of these cases, performance is “motor power limited” (Table 3-1, constraint 1).

Figures 3.16 and 3.17 (for “lineshaft” and “submersible” pumps respectively) are similar, but for 150 mm ID slim holes. Figure 3.16 shows that, for slim-hole lineshaft pumps, the liquid discharge rate is essentially independent of reservoir pressure since the well performance is “flow rate limited”. The pump inlet depth, on the other hand, is sharply limited by dynamic loading constraints (Table 3-1, constraint 3) which results in very limited temperature ranges of utility. The maximum reservoir temperature is less than 100°C for piezometric surface depths of 150 meters or greater. The situation is much better for 150 mm submersible pumps (Fig. 3.17); as reservoir pressure decreases (increasing Z), motor power limitations are encountered at lower and lower reservoir temperatures.

Figure 3.11. Influence of well inside diameter upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for wells equipped with submersible-type downhole pumps.

Figure 3.12. Influence of downhole pump type (lineshaft or submersible) upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for 300 mm ID standard production wells.

Figure 3.13. Influence of downhole pump type (lineshaft or submersible) upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for 150 mm ID slim holes.

Figure 3.14. Influence of reservoir pressure upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for 300 mm ID standard production wells equipped with lineshaft-type downhole pumps.

Figure 3.15. Influence of reservoir pressure upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for 300 mm ID standard production wells equipped with submersible-type downhole pumps.

Figure 3.16. Influence of reservoir pressure upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for 150 mm ID slim holes equipped with lineshaft-type downhole pumps.

Figure 3.17. Influence of reservoir pressure upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for 150 mm ID slim holes equipped with submersible-type downhole pumps.

Figure 3.18 - 3.21 illustrate the effect of non-zero reservoir noncondensable gas content on pumped-well performance. Raising the CO_2 content has similar effects to raising the reservoir temperature, since both increase the bubble-point pressure. Figure 3.18 shows the performance of a 300 mm large-diameter production well with a lineshaft pump. As CO_2 content increases, the bubble point pressure likewise increases, so the pump inlet depth for a particular reservoir temperature increases. The inlet depth reaches the limiting value (600 m - the feedpoint depth in this case) at lower temperatures for higher CO_2 values. The lineshaft strength constraint (constraint 1, Table 3-1) also comes into play for CO_2 content = 0.50 percent or greater. Figure 3.19 (again for 300 mm ID but for submersible pumps) shows similar trends. Performance is "motor power limited" throughout; the temperature range is broader only because the total flowrate is lower than in the lineshaft case, so the pump inlet depth does not reach the limiting value until the CO_2 content is somewhat higher than for the corresponding lineshaft case.

The effects of CO_2 variation are similar for slim holes. Figure 3.20 (for lineshaft pumps) shows that increasing the CO_2 content to only 0.25 percent reduces the operational temperature maximum below 100°C . For submersible pumps (in 150 mm ID holes; Figure 3.21), raising the CO_2 content forces the pump inlet depth to increase, increases the required pumping head, and causes the onset of "motor power limited" behavior at lower and lower temperatures.

The effects of varying the feedpoint depth are minor, except insofar as the feedpoint depth constitutes a limitation on the pump inlet depth. Figures 3.22 - 3.25 show the results of varying feedpoint depth from 300 m to 1200 m for all four cases (150 mm, 300 mm ID; lineshaft and submersible pumps). All other parameters are the same as before (no CO_2 , piezometric surface depth = 100 m, and productivity index = 4 kg/s/bar). Figure 3.22 (for 300 mm ID production wells with lineshaft pumps) shows that the performance for 900 and 1200 meter feedpoint depths is almost identical, and that the 600 meter case is also the same up to $T = 189^\circ\text{C}$ (at which point the pump inlet depth reaches 600 m). Since, even at 100°C , the inlet depth exceeds 300 meters, the feedpoint depth = 300 meters case is not suitable for lineshaft pump application in this case. In all cases shown in Figure 3.22, performance is "flow rate limited" for temperatures below 193°C or so; at higher temperatures, constraint 1 (Table 3-1; "maximum lineshaft strength") comes into play and performance becomes "power limited". Figure 3.23 shows similar results for 300 mm submersible pump installations - all cases are "motor power limited". Effects of feedpoint depth are slightly more noticeable for 150 mm slim holes (Figures 3.24 and 3.25), due mainly to the greater importance of lateral heat losses for slim holes and the greater heat losses to be expected for deeper wells due to increased surface area.

Finally, consider the effect of changing the productivity index. All other parameters are fixed (feedpoint depth = 600 m, piezometric surface depth = 100 m, and no CO_2) for the results shown in Figures 3.26 - 3.29. Figure 3.26 (lineshaft pumps in 300 mm ID wells) clearly shows that pump behavior is "flow rate limited" in these cases, so the effect of reducing the productivity index is qualitatively similar to that of reducing the reservoir pressure (that is, increasing the piezometric surface depth). The pump inlet depth is a strong function of productivity index, and reaches 600 meters (the feedpoint depth) at 100°C for productivity index > 2 kg/s/bar. The situation is quite different for large-diameter wells with submersible pumps (Figure 3.27). Here, for productivity index = 2 and 4 kg/s/bar, performance is "motor power limited" for all temperatures; for the infinite productivity index case, performance is "motor power limited" for reservoir temperatures exceeding 140°C .

and is “flow rate limited” for lower reservoir temperatures. The pump inlet depth is a strong function of productivity index (as was the case for the lineshaft pumps). It is also noteworthy that, in the case of high reservoir permeability (infinite productivity index), submersible pumps will actually outperform lineshaft pumps in these large-diameter wells (compare Figure 3.27A with 3.26A), at least up to 205°C reservoir temperature.

For slim holes, the effects of productivity index are much less dramatic (for essentially the same reasons as in the case of self-discharging wells). Figures 3.28 and 3.29 show results for lineshaft pumps and submersible pumps respectively in 150 mm ID slim holes. For lineshaft pumps (Figure 3.28), behavior is “flow rate limited” and the reservoir temperature range of feasibility is constrained by pump inlet depth limits imposed by “constraint 3” (Table 3-1; lineshaft dynamic loading). For submersible pumps, the inlet depth is slightly influenced by productivity index, and the temperature at which performance shifts from “flow rate limited” to “motor power limited” increases somewhat with increasing productivity index.

Figure 3.18. Influence of noncondensable reservoir gas upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for 300 mm ID standard production wells equipped with lineshaft-type downhole pumps.

Figure 3.19. Influence of noncondensable reservoir gas upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for 300 mm ID standard production wells equipped with submersible-type downhole pumps.

Figure 3.20. Influence of noncondensable reservoir gas upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for 150 mm ID slim holes equipped with lineshaft-type downhole pumps.

Figure 3.21. Influence of noncondensible reservoir gas upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for 150 mm ID slim holes equipped with submersible-type downhole pumps.

Figure 3.22. Influence of well feedpoint depth upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for 300 mm ID standard production wells equipped with lineshaft-type downhole pumps.

Figure 3.23. Influence of well feedpoint depth upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for 300 mm ID standard production wells equipped with submersible-type downhole pumps.

Figure 3.24. Influence of well feedpoint depth upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for 150 mm ID slim holes equipped with lineshaft-type downhole pumps.

Figure 3.25. Influence of well feedpoint depth upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for 150 mm ID slim holes equipped with submersible-type downhole pumps.

Figure 3.26. Influence of reservoir permeability upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for 300 mm ID standard production wells equipped with lineshaft-type downhole pumps.

Figure 3.27. Influence of reservoir permeability upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for 300 mm ID standard production wells equipped with submersible-type downhole pumps.

Figure 3.28. Influence of reservoir permeability upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for 150 mm ID slim holes equipped with lineshaft-type downhole pumps.

Figure 3.29. Influence of reservoir permeability upon (A) liquid discharge rate at wellhead, (B) pump electrical requirements, (C) pump inlet depth and (D) wellhead temperature loss relative to feedpoint depth for 150 mm ID slim holes equipped with submersible-type downhole pumps.

4

Single-Flash Steam Plants

4.1 Utility of Single-Flash Steam Plants

Single-flash steam-turbine electrical generating plants are the most common type of electrical generating installation in the geothermal industry. The simplest geothermal system available is a self-discharging well driving a backpressure-type steam turbine, illustrated schematically in Figure 4.1. A mixture of hot water and steam flows up the wellbore to the wellhead, and into a wellhead cyclone separator. The separated liquid is discarded (in Figure 4.1, in an injection well—surface disposal is also possible). The separated steam drives a backpressure steam turbine—the spent steam is exhausted to the atmosphere, at one atmosphere pressure. The turbine, in turn, drives an electrical generator (not shown).

A more complicated system, illustrated schematically in Figure 4.2, is a single-flash condensing steam turbine. In this case, the steam from the turbine does not exhaust directly to the atmosphere. Instead, it enters a “condenser” which is maintained at a fixed temperature (usually between 35°C and 45°C—a condenser temperature of 39°C was assumed for the present calculations). Because of the low temperature, the pressure in the condenser is substantially below atmospheric pressure, so that the steam expands to a greater volume than in the backpressure case and does more mechanical work on the turbine blades. In the condenser, the steam is condensed back to liquid water, leaving only noncondensable gases (such as CO₂) in the vapor phase. The noncondensable gas (NCG) is pumped out of the condenser and may be either exhausted to the atmosphere or, if significant toxic components such as H₂S are present, passed through any of several possible chemical abatement systems. The condensed liquid water (the “steam condensate”) which is pumped out of the condenser may be provided as make-up water to the cooling system as needed—any excess may be reinjected. Sometimes, separate disposal systems are provided for the excess steam condensate and the separated brine; the streams may also be combined and injected together, as indicated in Figure 4.2. A cooling system (not shown) is necessary to maintain the condenser at 39°C and dissipate the latent heat of vaporization of the steam. Usually, cooling systems are of the liquid-evaporative type, but air-cooling is also possible. Advantages of single-flash condensing steam turbines over backpressure units include more efficient use of wellhead steam and elimination of the air-pollution hazard from the atmospheric exhaust of the backpressure turbine. Disadvantages include greater capital cost, higher maintenance costs, increased mechanical complexity, and the requirement to provide a cooling system.

It is also possible to operate single-flash steam turbine wellhead geothermal generating plants using wells equipped with downhole pumps. This is illustrated in Figures 4.3 (for backpressure steam turbines) and 4.4 (for condensing units). Here, the hot geofluids are delivered to the wellhead in a compressed-liquid condition by the downhole pump (which may be of either the lineshaft or submersible type). Prior to entering the separator, the fluid is expanded through an orifice or a valve to create a two-phase situation at the desired operating pressure within the separator. Thereafter, the system is the same as for the self-discharging case.

Figure 4.1. Schematic diagram of a backpressure single-flash steam turbine wellhead generator driven by a self-discharging geothermal well.

Figure 4.2. Schematic diagram of a condensing single-flash steam turbine wellhead generator driven by a self-discharging geothermal well.

Figure 4.3. Schematic diagram of a backpressure single-flash steam turbine wellhead generator driven by a geothermal well equipped with a downhole pump.

Figure 4.4. Schematic diagram of a condensing single-flash steam turbine wellhead generator driven by a geothermal well equipped with a downhole pump.

4.2 Steam Expansion Through the Turbine

For single-flash steam turbines, only the separated steam is important; the liquid water from the separator plays no role in the power generation process. As noted earlier, it has been assumed that all surface generating equipment will be located immediately adjacent to the production wellhead, so that pressure losses between the wellhead and the separator and between the separator and the turbine inlet may be neglected. Thus, the turbine inlet gas consists, in general, of a saturated $\text{H}_2\text{O}/\text{CO}_2$ mixture (no liquid, no superheat) at wellhead pressure and temperature. For a particular type of power plant, the power available will therefore depend only on the wellhead pressure (inlet pressure), the CO_2 content of the steam, and the steam flow rate.

As the steam enters the turbine and begins to expand, mechanical work is done on the turbine blades. For an "ideal" system, the work done is given by:

$$dW = P dV \quad (4.1)$$

where "W" is mechanical work (*i.e.* Joules), P is pressure, and V is specific volume of the steam (*i.e.* cubic meters per kilogram). As the expanding steam does mechanical work, the specific internal energy declines correspondingly:

$$dE = -dW \quad (4.2)$$

Since the inlet steam is saturated, this decline in internal energy will result in the immediate condensation of liquid droplets within the turbine, so that two-phase conditions will prevail throughout the expansion. Calculations of the expansions of gaseous mixtures consisting of varying proportions of CO_2 at various starting pressure values were carried out numerically, integrating equations (4.1) and (4.2). These integrations start at inlet pressure conditions, and follow the expansion until (1) the pressure reaches one atmosphere ("backpressure" turbine case) or (2) the temperature reaches the value maintained in the condenser by the cooling system ("condensing" turbine case). For all cases reported herein, the condenser temperature was taken to be 39°C . In the absence of CO_2 , this corresponds to a condenser pressure of 0.0698 bars. This value is representative of the state of the art for small condensing turbines (Forsha, 1994; Nichols, 1995).

Numerical integrations were performed for a wide variety of different combinations of inlet pressure and CO_2 content. Figure 4.5 shows the results of a typical expansion calculation. In this case, the inlet pressure is 3 bars and the inlet gas consists of 90% H_2O and 10% CO_2 by mass. It should be noted that the CO_2 mass fractions of practical interest for the separated steam are much higher than those in the deep liquid reservoir (maximum of 1% CO_2 in the present cases). The reason is that, when liquid water with dissolved CO_2 boils, the CO_2 preferentially enters the gas phase so that, even if only a small fraction of the total H_2O present has boiled to steam, almost all of the CO_2 originally present will be found in the gas phase and the remaining liquid will be almost completely depleted of CO_2 .

Figure 4.5 shows changes in conditions during the expansion as functions of specific volume. Initially, the specific volume of the $\text{CO}_2/\text{H}_2\text{O}$ gaseous mixture at 3 bars pressure is 0.569 cubic meters per

ADIABATIC EXPANSION OF VAPOR SEPARATED AT THREE BARS
VAPOR COMPOSITION BY MASS: 90% H₂O, 10% CO₂

Figure 4.5. Adiabatic expansion of steam with 10% CO₂ (by mass) from 3 bars initial pressure to “backpressure turbine” outlet conditions (1.013 bars) and to “condensing turbine” outlet conditions (39°C).

kilogram, the temperature is 132.03°C, and no liquid is present. During the early part of the expansion, the pressure declines rapidly. When the pressure reaches 1.01325 bars (one atmosphere; the end of the integration for the “backpressure” case), the specific volume has increased to 1.477 cubic meters per kilogram (~2.6 times the initial volume) and the temperature has declined to 98.7°C (slightly less than the 100°C value which would have been reached at this pressure were it not for CO₂ partial pressure effects). The total work done by the expansion up to this point amounts to 152.5 Joules per gram of inlet CO₂/H₂O mixture. Also, 5.6% of the total inlet mass has condensed to liquid by this stage of expansion.

Further expansion results in further decline in pressure and temperature, increase in liquid condensate, and additional mechanical work. When the specific volume reaches 15.33 cubic meters per kilogram (~27 times greater than the initial volume), the temperature reaches condenser temperature (39°C). The pressure is now only 0.0737 bars (slightly higher than the 0.0698 bar value in the pure-H₂O case); 15.4 percent of the inlet gas has now condensed to liquid, and the total mechanical work done has reached 456.7 Joules per gram of inlet vapor (three times greater than the “backpressure” value).

4.3 Plant Performance

Of course, the actual electrical generating capacity is significantly lower than the theoretical work rate of expansion. For a backpressure turbine, the net electrical power may be represented by:

$$\dot{E}_{\text{net}} = \dot{M}_v W_{\text{bp}} e_t e_b e_g \quad (4.3)$$

where

- $\dot{M}_v$ = mass flow rate through turbine,
- W_{bp} = specific work done by expansion to 1.01325 bars,
- e_t = turbine efficiency
- e_b = gearbox efficiency, and
- e_g = generator efficiency.

For condensing turbines, the available work of expansion is greater, but parasitic loads associated with operating the condenser/cooling system must be accommodated:

$$\dot{E}_{\text{net}} = \dot{M}_v \times (W_{\text{cn}} e_t e_b e_g - L) \quad (4.4)$$

where

- W_{cn} = specific work done by expansion to 39°C, and
- L = parasitic load.

Based on design studies for small steam turbogenerators reported by Forsha (1994), and after consultation with Nichols (1995), the following efficiency values were selected for use in this study (for both backpressure and condensing designs):

$$\begin{aligned} e_t \text{ (turbine efficiency)} &= 0.75 \\ e_b \text{ (gearbox efficiency)} &= 0.98 \\ e_g \text{ (generator efficiency)} &= 0.96 \end{aligned}$$

The parasitic loads for the condensing case ("L" in Eq. 4.4 above) consist of three parts—the electrical power required to operate the condensate pump, the NCG vacuum pump, and the cooling tower fans. The load for the condensate pump and the cooling tower fans will be in proportion to the H_2O flow rate out of the condenser; load factors of 8 kWe/kg/second for the condensate pump and of 5 kWe/kg/second for the cooling tower fans (13 kWe per kilogram of H_2O per second total) were adopted from Forsha (1994). In contrast, the parasitic load required to operate the NCG vacuum pump will be in proportion to the amount of NCG (*i.e.*, CO_2) entering the condenser. This was calculated by computing the adiabatic work required to pump pure CO_2 gas from condenser pressure and temperature up to one atmosphere pressure, and then applying an overall efficiency factor of 65% (incorporating both motor efficiency and pump efficiency). Results are shown in Figure 4.6. The NCG pump load per unit amount of CO_2 is fairly insensitive to condenser temperature over the range of interest, and decreases somewhat with increasing condenser pressure. For low CO_2 concentrations at 39°C, the load factor is about 255 kWe per kilogram of CO_2 per second. Examples of the total parasitic loads are provided in Table 4-1. Note that total parasitic load increases substantially as the CO_2 mass fraction in the turbine inlet vapor increases.

Figure 4.7 shows the resulting electrical capacity (per kilogram of inlet vapor per second) for both backpressure and condensing turbines as functions of inlet pressure and CO_2 mass fraction in the steam. For systems using self-discharging wells, Figure 4.7 represents the electric power available for export; if a downhole pump is used, the parasitic load of the downhole pump remains to be deducted. The generating capacity increases with increasing pressure and declines with increasing CO_2 content. The reason why output power decreases with increasing CO_2 content for backpressure turbines is simply that the molecular weight of CO_2 (44) exceeds that of H_2O (18) so that, as CO_2 content increases, the average molecular weight increases and the volume occupied by a particular mass of vapor decreases. In the condensing case, this effect is augmented by the increasing parasitic load of the NCG pump with increasing gas content (see above) and by the fact that the presence of CO_2 tends to increase the pressure in the condenser, reducing steam expansion in the turbine.

Figure 4.6. Parasitic electrical demand of NCG vacuum pump for condensing steam turbine plants.

Table 4.1. Examples of parasitic loads for condensing single-flash steam turbine generating plants.

	Case A	Case B	Case C
Turbine Inlet Flows:	(kg/s)	(kg/s)	(kg/s)
Inlet H ₂ O Flow Rate	20	19	18
Inlet CO ₂ Flow Rate	0	1	2
Total Inlet Flow Rate	20	20	20
Parasitic Loads:	(kWe)	(kWe)	(kWe)
Cooling Tower Fans	100	95	90
Condensate Pump	160	152	144
NCG Vacuum Pump	0	250	496
Total Parasitic Load	260	497	730

Figure 4.7. Net electrical power available from backpressure and condensing turbogenerators as functions of steam inlet pressure and CO_2 mass fraction. No deduction for parasitic load of downhole pump (if present).

5

Double-Flash Steam Plants

5.1 Types of Double-Flash Plants

Double-flash steam turbine geothermal power plants are less common than single-flash plants, but several are in use around the world. Double-flash plants make use of the heat in the separated liquid brine from the separator which is rejected by single-flash designs, and thereby make more efficient use of the geofluids produced from the well. Two basic types of double-flash plant are considered, which differ in the type of power cycle employed in the high-pressure (primary) stage. If the first stage consists of a backpressure steam turbine and the second (low-pressure) stage is a condensing steam turbine, we will call the system a “backpressure/condensing double-flash plant”; if both stages use condensers, the system will be referred to simply as a “condensing double-flash plant”. Schematic diagrams of both types are illustrated in Figures 5.1 and 5.2. As in the case of single-flash steam turbine plants, it is feasible to operate double-flash plants using compressed liquid from wells with downhole pumps by expanding the compressed wellhead liquid into the primary (high-pressure; “HP”) separator, as indicated in Figures 5.3 and 5.4. The advantage of using double-flash plants (instead of simpler single-flash plants) is the more efficient use of steam, as discussed above. Disadvantages include greater capital cost and mechanical complexity. Also, due to the more extreme degree of pressure and temperature reduction in the low-pressure separator, scaling problems are often encountered in double-flash plants which increase maintenance costs.

The “condensing” type of double-flash plant is by far the most common; the “backpressure/condensing” concept has been proposed as potentially useful for reservoirs containing large quantities of noncondensable (but non-toxic) gas. As will be seen, the present calculations indicate that the “backpressure/condensing” configuration is rarely advantageous, but it is included for the sake of completeness. In either case, the water/steam mixture from the wellhead of a self-discharging well is introduced directly into the primary high-pressure (“HP”) separator (or alternatively the liquid from a downhole pump is expanded into the HP separator). The liquid from the HP separator (sent directly to disposal in single-flash designs) is then expanded into a secondary “low-pressure” (“LP”) separator and additional steam is obtained. Finally, the liquid water from the LP separator is sent to disposal.

In the “backpressure/condensing” design, the steam from the HP separator is supplied to a conventional backpressure steam turbine with an atmospheric exhaust. Since virtually all the noncondensable gas (“NCG”) in the inflowing reservoir fluid will be partitioned into the gas phase in the HP separator, this means that all NCG will be discharged to the atmosphere at this stage and provide no further complications to the plant (although they could clearly cause complications of other kinds). The steam from the LP separator drives a separate low-pressure condensing steam turbine, with a condenser, a condensate pump, and a cooling system (and possibly a small residual NCG extraction system—not shown in Figures 5.1 and 5.3 nor taken into account in the mathematical model owing to the negligible amounts of NCG remaining).

Figure 5.1. Schematic diagram of a backpressure/condensing double-flash steam turbine wellhead generator driven by a self-discharging geothermal well.

Figure 5.2. Schematic diagram of a condensing double-flash steam turbine wellhead generator driven by a self-discharging geothermal well.

Figure 5.3. Schematic diagram of a backpressure/condensing double-flash steam turbine wellhead generator driven by a geothermal well equipped with a downhole pump.

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The “condensing” double-flash plant has no direct atmospheric turbine exhaust. The high-pressure steam and the low-pressure steam both drive condensing steam turbines. Figures 5.2 and 5.4 suggest that two separate turbines are involved (a HP turbine and a LP turbine). Although it suffices to treat the system in this way for mathematical purposes, in actual fact the HP and LP turbines are usually co-axially mounted on the same shaft, with a single turbine housing; HP steam is introduced at the low-volume inlet of the turbine and LP steam is introduced farther downstream to provide additional power.

5.2 Modeling Double-Flash Steam Plants

The total electrical generating capacity available from a double-flash steam turbine geothermal power plant may be regarded as simply the sum of the net power generated by the high-pressure (HP) and low-pressure (LP) turbines. The generating capacity of the HP stage may be computed in exactly the same way as for a single-flash steam plant (“backpressure” or “condensing” as appropriate—see Section 4). Accordingly, the essential task in modeling a double-flash plant is to appraise the extra power available from the secondary (LP) stage.

The LP stage of a double-flash steam plant (of either design) consists, in effect, of a single-flash condensing steam plant fueled by saturated liquid water from the HP separator. Thus, the electrical generating capacity of the LP stage will, in general, depend only upon:

- (1) the flow rate of liquid from the HP separator,
- (2) the temperature of this liquid,
- (3) the CO₂ content of the liquid, and
- (4) the pressure maintained in the LP separator (equal to the LP turbine inlet pressure).

At this stage, the problem can be significantly simplified by noting that, under practical conditions, the CO₂ content of the liquid coming from the HP separator will be very low, since most of the CO₂ from the wellhead will be carried off with the HP steam. Accordingly, it suffices to assume that the inlet CO₂ content to the LP stage is equal to zero. Therefore, for a particular flow rate of HP liquid, the net electrical generating capacity of the LP stage will depend only upon the temperature of the HP liquid supply and the LP separator pressure.

The objective of this study is to establish the maximum electrical power which may be generated under particular conditions. Consequently, the next task is to find, for a particular HP liquid supply temperature, the LP separator pressure which will result in the maximum net generating capacity from the LP stage. As LP separator pressure increases, the generating capacity per unit amount of steam flow will increase, but the amount of steam available will decrease. In the limit as the LP pressure reaches the HP separator pressure, no LP steam will be produced at all. Accordingly, an “optimum LP separator pressure” can be found (for each value of HP liquid supply temperature) which will maximize the electrical generating capacity of the LP stage, and this “optimum” LP pressure will lie between the condenser pressure (~0.07 bars) and the pressure in the HP separator. The computational procedure employed to establish this relationship was (1) choose a particular value of HP separator temperature, and then (2) for a variety of values for the LP separator pressure, calculate (a) the LP separator steam rate (obtainable from the steam tables), and multiply the LP steam rate by (b) the appropriate steam specific

electrical generating capacity (from Figure 4.7; curve for single-flash condensing turbine with 0 percent CO_2). Find the largest of these values and the pressure at which it is found; these are the “optimum LP turbine net capacity” and the “optimum LP separator pressure” respectively.

Results of calculations of this type are illustrated in Figure 5.5. The upper curve shows the optimum LP stage electrical capacity (per kilogram of liquid supply from the HP separator per second) and the lower curve shows the “optimum LP separator pressure” at which this optimum electrical output is realized, both as functions of HP separator temperature. Note that for HP separator temperatures less than 163°C , the optimum LP separator pressure is sub-atmospheric. A temperature of 163°C corresponds to a partial pressure of H_2O in the HP separator of about 6.7 bars. Figure 5.6 (upper) shows the optimum LP separator pressure as a function of H_2O partial pressure in the HP separator (instead of as a function of temperature—the steam tables are used to translate temperature into H_2O partial pressure), and the lower curve expresses this optimum LP separator pressure as a fraction of the HP H_2O partial pressure. If no CO_2 is present in the HP separator, the actual total HP separator pressure will be equal to the H_2O partial pressure—if CO_2 is present, the HP separator total pressure will be greater (owing to CO_2 partial pressure). Tabular values for these optimized relationships between HP and LP stage conditions are also provided in Table 5.1.

Figure 5.5. Upper: net electrical generating capacity of low-pressure condensing steam turbine per unit flowrate of hot liquid brine from the high-pressure separator at optimum low-pressure separator pressure as a function of the temperature in the high-pressure separator. Lower: optimum low-pressure separator pressure as a function of high-pressure separator temperature.

Figure 5.6. Upper: optimum low-pressure separator pressure as a function of the partial pressure of steam in the high-pressure separator.
Lower: optimum low-pressure separator pressure expressed as a fraction of the high-pressure separator partial pressure of steam.

Table 5.1 Optimum (maximum-power) LP stage operating conditions for various HP separator conditions for double-flash plants.

HP Separator Temperature (°C)	HP Separator H ₂ O Partial Pressure (bars)	Optimum LP Separator Pressure (bars)	LP Stage Electrical Generating Capacity (kWe)
80	0.47	0.20	2.9
90	0.70	0.25	4.6
100	1.01	0.31	6.7
110	1.43	0.38	9.1
120	1.99	0.46	11.9
130	2.70	0.56	15.0
140	3.61	0.67	18.5
150	4.76	0.80	22.3
160	6.18	0.95	26.5
170	7.92	1.13	31.0
180	10.03	1.33	35.9
190	12.55	1.55	41.2
200	15.55	1.81	46.9
210	19.08	2.11	52.9
220	23.20	2.45	59.4

Compressed-liquid output flow from HP separator = one kilogram per second.

6

Binary Plants and Steam/Binary Hybrids

6.1 Binary Geothermal Plants Using Downhole Pumps and Recuperation

Binary geothermal power plants are very popular in the geothermal industry, particularly for resources of relatively low temperature. They are usually used in conjunction with production wells equipped with downhole pumps. Recently, Greg Mines of the Idaho National Engineering Laboratory undertook a computational design study for small binary geothermal power plants which would be suitable for off-grid applications (Mines, 1997). Unfortunately, this study has not so far been published. The material in this section of the present report therefore constitutes a summary of Mines' study, together with material which may readily derived from his results. Greg Mines' cooperation in the present study is gratefully acknowledged.

Figure 6.1 illustrates the simplest possible configuration for a wellhead geothermal binary power station. Hot liquid geofluid is brought to the wellhead under pressure using a downhole pump; this hot brine is immediately introduced into a wellhead heat exchanger system, through which the hot geothermal liquid flows (from left to right in Figure 6.1). As the brine passes through the heat exchanger, it gives up heat and cools; the cooled brine is then sent to disposal (usually an injection well, as shown). Simultaneously, a secondary hydrocarbon ("HC") working fluid passes through the heat exchanger in the opposite direction, gaining heat from the geothermal brine. The hydrocarbon is in a liquid state when it enters the heat exchanger (on the right in Figure 6.1), but as it passes through the heat exchanger system (from right to left), it first begins to boil, then boils off completely to vapor and may acquire a certain degree of superheat prior to departing the heat exchanger. Next, the hydrocarbon vapor passes through an axial-flow turbine (which is connected to an electrical generator—not shown in Figure 6.1). The spent hydrocarbon vapor (which may now consist of a mixture of hydrocarbon vapor and condensed hydrocarbon liquid) is discharged into a condenser which cools the turbine exhaust and condenses it entirely back into an all-liquid condition. This is accomplished using an external cooling system (not shown in Figure 6.1), as in the case of condensing steam turbines. Next, the condensed liquid hydrocarbon is pumped back up to high pressure by a hydrocarbon feed pump; the compressed liquid hydrocarbon then returns to the heat exchanger to be re-vaporized for another circuit through the system.

The performance of the above simple design can be significantly enhanced by incorporating "recuperation", as indicated in Figure 6.2 (Mines, 1996). Using recuperation, some of the waste heat from the hot hydrocarbon turbine exhaust is used to pre-heat the compressed liquid hydrocarbon from the feed pump upstream of the heat exchanger. Recuperation improves system performance in two ways: (1) by cooling and condensing some of the hydrocarbon turbine exhaust, the burden (and the associated parasitic loads) upon the cooling system is reduced, and (2) by preheating the compressed liquid hydrocarbon prior to introduction to the heat exchanger, the heat energy in the wellhead brine is more efficiently utilized. All calculations of binary plant performance presented in this report incorporate recuperation.

Figure 6.1. Schematic diagram of a simple binary-cycle geothermal power plant driven by a production well equipped with a downhole pump.

Figure 6.2. Schematic diagram of a binary-cycle power plant incorporating recuperation driven by a production well equipped with a downhole pump.

The performance of a binary-cycle geothermal power plant obviously depends upon the thermodynamic properties of the secondary hydrocarbon working fluid used in the binary circuit. In the past, chlorinated hydrocarbons (such as Freon) were used as secondary fluids for geothermal binary plants, but this is no longer practiced owing to the damage that such materials can cause to the earth's ozone layer. At present, binary plants are restricted to using ordinary hydrocarbons such as propane, isobutane, and isopentane. Compared to pure hydrocarbons, it is well-established that certain hydrocarbon mixtures can be profitably employed to improve the performance of binary-cycle power plants. Using such mixtures permits the adjustment and optimization of the thermodynamic properties of the secondary working fluid. Generally speaking, using optimized hydrocarbon mixtures will improve the performance (in terms of electrical generation capacity for a particular hot brine supply, relative to using pure hydrocarbons) by around 30 percent at low brine inlet temperatures ($\sim 100^\circ\text{C}$); the advantage declines with increasing heat exchanger brine inlet temperature (to ~ 5 percent or so at 200°C brine temperature).

For the present study, however, attention was restricted to binary plants using pure hydrocarbons as the secondary working fluid. The reason for this restriction (Mines, 1996) has to do with the remote-site off-grid applications of particular interest. All "closed" systems leak, and over months of time leakage from the secondary HC circuit in a geothermal binary power station can be significant. If a mixture of hydrocarbons is used, the lighter molecular weight components will tend to leak away faster than the heavier components, resulting in a progressive change with time of the composition of the hydrocarbon mixture. To correct this situation, it will be necessary that the local operators have (1) a supply of each of the various components available, (2) the capability to perform the necessary chemical analyses to determine the appropriate mixture required to recharge the system and to return the hydrocarbon composition to design conditions, and (3) the local capability to create such mixtures and maintain the hydrocarbon loop composition. In view of the remote rural electrification applications envisioned for this technology, the decision was made to restrict attention to pure substances for the hydrocarbon circuit, and to accept the performance penalty involved.

6.2 Performance of Single-Hydrocarbon Off-Grid Binary Plants

Mines (1997) considered six different candidate secondary working fluids: propane, isobutane, n-butane, isopentane, n-pentane, and hexane. He also considered nine different values for the inlet temperature of the liquid water to the heat exchanger: 80°C , 100°C , 120°C , 140°C , 160°C , 180°C , 200°C , 220°C and 240°C . For each of the resulting 54 combinations of working fluid and brine inlet temperature, he numerically optimized the design parameters of a recuperating binary plant (illustrated in Figure 6.2) to obtain the maximum net exportable electrical generating capacity (subject to all pertinent parasitic loads except for the downhole pump itself). An important constraint on the design process was the requirement that the brine outlet temperature from the heat exchanger must be no lower than:

$$T_{\text{OUT}_{\min}} = 0.965T_{\text{IN}} - 117 \quad (6.1)$$

where T_{IN} is heat exchanger brine inlet temperature, and both temperatures are expressed in degrees Celsius. The purpose of this requirement is to avoid silica scaling in the heat exchanger. Relaxing this

requirement would yield higher electrical generating capacities for high inlet temperatures, but would create plant maintenance problems which would be unacceptable under remote-operation conditions.

Figure 6.3 shows the optimum electrical generating capacity which Mines computed for each of the 54 cases, connected using smooth curves for temperature-interpolation purposes. Figure 6.4 shows how the optimum heat exchanger brine outlet temperature varies with inlet temperature for the six hydrocarbons considered—note the effect of restriction 6.1 (above) for higher brine inlet temperatures. As can be seen in Figure 6.3, there is no single hydrocarbon working fluid that provides the best performance over the entire range. For the system depicted in Figure 6.2, Mines' calculations suggest that hydrocarbons be selected as follows:

Heat Exchanger Brine Inlet Temperature (°C)	Hydrocarbon Working Fluid
80 - 109	Hexane
109 - 164	Propane
164 - 197	Isobutane
197 - 240	N-Butane

The electrical generating capacity of the binary plant (per kilogram per second of hot liquid brine entering the heat exchanger) is the upper envelope of the various curves shown in Figure 6.3.

6.3 Self-Discharging Wells—Binary Plants With Cold-Water Recirculation

Binary plants are ordinarily driven by wells equipped with downhole pumps. Introducing the two-phase mixture of water and steam which emerges from a self-discharging wellhead into the heat exchanger is impractical because of scaling and corrosion problems. For larger-scale plants, more complicated binary designs have been developed with multiple heat exchangers (separate heat exchangers for separated water and steam), but these more elaborate designs are not suitable for the small-plant remote-site applications envisioned here owing to excessive capital and maintenance costs. The essential problem is that, to be successful, any rural electrification program involving small geothermal power plants will eventually require the mass-production of small modular geothermal power plants. If such a small binary plant is available but if, for some reason, a downhole pump is not available or if the well characteristics are such as to preclude downhole pump operation, what is the simplest way to drive the power plant using self-discharge with a wellhead separator?

The approach adopted in the present study is illustrated in Figure 6.5. Here, a self-discharging well is equipped with a wellhead separator and the separated liquid is directed to the heat exchanger inlet for the recuperating binary plant. After passing through the heat exchanger, a fraction of the cooled liquid

Figure 6.3. Optimized net electrical generating capacity (downhole pump parasitic load not considered) per unit hot brine flow rate into the heat exchanger for recuperating binary power plants for various secondary loop working fluids (after Mines, 1997).

Figure 6.4. Optimized brine discharge temperature from the heat exchanger for optimized recuperating binary power plants for various secondary loop working fluids (after Mines, 1997). Heat exchanger outlet temperature constrained to avoid silica precipitation in heat exchanger, as shown.

Figure 6.5. Schematic diagram of cold water recirculation system for recuperating binary power plant driven by self-discharging well.

is redirected, through a recirculation pump, back to a “reheater”; a vessel in which the steam from the separator is bubbled through the cooled exhaust brine, condensing the steam and heating the brine back up to heat-exchanger inlet temperature. As a result, a particular element of produced water may circulate through the heat exchanger several times before finally being reinjected, multiplying the effectiveness of the plant.

As an example, consider a production well which will self-discharge 50 kilograms per second of CO₂-free fluid at 5 bars wellhead pressure with a wellhead discharge enthalpy of 1000 Joules per gram. At five bars pressure, the separator temperature will be 152°C and the outflows from the separator will be 8.53 kg/s of steam and 41.47 kg/s of hot water (50 kg/s total). Using a propane working fluid in the binary plant with a water inlet temperature of 152°C, the cooled water will leave the heat exchanger at 51°C (green line in Figure 6.4). If 42.13 kg/s of this cooled water is combined with the 8.53 kg/s of saturated 5-bar steam from the separator, it will produce 50.66 kg/s of hot liquid at 152°C. Combining this 50.66 kg/s of hot water from the reheater with the original 41.47 kg/s of 152°C water flowing from the separator yields a total of 92.13 kg/s of hot water entering the binary plant’s heat exchanger at 152°C—sufficient to generate 3031 kilowatts of electricity (from Figure 6.3). The flow rate down the injection well is, of course, 50 kg/s. Thus, in this case, the water mass flow rate through the heat exchanger is almost double the total wellhead fluid production rate and is 222 percent of the hot liquid flow rate from the separator alone.

The parasitic load required to operate the recirculation pump must be deducted from the generating capacity. For this purpose, it was assumed for all calculations reported herein that the overall (electrical power to hydraulic power) efficiency of the recirculation pump is 65 percent and that the pumping head required to overcome the frictional pressure losses in the heat exchanger is 1.6 bars. This is significantly higher than values reported by plant manufacturers, but the higher value is employed to represent the probable effects of scaling in the heat exchanger blocking fluid passageways over time. Based upon these figures, the parasitic load for the recirculation pump in the above example will be about 6.8 kilowatts, reducing the net exportable power from 3031 kWe to 3024 kWe. Since the well is self-discharging, no downhole-pump deduction is needed.

6.4 Flash Steam/Binary Cycle Hybrid Plants

An alternative approach to using binary-cycle technology for self-discharging wells is to employ a steam/binary hybrid plant design in which two turbines are used—a steam turbine using steam from the wellhead separator, and a hydrocarbon turbine driven by hydrocarbon vapor created in the heat exchanger for which the energy is provided by cooling the separated liquid water. The steam turbine may be either of the backpressure type (see Figure 6.6) or the condensing type (Figure 6.7). It is even possible to operate binary plants of this type using wells with downhole pumps, by expanding the produced wellhead liquid into a wellhead separator as usual (Figures 6.8, 6.9), although, as will be shown, the backpressure steam / binary hybrid arrangement (Figure 6.8) is outperformed by the simpler binary plant (Figure 6.2) under virtually all conditions considered in this study. The combination of pumped wells with the condensing steam / binary hybrid design is, however, capable of outperforming the (pumped well/binary plant) combination (Figure 6.2) if the reservoir temperature is high enough (over 210°C or so).

Figure 6.6. Schematic diagram of a backpressure steam / binary hybrid wellhead generator driven by a self-discharging geothermal well.

Figure 6.7. Schematic diagram of a condensing steam / binary hybrid wellhead generator driven by a self-discharging geothermal well.

Figure 6.8. Schematic diagram of a backpressure steam / binary hybrid wellhead generator driven by a geothermal well equipped with a downhole pump.

Figure 6.9. Schematic diagram of a condensing steam / binary hybrid wellhead generator driven by a geothermal well equipped with a downhole pump.

Calculation of the plant performance for any of these steam/binary hybrid arrangements is straightforward. The electrical power available from the steam turbine stage is computed exactly as in Section 4 (for the corresponding single-flash steam plant—backpressure or condensing, as appropriate). The binary stage electrical output is computed as described above in Section 6.2, with the heat exchanger water inlet temperature equal to the separator temperature.

7

Optimizing System Performance

Sections 2 and 3 discussed how calculations of the fluid delivery capacity of geothermal wells were carried out (both with and without downhole pumps). Sections 4, 5 and 6 describe the mathematical models developed for the performance of a variety of candidate wellhead geothermal generating plants. The next task is to combine these descriptions and adjust the available free parameters to find optimum (*i.e.*, maximum electrical generating capacity) values for each combination of conditions. The approach to this optimization process is somewhat different for pumped wells as compared to self-discharging wells, as will be seen.

7.1 Wells With Downhole Pumps and Binary Plants

The simplest case to consider is that of a binary plant located at the wellhead of a well equipped with a downhole pump (either of the lineshaft or submersible design; see Figure 6.2, preceding section). In this case, there is really nothing left to optimize. The downhole pump delivers fluid at a particular rate and temperature (see Section 3, Figures 3.10–3.29; also see Appendices C, D, G and H) to the wellhead in a compressed liquid condition. The electrical generating capacity is then established by (1) using Figure 6.3 (preceding section) to select the best hydrocarbon working fluid and calculate the electrical generating capacity, and then (2) deducting the parasitic load required for the downhole pump.

7.2 Wells With Downhole Pumps and Wellhead Separators

All other combinations of wellhead power plants and wells with downhole pumps involve a wellhead separator, which operates at a pressure less than the bubble-point pressure for the wellhead conditions provided by the pump. These include:

- single-flash backpressure steam turbine (see Fig. 4.3),
- single-flash condensing steam turbine (see Fig. 4.4),
- double-flash backpressure/condensing steam turbine (see Fig. 5.3),
- double-flash condensing steam turbine (see Fig. 5.4),
- backpressure steam/binary hybrid (see Fig. 6.8), and
- condensing steam/binary hybrid (see Fig. 6.9).

The performance of the plant will depend upon the operating pressure maintained in the separator. Changing the separator pressure does not change the total fluid flow rate available nor the total flowing enthalpy or composition (CO_2 fraction), but it does change the separator temperature and the temperature, enthalpy, composition, and relative amounts of the water and steam which emerge from the separator and are used to drive various power plant components.

Accordingly, numerical calculations were undertaken, for each of the above plant designs, to find (for each value of wellhead liquid temperature and CO_2 content) the value of separator pressure which

yields the system with the greatest electrical generating capacity per unit mass of wellhead fluid delivered by the pump per second, prior to deduction for the parasitic load of the pump. Since the pump parasitic load is independent of separator pressure, it will influence the net exportable electrical capacity but will not influence the value of the optimum separator pressure or other pertinent design parameters for the plant.

Figure 7.1 shows, for single-flash backpressure steam turbines connected to wells equipped with downhole pumps, the optimum value (for maximum electrical generating capacity) of the separator pressure (equal to the turbine inlet pressure) as a function of both the temperature and the dissolved CO_2 mass fraction in the compressed liquid provided at the wellhead by the downhole pump. Figure 7.2 shows the optimized electrical generating capacity thereby attained (prior to deduction for downhole pump parasitic load) per unit wellhead liquid supply rate. In both plots, the small triangular shaded area near the origin represents the region in which no boiling will occur at one atmosphere separator pressure (the minimum value; equal to the turbine exhaust pressure for a backpressure turbine). Generally, as wellhead liquid temperature increases, both the optimum separator pressure and the electrical generating capacity also increase. Both the optimum pressure and the generating capacity also increase slightly with increasing CO_2 content over the range considered. Generating capacity is very small for supply temperatures less than 150°C or so, and will usually be insufficient even to meet the downhole pump parasitic load. At low temperatures (below 115°C – 120°C or so), the optimum separator pressure begins to increase with decreasing supply temperature. In this region, the amount of water vapor which will evolve in the separator is negligible; the optimization process is trying to create a “backpressure CO_2 gas turbine”. The electrical output is, of course, negligible.

Figures 7.3 and 7.4 illustrate corresponding optimized data for single-flash condensing steam turbines. In this case, the optimum operating pressure (Figure 7.3) is significantly lower than in the backpressure case (Figure 7.1); for liquid supply temperatures less than 160°C or so (depending slightly on CO_2 content), optimum performance will be attained using sub-atmospheric turbines. Both optimum operating pressure and specific generating capacity increase with increasing supply temperature, and it is noteworthy that the electrical generating capacity (Figure 7.4) is significantly greater than for the backpressure case (Figure 7.2), particularly at low temperatures. Also note that, in contrast to the backpressure case, increasing the CO_2 content reduces the electrical generating capacity slightly due to both increased parasitic loads for the NCG pump and higher condenser pressures.

The “backpressure/condensing” double-flash case with a downhole pump is illustrated in Figures 7.5–7.9. Figures 7.5 and 7.6 show the optimum pressures to be maintained for maximum power production in the HP and LP separators respectively as functions of supply liquid temperature and CO_2 content. Optimum HP separator pressures are very high whereas optimum LP separator pressures are sub-atmospheric for temperatures less than 162°C or so. The shaded area indicates the region in which performance of this double-flash plant design will be inferior to that of a single-flash condensing turbine. Figure 7.7 shows the maximum electrical power which can be generated for this plant/well configuration as a function of supply liquid temperature and CO_2 content. Most of the electrical generating capacity comes from the low-pressure condensing stage, as shown in Figure 7.8. In fact, the electrical generating capacity attained by using the “backpressure/binary” double-flash design is only slightly greater than that obtainable from a much simpler single-flash condensing steam plant, as Figure 7.9

Figure 7.1. Optimum separator pressure for single-flash wellhead backpressure steam turbines supplied by wells equipped with downhole pumps.

Figure 7.2. Maximum attainable electrical generating capacity for single-flash wellhead backpressure steam turbines supplied by wells equipped with downhole pumps.

Figure 7.3. Optimum separator pressure for single-flash wellhead condensing steam turbines supplied by wells equipped with downhole pumps.

Figure 7.4. Maximum attainable electrical generating capacity for single-flash wellhead condensing steam turbines supplied by wells equipped with downhole pumps.

Figure 7.5. Optimum HP separator pressure for double-flash wellhead backpressure/condensing steam turbines supplied by wells equipped with downhole pumps.

Figure 7.6. Optimum LP separator pressure for double-flash wellhead backpressure/condensing steam turbines supplied by wells equipped with downhole pumps.

SPECIFIC GROSS PLANT OUTPUT (KWe PER KILOGRAM LIQUID/SECOND)

NOTE: DOWNHOLE PUMP PARASITIC LOAD NOT YET DEDUCTED

PLANT TYPE: DOUBLE-FLASH BACKPRESSURE/CONDENSING STEAM TURBINE

SHADED AREA: INFERIOR TO SINGLE-FLASH CONDENSING STEAM TURBINE

Figure 7.7. Maximum attainable electrical generating capacity for double-flash wellhead backpressure/condensing steam turbines supplied by wells equipped with downhole pumps.

Figure 7.8. Optimum fraction of total electrical generating capacity from LP stage for double-flash wellhead backpressure/condensing steam turbines supplied by wells equipped with downhole pumps.

Figure 7.9. Amount by which optimized backpressure/condensing double-flash plants outperform optimized single-flash condensing steam plants, both supplied by wells equipped with downhole pumps.

shows (Figure 7.9 is, in effect, Figure 7.7 minus Figure 7.4). There is a very slight advantage to this plant design for high- CO_2 systems, but the tiny advantage represented by Figure 7.9 certainly does not justify the much greater maintenance costs or the capital costs of building an extra steam separator and separate backpressure turbine unit, particularly in view of the environmental issues posed by the atmospheric exhaust of the backpressure HP stage. Consequently, the “backpressure/condensing” double-flash design was dropped from further consideration for applications involving wells with downhole pumps.

The situation is significantly better for condensing double-flash turbines, as shown in Figures 7.10–7.14. Figure 7.10 shows how the optimum HP separator pressure varies with the temperature and CO_2 content of the wellhead supply liquid from the downhole pump. The HP separator pressure is sub-atmospheric for wellhead fluid temperatures below 125°C (1 percent CO_2) or 130°C (no CO_2) and reaches eight bars at $\sim 240^\circ\text{C}$. The LP separator is sub-atmospheric except for supply temperatures above 227°C – 228°C (Figure 7.11). The electrical generating capacity is substantial even at relatively low temperatures (Figure 7.12); for low CO_2 contents around 30 percent of the generating capacity arises from the low-pressure stage (Figure 7.13) but this percentage can be 50 percent or more under conditions of low temperature and high CO_2 content. As Figure 7.14 shows, the extent by which the double-flash condensing steam plant outperforms the corresponding single-flash plant (Figure 7.4) is significant, ranging from 25 percent to 30 percent more power over most of the range. Whether or not this additional performance justifies the additional plant capital and maintenance costs is not clear.

It is reasonably clear, however, that the backpressure steam / binary design has very little to recommend it, at least for operation with wells equipped with downhole pumps over the range of parameter values considered in this study. As shown in Figures 7.15–7.19, the system has no better performance than a simple binary plant (Section 7.1, above) except in a very narrow region at high temperatures ($> 238^\circ\text{C}$ for CO_2 -free fluids; $> 227^\circ\text{C}$ for 1 percent CO_2 in the wellhead liquid). Figure 7.18 shows that most of the power from the optimized hybrid plant will come from the hydrocarbon turbine in any event (very little comes from the backpressure steam turbine), and at best the capacity advantage over the range considered is less than 1 percent—over most of the range there is no advantage whatever. Clearly, the additional costs relative to a much simpler all-binary plant are not warranted. Therefore, the backpressure / binary hybrid design (like the backpressure/condensing double-flash plant) was dropped from further consideration for wells equipped with downhole pumps.

Finally, consider the condensing steam / binary hybrid concept, for which the results of the optimization calculations are shown in Figures 7.20–7.24. This scheme will only be useful for high temperature supply liquids ($> 210^\circ\text{C}$ or so). The optimum separator pressures are quite high (17 to 21 bars; see Figure 7.20) so that high heat exchanger inlet temperatures can be attained (Figure 7.21). At very high temperatures (240°C) the electrical generating capacity is the highest of all the plants considered, and represents a 7 percent improvement over a simple binary plant (Section 7.1). At 240°C wellhead temperature, one-third of the generating capacity of the optimized plant comes from the steam turbine.

The maximum optimized specific generating capacities of the five remaining plant designs (excluding the backpressure/condensing double-flash plant and the backpressure steam/binary hybrid) as functions of the temperature of the liquid delivered to the wellhead by the downhole pump (for zero dissolved CO_2) are

Figure 7.10. Optimum HP separator pressure for double-flash wellhead condensing steam turbines supplied by wells equipped with downhole pumps.

Figure 7.11. Optimum LP separator pressure for double-flash wellhead condensing steam turbines supplied by wells equipped with downhole pumps.

Figure 7.12. Maximum attainable electrical generating capacity for double-flash wellhead condensing steam turbines supplied by wells equipped with downhole pumps.

Figure 7.13. Optimum fraction of total electrical generating capacity from LP stage for double-flash wellhead condensing steam turbines supplied by wells equipped with downhole pumps.

Figure 7.14. Amount by which optimized condensing double-flash plants outperform optimized single-flash condensing steam plants, both supplied by wells equipped with downhole pumps.

Figure 7.15. Optimum separator pressure for backpressure steam / binary hybrid plants supplied by wells equipped with downhole pumps.

Figure 7.16. Heat exchanger inlet temperature at optimum conditions for backpressure steam / binary hybrid plants supplied by wells equipped with downhole pumps.

Figure 7.17. Maximum attainable electrical generating capacity for backpressure steam / binary hybrid plants supplied by wells equipped with downhole pumps.

Figure 7.18. Optimum fraction of total electrical generating capacity from binary stage for backpressure steam / binary hybrid plants supplied by wells equipped with downhole pumps.

Figure 7.19. Amount by which optimized backpressure steam / binary hybrid plants outperform ordinary recuperating binary plants, both supplied by wells equipped with downhole pumps.

Figure 7.20. Optimum separator pressure for condensing steam / binary hybrid plants supplied by wells equipped with downhole pumps.

Figure 7.21. Heat exchanger inlet temperature at optimum conditions for condensing steam / binary hybrid plants supplied by wells equipped with downhole pumps.

Figure 7.22. Maximum attainable electrical generating capacity for condensing steam / binary hybrid plants supplied by wells equipped with downhole pumps.

Figure 7.23. Optimum fraction of total electrical generating capacity from binary stage for condensing steam / binary hybrid plants supplied by wells equipped with downhole pumps.

Figure 7.24. Amount by which optimized condensing steam / binary hybrid plants outperform ordinary recuperating binary plants, both supplied by wells equipped with downhole pumps.

illustrated in Figure 7.25. Figure 7.26 is similar, but for a high CO_2 content (1 percent by mass) in the hot liquid supply. Clearly, the single-flash backpressure steam turbine approach does not make as efficient use of the hot water supply as the others, but backpressure steam plants are much simpler and cheaper than the other plant designs. The remaining four approaches (single- and double-flash condensing turbines, simple binary plant, and condensing steam/binary hybrid) lie within a factor of two of each other in performance. At high temperatures (240°C), the double-flash condensing steam plant outperforms the single-flash condensing plant by about 25 percent; the binary plant outperforms the double-flash plant by 18 percent and the condensing steam/binary hybrid offers an additional 8 percent performance advantage. At low temperatures (80°C) and low CO_2 contents (Figure 7.25), on the other hand, the hybrid plant is impractical (Figures 7.20–7.24); according to these calculations, both the single-flash and double-flash plants will outperform the binary plant. One reason for this rather surprising result is the decision to restrict attention to single-component hydrocarbons for the binary working fluid. As noted earlier, use of optimized hydrocarbon mixtures can improve low-temperature performance of binary-cycle plants significantly. For higher CO_2 contents (Figure 7.26) the performance of both condensing plant designs at low temperatures suffer considerably. High CO_2 levels actually improve the performance of the backpressure steam turbine by increasing attainable turbine inlet pressures for a particular supply fluid temperature.

7.3 Self-Discharging Wells

Optimizing the performance of power plants driven by self-discharging wells is considerably more complicated. If the well is pumped, the rate at which hot fluid is brought to the wellhead and the heat content (enthalpy) of that fluid are fixed. The separator pressure is adjusted to obtain the best results. In the case of a self-discharging well, however, the pressure maintained in the wellhead separator (equal to the wellhead pressure) influences both the total fluid flow rate and the discharge enthalpy available from the well. Thus, it is not possible to separately optimize plant performance by adjusting separator pressure as a function of wellhead temperature and CO_2 content as in the case of pumped wells (Figures 7.1–7.24). Instead, the entire system (well plus power plant) must be optimized simultaneously.

To illustrate, consider the relatively simple situation depicted in Figure 7.27. Figures 7.27A and 7.27B show the wellhead discharge characteristics for a 150 mm ID self-discharging slim hole with feedpoint at 600 m depth, 4 kg/s/bar productivity index, no reservoir CO_2 , piezometric surface depth = 100 m and 200°C reservoir temperature (see Section 2, Figure 2.4). For each value of the fluid discharge rate, the corresponding steam outflow rate from the separator can be computed from the total flow rate, the pressure, and the discharge enthalpy using the steam tables. This steam outflow rate may then be expressed as a function of wellhead pressure (= separator pressure) instead of as a function of total flow rate, using the relationship in Figure 7.27A to change variable. The results (separated steam rate as a function of pressure) are shown in Figure 7.27C. Note that the function is double-valued because of the maximum value in Figure 7.27A; only the higher-rate solution is of interest.

Next, we may consult Figure 4.7 (and use the curves corresponding to zero CO_2 content); these curves which express net plant generating capacity per unit steam rate as functions of supply pressure are also reproduced as Figure 7.27D for convenience. If we simply multiply the steam supply function (Figure 7.27C) times the plant performance functions (Figure 7.27D), we obtain the net electrical capacity (for

Figure 7.25. Maximum generating capacities as a function of wellhead temperature for wells equipped with downhole pumps using various wellhead power plant designs. Reservoir CO₂ content = zero.

Figure 7.26. Maximum generating capacities as a function of wellhead temperature for wells equipped with downhole pumps using various wellhead power plant designs. Reservoir CO₂ content = 1% by mass in liquid.

Figure 7.27. Example of performance optimization for single-flash steam turbine plants driven by self-discharging well.

either type of single-flash steam plant) for this slim hole as a function of operating pressure (wellhead pressure = separator pressure = turbine inlet pressure). These functions are shown in Figure 7.27E. Both generating capacity curves attain maximum values at particular values of the operating pressure. If the pressure is too high, output falls due to declining steam availability. If the pressure is too low, the generating capacity declines due to loss of pressure difference between turbine inlet and outlet.

The “optimum” operating conditions for the condensing turbine occur at lower pressures than for the backpressure plant, and the condensing plant is obviously capable of generating much more electricity from this well. For this particular case, the optimum operating pressure for the backpressure plant is 3.61 bars, at which pressure the generating capacity reaches its maximum value of 427 kWe. For the condensing plant, the optimum operating pressure is lower (1.75 bars) and the electrical generating capacity at that pressure is 1245 kWe.

The procedure is similar if the CO₂ content of the reservoir fluid is non-zero—the partitioning of the CO₂ between the liquid and the vapor phases must be taken into account, since the specific generating capacity (similar to Figure 7.27D) depends on the CO₂ fraction in the steam. The more complicated plant designs (recirculating binary plant, double-flash plants and flash/binary hybrid plants) produce power from both the steam from the separator and the separated liquid water, so that the optimization process involves more calculations, but the principle is the same. The computed wellhead discharge characteristics (for particular values of well diameter, feedpoint depth, reservoir pressure, reservoir permeability, reservoir CO₂ content and reservoir temperature) are used to establish the outflow rates, compositions, pressures and temperatures of the water and steam from the separator at various values of the separator pressure. For each pressure value, the total net electrical generating capacity is then computed; the values associated with the separator pressure value which produces the greatest amount of generating capacity are stored for future use. These optimum values (for various values of the free parameters) are presented graphically in Appendices A and B of this report, and will also be discussed in Section 8.

Here, attention is confined to a particular “base case” and the effects of perturbing the base case by varying individual key parameters to indicate the effects of these parameters upon the optimized operating conditions for each plant design concept when used with self-discharging wells. As before, the “base case” is:

Well inside diameter:	150 mm
Piezometric surface depth:	100 m
Well productivity index:	4 kg/s/bar
Well feedpoint depth:	600 m
Reservoir CO ₂ content:	None

Figure 7.28 shows the optimum separator pressure (= wellhead pressure = turbine inlet pressure) for maximum-electrical-capacity backpressure steam turbine plant designs as a function of reservoir temperature for the base case, and for other cases involving variations in each of the above five key parameters. As noted earlier, optimum separator pressure increases sharply with increasing reservoir temperature. Although reservoir temperature is the most important parameter influencing the optimum

Figure 7.28. Influence of key parameters on optimum separator pressure for single-flash backpressure steam turbine plant supplied by self-discharging well.

separator pressure, there are tendencies for separator pressure to increase with (1) increasing well diameter, (2) decreasing piezometric surface depth (or increasing reservoir pressure), (3) increasing productivity index, (4) decreasing feedpoint depth, and (5) increasing CO₂ content. Over the ranges considered, reservoir pressure and CO₂ content appear to be the most important.

Figure 7.29 is similar, but for condensing single-flash steam turbine designs. Optimum operating pressures are much lower than for backpressure turbines, but are likewise very sensitive to reservoir temperature. Qualitatively the same trends are present in the influences of the other five key parameters, but in much-attenuated form. In fact, the optimum operating pressure for a single-flash condensing steam turbine is remarkably insensitive to all of the other parameters, and depends for practical purposes on reservoir temperature alone.

Figures 7.30, 7.31 and 7.32 describe optimization results for double-flash backpressure/condensing turbine designs, and show optimum HP separator pressure, optimum LP separator pressure, and optimum LP steam generating capacity (as a percentage of the total) respectively. As before, the most important parameter is the reservoir temperature. Also, as in the preceding cases, both the optimum HP and LP separator pressures increase with (1) increasing well diameter, (2) decreasing piezometric surface depth, (3) increasing productivity index, (4) decreasing feedpoint depth, and (5) increasing CO₂ content. Except at temperatures above 220°C or so, optimum LP separator pressures are sub-atmospheric. At high temperatures, the optimum designs generate about half the gross electricity from the LP condensing stage; as reservoir temperature decreases, this fraction increases and reaches 100 percent at a reservoir temperature which is usually in the 140°C–160°C range which depends mainly on reservoir pressure and CO₂ content. Below this limiting temperature, the design is impractical.

The corresponding situation for the condensing double-flash steam plant designs is illustrated in Figures 7.33–7.35. Trends in HP and LP separator pressures and their sensitivities to various parameters are the same as in the prior cases, but pressures are significantly lower than for the backpressure/condensing double-flash case. Also, optimum double-flash condensing plant designs typically generate only about 20 percent of the power from the LP stage—this parameter is most sensitive to reservoir pressure and CO₂ content.

Optimum separator pressures for the recirculating binary plant concept as functions of reservoir temperature are shown in Figure 7.36. Generally, trends are similar to the preceding cases. Optimum separator pressures tend to be quite high. Figures 7.37 and 7.38 (for the backpressure steam / binary hybrid design) and 7.39 and 7.40 (for the condensing steam / binary hybrid) likewise display similar trends and sensitivities to parameter variation. At optimum design conditions, at high temperatures the backpressure steam / binary hybrid will generate about two-thirds of the power from the binary stage; the corresponding percentage for the condensing hybrid system tends to increase from about 20 percent at intermediate reservoir temperatures (~170°C or so) to around 40 percent at 240°C. Both are sensitive to reservoir pressure and CO₂ content as well as to reservoir temperature.

As a practical matter, of course, it will not be cost-effective to custom-build a geothermal power station for each small wellhead off-grid project. The optimized results (for, for instance, optimum operating

Figure 7.29. Influence of key parameters on optimum separator pressure for single-flash condensing steam turbine plant supplied by self-discharging well.

Figure 7.30. Influence of key parameters on optimum HP separator pressure for double-flash backpressure/condensing steam turbine plant supplied by self-discharging well.

Figure 7.31. Influence of key parameters on optimum LP separator pressure for double-flash backpressure/condensing steam turbine plant supplied by self-discharging well.

Figure 7.32. Influence of key parameters on optimum LP generating capacity fraction for double-flash backpressure/condensing steam turbine plant supplied by self-discharging well.

Figure 7.33. Influence of key parameters on optimum HP separator pressure for double-flash condensing steam turbine plant supplied by self-discharging well.

Figure 7.34. Influence of key parameters on optimum LP separator pressure for double-flash condensing steam turbine plant supplied by self-discharging well.

Figure 7.35. Influence of key parameters on optimum LP generating capacity fraction for double-flash condensing steam turbine plant supplied by self-discharging well.

Figure 7.36. Influence of key parameters on optimum separator pressure for binary plant with cold-water recirculation system supplied by self-discharging well.

Figure 7.37. Influence of key parameters on optimum separator pressure for backpressure steam / binary hybrid plant supplied by self-discharging well.

Figure 7.38. Influence of key parameters on optimum binary stage generating capacity fraction for backpressure steam / binary hybrid plant supplied by self-discharging well.

Figure 7.39. Influence of key parameters on optimum separator pressure for condensing steam / binary hybrid plant supplied by self-discharging well.

Figure 7.40. Influence of key parameters on optimum binary stage generating capacity fraction for condensing steam / binary hybrid plant supplied by self-discharging well.

pressure and electrical capacity for a single-flash steam turbine as functions of well and reservoir parameters) are intended as a guide to selection of proper equipment to get the best results. Thus, the degree to which the ability to obtain useful amounts of power depends upon the extent to which the plant conforms to the "optimum" plant description is clearly pertinent. Fortunately, it appears that the performance maxima tend to be reasonably "broad". Figure 7.41 illustrates some results for our "base case" (no CO_2 , 150 mm ID, 600 m feedpoint depth, 100 m piezometric surface depth, 4 kg/s/bar productivity index) as functions of reservoir temperature for a self-discharging slim hole connected to a wellhead single-flash condensing steam turbine. The solid curve indicates the electrical generating capacity that will be attained at each value of the reservoir temperature if the exact predicted optimum operating pressure is employed in the system design. The various colored symbols show the electrical capacities if, instead, systems operating at certain fixed pressures are used. For example, it is reasonably clear that using a separator/turbine system designed to use 8-bar steam would seriously underutilize the potential of this well. Significant improvement can be attained using a 4-bar system so long as the reservoir temperature exceeds 190°C or so (although this pressure is still too high for the entire range—the "optimum" operating pressure at 240°C is 3.1 bars, and is lower for lower reservoir temperatures). A 2-bar system is actually optimum if the reservoir temperature is 210°C, but as Figure 7.41 clearly shows, a 2-bar system will provide nearly optimum results for a broad range of reservoir temperatures, from about 180°C to above 240°C. A one-bar system (optimum for 166°C reservoir temperature) can be used with little performance degradation for reservoir temperatures between 145°C and 220°C.

Figure 7.41. Example of the effects of system operation at non-optimum conditions.

8

Electrical Capacities of Geothermal Wells With Wellhead Generators

8.1 Contents of the Appendices

The Appendices to this report provide detailed results (in graphical form) of the optimized electrical generating capacities and associated conditions under which the optimum capacity is realized for a broad range of conditions. Appendix A provides results for the various combinations of the free parameters for individual self-discharging wells producing from reservoirs without CO₂ ("Series 1"). Appendix B provides results from the self-discharging "Series 2" calculations which explore the effects of reservoir CO₂. Appendices C and D report the performance of lineshaft pumps for the "Series 1" and "Series 2" calculations (as discussed in Section 3.6). Appendix E presents "Series 1" results for wells equipped with lineshaft pumps, and Appendix F provides the corresponding lineshaft-pump "Series 2" results. Appendices G and H provide the performance results for submersible pumps discussed in Section 3.6. "Series 1" results for wells equipped with submersible pumps are to be found in Appendix I, and the "Series 2" results are in Appendix J. A summary of the calculations reported in the various Appendices is provided in Table 8.1.

The interested reader may consult these Appendices to find detailed information concerning the computed results obtained in this project. In this section, we examine the general level of electrical generating capacity that can be obtained from single geothermal wells of various sizes using different types of wellhead generating equipment, both with and without downhole pumps installed in the wells, and examine the influence of certain key parameters upon these results.

8.2 Influence of Well Diameter on Electrical Capacity

First, consider the effect of borehole inside diameter upon the optimized maximum electrical generating capacity using each of the seven different wellhead generator design concepts considered in this study. Figures 8.1 and 8.2 show (as black curves) the maximum optimized electrical generating capacity for all seven wellhead plant designs as functions of reservoir temperature for self-discharging wells with inside diameters of 75 mm, 150 mm and 300 mm. Also shown (as grey curves) are the generating capacities for wells of 150 mm ID in which submersible pumps have been installed, and for large production wells (300 mm ID) equipped with lineshaft downhole pumps. Note that the electrical capacities reported here in Section 8 are all "net exportable" power—all parasitic loads, including those to supply downhole pumps, have been deducted. Other conditions correspond to the usual "base case" (no reservoir CO₂, 600 m feedpoint depth, 100 m piezometric surface depth, and 4 kg/s/bar productivity index).

Generally speaking, electricity production is impractical for reservoir temperatures below 150°C or so unless a downhole pump is employed. Using a downhole pump, useful quantities of electricity can be generated for the larger well diameters (in which the pump will fit) for reservoir temperatures down to

the 100°C–120°C range using almost any of the plant designs except the single-flash backpressure turbine (both the backpressure/condensing double-flash turbine and the backpressure/binary hybrid are also impractical at low temperatures, as noted earlier).

The smallest well diameter considered (75 mm) is too small to accommodate a downhole pump. Even these tiny wells are capable of producing electrical power in the useful range, however. For the seven various wellhead power plant design concepts considered, a 75 mm ID self-discharging well can produce at least 100 kWe of electricity so long as the reservoir temperature exceeds the following values, according to Figures 8.1 and 8.2:

Single-flash backpressure steam turbine:	209°C
Single-flash condensing steam turbine:	171°C
Double-flash backpressure/condensing steam turbine:	186°C
Double-flash condensing steam turbine:	166°C
Binary plant with cold-water recirculation:	179°C
Backpressure steam / binary hybrid:	167°C
Condensing steam / binary hybrid:	166°C

8.3 Effect of Power Plant Design on Electrical Capacity

As has already been shown, the various plant design concepts selected for examination in this study vary in their electrical production capacity (they also vary considerably in probable capital and maintenance costs). Considering electrical capacity alone, the selection of the “best” design concept will depend upon reservoir temperature and other reservoir parameters as well as upon the diameter of the well and the availability (or lack thereof) of a downhole pump.

Figure 8.3 shows, for a large 300 mm ID self-discharging production size well, how the electrical generating capacity varies with reservoir temperature for each plant design (other conditions are, as usual: piezometric surface depth = 100 m, feedpoint depth = 600 m, productivity index = 4 kg/s/bar, no CO₂). In all cases, the capacity reaches a maximum in the 210°C–220°C range and then declines with increasing temperature due to two-phase behavior within the formation at the feedpoint depth. Under these conditions, the various designs involving backpressure steam turbines (shown in blue) make the least effective use of the well, with lowest capacity from the simple single-flash backpressure turbine and the highest from the backpressure steam / binary hybrid. The binary plant with the cold-water recirculation system (in red) outperforms the backpressure steam / binary hybrid plant throughout the temperature range. A simple single-flash condensing steam plant outperforms all of the above for reservoir temperatures below 210°C. The more complicated double-flash condensing plant and the condensing steam / binary hybrid provide the most electrical capacity of all, outperforming the single-flash condensing steam plant by 10 percent–20 percent, and are essentially equivalent for reservoir temperatures up to 200°C or so; above this temperature, the hybrid plant has a slight (~10 percent) advantage.

Results for the same 300 mm ID production well are shown in Figures 8.4, but in this case the well has been equipped with a lineshaft-type downhole pump. There are really only four independent curves in Figure 8.4; as noted in Section 7, in this range the steam / binary hybrid plants provide “optimum”

Table 8.1. Detailed calculations available in Appendices.

	Appendix									
	A	B	C	D	E	F	G	H	I	J
Downhole Pump:										
None (self-discharging well)	X	X								
Lineshaft pump			X	X	X	X				
Submersible pump							X	X	X	X
Plant Type:										
None (well performance only)			X	X			X	X		
Single-flash backpressure steam turbine	X	X			X	X			X	X
Single-flash condensing steam turbine	X	X			X	X			X	X
Double-flash backpressure/condensing turbine	X	X								
Double-flash condensing steam turbine	X	X			X	X			X	X
Binary plant					X	X			X	X
Binary plant using cold water recirculation	X	X								
Backpressure steam/binary plant hybrid	X	X								
Condensing steam/binary plant hybrid	X	X			X	X			X	X
Well Inside Diameter:										
75 mm	X	X								
100 mm	X	X					X	X	X	X
125 mm	X	X					X	X	X	X
150 mm	X	X	X	X	X	X	X	X	X	X
200 mm	X	X	X	X	X	X	X	X	X	X
250 mm			X	X	X	X	X	X	X	X
300 mm	X	X	X	X	X	X	X	X	X	X

Table 8.1 (cont'd). Detailed calculations available in Appendices.

	Appendix									
	A	B	C	D	E	F	G	H	I	J
Well Feedpoint Depth:										
300 meters	X		X		X		X		X	
600 meters	X	X	X	X	X	X	X	X	X	X
900 meters	X		X		X		X		X	
1200 meters	X		X		X		X		X	
Well Productivity Index:										
2 kg/s/bar	X		X		X		X		X	
4 kg/s/bar	X	X	X	X	X	X	X	X	X	X
Infinity	X		X		X		X		X	
Reservoir Liquid CO₂ Content:										
No CO ₂	X	X	X	X	X	X	X	X	X	X
0.25%		X		X		X		X		X
0.50%		X		X		X		X		X
0.75%		X		X		X		X		X
1.00%		X		X		X		X		X
Reservoir Temperature:										
100°C to 240°C	X	X	X	X	X	X	X	X	X	X
Piezometric Surface Depth:										
0 to 250 meters	X	X	X	X	X	X	X	X	X	X

PIEZOMETRIC SURFACE DEPTH: 100 m
WELL PRODUCTIVITY INDEX: 4 kg/s/bar

WELL FEEDPOINT DEPTH: 600 m
RESERVOIR CO₂ CONTENT: None

- ▲ self-discharging well, 75 mm ID
- self-discharging well, 150 mm ID
- with submersible pump, 150 mm ID
- self-discharging well, 300 mm ID
- with lineshaft pump, 300 mm ID

Figure 8.1. Maximum net electrical generating capacities as functions of reservoir temperature for optimized flash-steam plants using self-discharging and pumped wells of various diameters.

PIEZOMETRIC SURFACE DEPTH: 100 m
WELL PRODUCTIVITY INDEX: 4 kg/s/bar

WELL FEEDPOINT DEPTH: 600 m
RESERVOIR CO₂ CONTENT: None

- ▲ self-discharging well, 75 mm ID
- self-discharging well, 150 mm ID
- with submersible pump, 150 mm ID
- self-discharging well, 300 mm ID
- with lineshaft pump, 300 mm ID

Figure 8.2. Maximum net electrical generating capacities as functions of reservoir temperature for optimized binary-cycle and flash/binary hybrid plants using self-discharging and pumped wells of various diameters.

SELF-DISCHARGING WELL
PIEZOMETRIC SURFACE DEPTH: 100 m
WELL PRODUCTIVITY INDEX: 4 kg/s/bar

WELL INSIDE DIAMETER: 300 mm
WELL FEEDPOINT DEPTH: 600 m
RESERVOIR CO₂ CONTENT: None

Figure 8.3. Maximum net electrical generating capacities as functions of reservoir temperature for seven optimized wellhead power plant designs using a large-diameter (300 mm ID) self-discharging production well.

WELL WITH LINESHAFT DOWNHOLE PUMP
PIEZOMETRIC SURFACE DEPTH: 100 m
WELL PRODUCTIVITY INDEX: 4 kg/s/bar

WELL INSIDE DIAMETER: 300 mm
WELL FEEDPOINT DEPTH: 600 m
RESERVOIR CO₂ CONTENT: None

Figure 8.4. Maximum net electrical generating capacities as functions of reservoir temperature for seven optimized wellhead power plant designs using a large-diameter (300 mm ID) production well equipped with a lineshaft downhole pump.

steam-turbine generating fractions of zero, so they are both equivalent to a simple binary plant. Similarly, the double-flash backpressure/condensing plant is really just equivalent to a single-flash condensing steam plant. The single-flash backpressure steam turbine design is vastly inferior to the others in this case, as Figure 8.4 shows. The binary plant provides the best performance of all for reservoir temperatures over 145°C or so, and outperforms the single-flash condensing turbine for all temperatures above 120°C. This is, of course, in sharp contrast to the situation for the 300 mm ID self-discharging well (Figure 8.3).

Next, we consider a relatively large-diameter slim hole (150 mm ID). Figure 8.5 indicates the performance in the absence of a downhole pump of each of the seven power plant design concepts as functions of reservoir temperature. For this smaller diameter well, the performance degradation caused by two-phase flow in the formation does not appear except for temperatures above 235°C or so. Qualitatively, the results are similar to those from the larger (300 mm ID; Figure 8.3) self-discharging well. As usual, the single-flash backpressure turbine makes the least effective use of the resource. The double-flash backpressure/condensing plant and the backpressure steam / binary hybrid represent an improvement—they are essentially equivalent in capacity for reservoir temperatures below 170°C, with the hybrid plant having the advantage at higher temperatures. All of these plants are outperformed by the simple binary plant with a cold-water recirculation system at all temperatures. Again, the single-flash condensing steam turbine provides a further improvement in performance (compared to the binary plant) for temperatures below 215°C, and the highest capacities of all are provided by the double-flash condensing steam plant and the condensing steam / binary hybrid plant, with a slight advantage to the latter for reservoir temperatures above 200°C.

Figure 8.6 indicates the performance results for the same 150 mm ID slim hole equipped with a submersible downhole pump. Again, there are really only four independent curves in Figure 8.6. The binary plant, backpressure steam / binary hybrid and condensing steam / binary hybrid all collapse into a single curve for temperatures less than about 210°C. Above that temperature, there is a slight advantage to the condensing hybrid plant, but this advantage is probably only theoretical since, as indicated by the broken lines, sustained operation of a submersible downhole pump is not presently practical for reservoir temperatures above 205°C or so. Similarly, the double-flash backpressure/condensing steam plant has no advantage over the simpler single-flash condensing steam plant within the practical temperature range. As previously, the binary plant provides the maximum electrical capacity for reservoir temperatures above 145°C. The single-flash backpressure steam turbine is less effective than any of the others by a factor of at least four for practical temperatures (< 205°C).

Finally, Figure 8.7 shows performance results for a small-diameter (75 mm ID) self-discharging slim hole. As noted earlier, downhole pumps are impractical for wells of such small diameter. The general pattern of performance is similar to that for the larger-diameter self-discharging wells (Figures 8.5 and 8.3; 150 mm ID and 300 mm ID), with the binary plant with cold-water recirculation outperforming all three designs involving backpressure steam turbines, and in turn being outperformed by the various condensing steam plant designs. It is certainly worthy of note, however, that electrical capacities of several hundred kilowatts are possible using self-discharging 75 mm ID slim holes so long as reservoir temperatures exceed 200°C or so.

SELF-DISCHARGING WELL
PIEZOMETRIC SURFACE DEPTH: 100 m
WELL PRODUCTIVITY INDEX: 4 kg/s/bar

WELL INSIDE DIAMETER: 150 mm
WELL FEEDPOINT DEPTH: 600 m
RESERVOIR CO₂ CONTENT: None

Figure 8.5. Maximum net electrical generating capacities as functions of reservoir temperature for seven optimized wellhead power plant designs using a large (150 mm ID) self-discharging slim hole.

Figure 8.6. Maximum net electrical generating capacities as functions of reservoir temperature for seven optimized wellhead power plant designs using a large (150 mm ID) slim hole equipped with a submersible downhole pump.

Figure 8.7. Maximum net electrical generating capacities as functions of reservoir temperature for seven optimized wellhead power plant designs using a small (75 mm ID) self-discharging slim hole.

8.4 Utility of Downhole Pumps for Various Plant Designs

The results reported above (see Figures 8.4 and 8.6) strongly indicate that the combination of downhole pumps with wellhead single-flash backpressure steam turbines is particularly ineffective. We expect, of course, that the relative utility of downhole pumps will depend upon which plant design concept is selected, as well as upon well diameter, reservoir temperature, and the remaining free parameters as well. In this discussion, attention is restricted to the four plant designs which are truly independent and practical for operation with downhole pumps—single-flash backpressure steam turbines, single-flash condensing steam turbines, double-flash condensing steam turbines, and binary plants (with a cold-water recirculation system for application to self-discharging wells). Also, we again consider only “base case” conditions—no CO₂, 600 m feedpoint depth, 4 kg/s/bar productivity index, and 100 m piezometric surface depth.

Figure 8.8 shows, for these four plant designs, the performance obtainable (as a function of reservoir temperature) from a 300 mm ID production well, both with and without a lineshaft-type downhole pump. As noted above, the combination of pumped wells with single-flash backpressure steam turbines seems particularly ineffective—Figure 8.8 indicates that, in fact, such a plant will produce more electricity without the pump than with the pump. The other three plant designs, by contrast, benefit from the presence of the downhole pump, both in the amount of power available at a particular reservoir temperature and in the minimum reservoir temperature required to produce useful amounts of power at all. At least up to ~185°C (the maximum reservoir temperature at which this pump may operate in a 600 meter well to avoid pump cavitation), the electrical power obtainable using the pump significantly exceeds that available with self-discharge for all three designs.

Figure 8.9 compares, in a similar fashion, the electrical capacities available using a submersible downhole pump versus the self-discharge case for a 150 mm ID slim hole. The broken lines indicate where temperatures exceed 205°C and present-day submersible pumps will be unreliable. Again, the performance of the single-flash backpressure steam turbine is actually better without the pump. The remaining plant designs benefit from the pump for sufficiently low reservoir temperatures, but the pump's advantage decreases with increasing reservoir temperature and (in contrast to the larger 300 mm ID well discussed above) becomes a liability at higher temperatures. Downhole pumps seem to be most useful for binary plants, but even in this case if the reservoir temperature exceeds 200°C a self-discharging 150 mm ID well can produce more power from a binary plant with a cold-water recirculation system than a similar well with a downhole submersible pump driving a conventional binary plant. Downhole pumps cease being advantageous for single-flash and double-flash condensing steam plants at reservoir temperatures of 154°C and 158°C, respectively.

For “base case” conditions, the electrical generating capacity of a 300 mm ID production well is illustrated in Figure 8.10, which compares the performance using a lineshaft downhole pump and a binary wellhead plant (in grey) with self-discharging performance using three different types of flash plant (in black). As noted above, the downhole pump / binary plant combination provides significantly better performance throughout the reservoir temperature range in which it is practical. Figure 8.11 provides the corresponding comparison for a 150 mm ID slim hole (with a submersible pump driving the binary

Figure 8.8. Comparisons of system performance using a 300 mm ID production well with and without a lineshaft-type downhole pump driving four popular wellhead plant designs. Binary plant with self-discharging well uses cold-water recirculation system.

Figure 8.9. Comparisons of system performance using a 150 mm ID slim hole with and without a submersible-type downhole pump driving four popular wellhead plant designs. Binary plant with self-discharging well uses cold-water recirculation system.

Figure 8.10. Comparison of optimized performance of system using downhole pump and wellhead binary plant with performance of self-discharging well driving various flash-steam plants for well inside diameter = 300 millimeters.

Figure 8.11. Comparison of optimized performance of system using downhole pump and wellhead binary plant with performance of self-discharging well driving various flash-steam plants for well inside diameter = 150 millimeters.

plant in this case). Tabular results are provided in Table 8.2 for these latter cases. The advantage of the traditional pumped well / binary plant design is clearly much less for slim holes than for large production wells, and the range of reservoir temperatures over which this arrangement is advantageous appears to decrease with decreasing well diameter.

8.5 Influence of Parameter Variations on Optimum Electrical Capacity

All calculations presented so far in Section 8 pertain to “base case” conditions (that is, piezometric surface depth = 100 m; well feedpoint depth = 600 m; well productivity index = 4 kg/s/bar; and no reservoir CO₂). Next, we examine the effects of varying these parameters, one at a time, upon the computed electrical generating capacity of some the systems which have been studied.

Figure 8.12 considers a self-discharging large-diameter production well (300 mm ID) driving a well-head single-flash condensing steam turbine power plant. Electrical generating capacity increases with (1) increasing reservoir pressure (decreasing piezometric surface depth) and (2) increasing well productivity index. Increasing the feedpoint depth or the reservoir CO₂ content may either increase or decrease the electrical capacity, depending on reservoir temperature. Decreasing feedpoint depth slightly increases electrical capacity at low temperatures, but shallower wells enter the two-phase flow regime in the formation at lower reservoir temperatures, adversely affecting performance. Similarly, at low reservoir temperatures, higher CO₂ contents provide additional gas-lift improving system performance, but the additional CO₂ can also cause two-phase flow in the reservoir to occur at lower temperatures.

Similarly, Figure 8.13 illustrates the situation for a large-diameter (300 mm) well equipped with a lineshaft-type downhole pump and a wellhead binary power plant. Reservoir pressure has a significant influence on performance potential, but only at low temperatures. There is also a significant effect of productivity index at low reservoir temperatures; the maximum operational reservoir temperature also declines substantially with decreasing productivity index, and (as indicated by the parenthesis) the curve for 2 kg/s/bar is not shown since the maximum reservoir temperature for that case is less than 100°C. The system will not function for a feedpoint depth of 300 meters since the minimum inlet depth to avoid steam entry into the pump is deeper than 300 m, but otherwise the performance is relatively insensitive to feedpoint depth. Note also that the system will not operate at high CO₂ contents (0.75 percent and 1.00 percent).

The situation is somewhat different if the lineshaft pump is replaced by a submersible-type downhole pump (Figure 8.14). Although in general the electrical capacities are somewhat lower using a submersible pump than is the case with a lineshaft pump in this large-diameter (300 mm) well (owing to motor-power limitations for the submersible design), the breadth of operating conditions is actually greater. Power can be produced for all values of the productivity index and reservoir CO₂ content considered, for example, unlike the previous case. It is important to remember, of course, that practical applications of submersible pumps are presently limited to reservoir temperatures below 205°C. These results suggest, however, that improvements in downhole motors (to deliver greater amounts of shaft power and to withstand higher temperatures) may cause the performance of large-diameter submersible pumps to reach and surpass that of lineshaft pumps in the future, even for conventional large-diameter production wells.

Table 8.2. Maximum electrical generating capacities for pumped and self-discharging 150 mm ID slim holes.

Plant:	Binary Plant	Single-Flash Backpressure Steam Plant	Single-Flash Condensing Steam Plant	Double-Flash Condensing Steam Plant
Pump:	Submersible Pump	Self-Discharge	Self-Discharge	Self-Discharge
Reservoir Temperature (°C)	(kWe)	(kWe)	(kWe)	(kWe)
100	37	None	None	None
110	81	None	None	None
120	141	None	None	None
130	217	None	None	None
140	311	None	None	None
150	432	19	263	325
160	575	61	398	498
170	698	120	561	704
180	861	199	761	952
190	1005	301	993	1238
200	1117	428	1261	1567
210	1252*	581	1563	1938
220	1345*	762	1909	2352
230	1356*	974	2286	2810
240	1286*	1068	2330	2875

*Maximum submersible motor temperature exceeded.

Figure 8.12. Effects of key parameters on optimized net electrical generating capacities of wellhead single-flash condensing steam turbines driven by self-discharging 300 mm ID production wells.

BINARY PLANT DRIVEN BY WELL EQUIPPED WITH LINESHAFT DOWNHOLE PUMP
WELL INSIDE DIAMETER: 300 mm

BASE-CASE CONDITIONS:

PIEZOMETRIC SURFACE DEPTH: 100 m
WELL PRODUCTIVITY INDEX: 4 kg/s/bar

WELL FEEDPOINT DEPTH: 600 m
RESERVOIR CO₂ CONTENT: None

Figure 8.13. Effects of key parameters on optimized net electrical generating capacities of wellhead binary plants driven by 300 mm ID production wells equipped with lineshaft downhole pumps.

Figure 8.14. Effects of key parameters on optimized net electrical generating capacities of wellhead binary plants driven by 300 mm ID production wells equipped with submersible downhole pumps.

Next, attention will be shifted from standard-size production-well applications (300 mm ID) to large slim holes (150 mm ID) where downhole pumps may still be applied. Figure 8.15 provides results for a self-discharging 150 mm ID slim hole driving a single-flash condensing wellhead steam turbine. Qualitatively, the results are similar to the 300 mm case (compare to Figure 8.12), but the sensitivities of the electrical capacity to the various parameters is somewhat less, particularly the sensitivity to productivity index. Figure 8.16 shows results for the 150 mm ID, lineshaft-pump, binary-plant case. As has already been established, the range of applicability of lineshaft pumps is very limited at the 150 mm size. Piezometric surface depths of 150 m or greater cannot be accommodated at all, nor can productivity index = 2 kg/s/bar, nor CO₂ contents even as low as 0.25% by mass. The situation is vastly better if a submersible pump is applied (Figure 8.17); predicted optimum performance is fairly insensitive to most of the parameters, although some degradation occurs for high levels of CO₂ and for low reservoir pressures. The limitation to 205°C reservoir temperature also applies.

Finally, consider a small-diameter (75 mm) slim hole. Since this is too small for downhole pumps, attention is necessarily limited to the self-discharging, single-flash condensing-plant case (Figure 8.18). Performance generally improves with increasing reservoir pressure, decreasing feedpoint depth, and increasing reservoir CO₂ content. Sensitivity to feedpoint depth is greater than in the larger-diameter cases owing to the greater importance of lateral heat losses from the well to the formation for smaller-diameter wells. Performance is essentially insensitive to reservoir permeability, however, for such a small system.

SINGLE-FLASH CONDENSING STEAM TURBINE DRIVEN BY SELF-DISCHARGING WELL
WELL INSIDE DIAMETER: 150 mm

BASE-CASE CONDITIONS:

PIEZOMETRIC SURFACE DEPTH: 100 m

WELL FEEDPOINT DEPTH: 600 m

WELL PRODUCTIVITY INDEX: 4 kg/s/bar

RESERVOIR CO₂ CONTENT: None

Figure 8.15. Effects of key parameters on optimized net electrical generating capacities of wellhead single-flash condensing steam turbines driven by self-discharging 150 mm ID slim holes.

BINARY PLANT DRIVEN BY WELL EQUIPPED WITH LINESHAFT DOWNHOLE PUMP
WELL INSIDE DIAMETER: 150 mm

BASE-CASE CONDITIONS:

PIEZOMETRIC SURFACE DEPTH: 100 m
WELL PRODUCTIVITY INDEX: 4 kg/s/bar

WELL FEEDPOINT DEPTH: 600 m
RESERVOIR CO₂ CONTENT: None

Figure 8.16. Effects of key parameters on optimized net electrical generating capacities of wellhead binary plants driven by 150 mm ID slim holes equipped with lineshaft downhole pumps.

BINARY PLANT DRIVEN BY WELL EQUIPPED WITH SUBMERSIBLE DOWNHOLE PUMP
WELL INSIDE DIAMETER: 150 mm

BASE-CASE CONDITIONS:

PIEZOMETRIC SURFACE DEPTH: 100 m

WELL FEEDPOINT DEPTH: 600 m

WELL PRODUCTIVITY INDEX: 4 kg/s/bar

RESERVOIR CO₂ CONTENT: None

Figure 8.17. Effects of key parameters on optimized net electrical generating capacities of wellhead binary plants driven by 150 mm ID slim holes equipped with submersible downhole pumps.

Figure 8.18. Effects of key parameters on optimized net electrical generating capacities of wellhead single-flash condensing steam turbines driven by self-discharging 75 mm ID slim holes.

9

Conclusions and Recommendations

The most important conclusion of this study is that slim holes driving wellhead geothermal power plants are capable of meeting the needs of rural electrification projects which require electrical generating capacity in the 100 kWe - 1000 kWe range. Self-discharging 75 mm ID slim holes can produce more than 100 kWe so long as reservoir temperatures exceed 170°C or so; at higher temperatures (240°C), 75 mm slim holes are capable of up to 600 kWe. Larger self-discharging slim holes (150 mm ID) are capable of over 1000 kWe capacity for reservoir temperatures greater than 180°C, and can reach 3000 kWe at 240°C; even for temperatures as low as 145°C, capacities of 250 kWe are possible. At lower reservoir temperatures, downhole pumps will be required; a 150 mm ID slim hole equipped with a submersible-type downhole pump can produce over 100 kWe at a reservoir temperature of only 105°C; the potential rises to 430 kWe at 150°C and to 1100 kWe at 200°C.

For slim-hole applications, lineshaft-type downhole pumps are impractical. Exploitation of low-temperature resources using slim holes will require submersible-type downhole pumps, usually driving either wellhead binary plants or wellhead single-flash condensing steam plants with low operating pressures. Generally speaking, of the more complex plant designs considered in this study (double-flash and hybrid plants), the most promising would appear to be the double-flash condensing plant. Double-flash and hybrid plants which incorporate a backpressure-type high-pressure stage are of extremely limited utility.

9.1 Optimization Incorporating Project Economics

For the present study, well/plant combinations were evaluated and optimized based upon a simple performance criterion: maximum exportable power (that is, gross power from the generator minus various parasitic loads which include NCG pumps, condensate pumps, cooling tower fans, recirculation pumps, hydrocarbon feed pumps, and downhole production pumps of either lineshaft or submersible design). This general approach is appropriate for the objective of the present study (which was to identify and appraise those conditions under which small-scale geothermal wellhead power stations can be effectively applied to off-grid rural electrification projects).

Other issues are obviously pertinent as well, however. In particular, the costs involved in producing the electricity (both initial capital costs for exploration, drilling, and plant construction and long-term maintenance expense to keep the project operating) have not been taken into account except in the most cursory fashion. Clearly, if development plan "A" will involve substantially greater costs of this type over the project's lifetime, then development plan "B" (with somewhat lower electrical generating capacity but lower capital and maintenance costs) should not be automatically rejected. It is for this reason that the design concept involving a single-flash backpressure steam turbine plant driven by a self-discharging well has been retained throughout this study despite the relatively low efficiency of this

approach. Although several of the more elaborate wellhead generator designs (double-flash plants, hybrids) appear to offer higher electrical capacities than some of the simpler designs, the relative advantages may be so marginal as not to justify the additional expense.

Incorporation of economic issues is a non-trivial matter, of course. Project costs include the costs of drilling and completing the production well (and the injection well), constructing and installing the wellhead generator system, purchase and installation of a downhole pump (if needed), and all the costs entailed in the maintenance and support of the system throughout its economic lifetime. Risk factors and lead-time costs will vary from one cost component to the next. Some of these costs will vary with time, and with local economic conditions (including such issues as local tax law, environmental regulations, labor costs, availability and costs of expendables, currency fluctuations, etc.). Clearly, any "one size fits all" economic analysis will be only approximate. Despite this, it seems clear that the economic feasibility of minigeothermal rural electrification using slim holes as the production wells requires evaluation and comparison with other competitive approaches.

In addition to the above "global" (project-scale) issues, other economic considerations warrant examination in such matters as selection of power plant and well design. As an example, Figure 7.27 shows that the "optimum" (maximum power) operating pressure for a single-flash condensing turbine power plant under particular well and reservoir conditions is 1.75 bars (at which the generating potential is 1245 kWe). As a practical matter, however, the capital cost of a steam turbine plant increases significantly (for a fixed generating capacity) as the operating pressure decreases, owing to the increasing size of the turbine assembly. As noted in Figure 7.41, performance can still be quite acceptable if higher design operating pressures are used for this particular case:

Operating Pressure (bars)	Electrical Capacity (kWe)
1.75	1245
2.00	1241
2.50	1217
3.00	1164

so that it is highly likely that a system optimization based on maximizing "electrical capacity per unit project capital cost" or some similar criterion would result in higher operating pressures (and somewhat lower generating capacities) than the present optimization which simply maximizes electrical capacity without regard to costs.

9.2 Optimization Incorporating Long-Term Effects on the Reservoir

For purposes of this study, it was assumed that the rate of energy production from the geothermal reservoir (for purposes of small off-grid electrification projects) is small enough (relative to the rate of natural recharge) that the resource can be regarded as inexhaustible. In reality, of course, geothermal power production will have the effect of depleting the geothermal resource (by both pressure and temperature reduction). For sufficiently low rates of electricity generation, this rate may be low enough that natural recharge will maintain the system indefinitely.

For small reservoirs, high exploitation rates, low-permeability systems and/or other adverse conditions, however, the extraction of geothermal energy may cause the reservoir conditions (pressure, temperature, CO₂ content, etc.) to change during the practical lifetime of the small-scale geothermal project by sufficient amounts that the system properties change significantly and require re-evaluation of the proper future energy production strategy. Accordingly, another issue involved in the selection of a power production strategy is the deleterious influence upon the resource of the various candidate plans for the project, and the degree to which the proposed system design can be readily re-configured to accommodate long-term changes in reservoir conditions.

As a crude example, assume that under certain circumstances the local market demand for power can be met by either (1) a 150 mm ID slim hole with a submersible downhole pump driving a wellhead binary plant, or (2) a 300 mm ID self-discharging production well driving a backpressure single-flash steam plant. Assume further that the costs of the two systems are comparable (with the increased costs of the binary plant and the downhole pump offset by extra drilling costs for the large diameter production well). Under these conditions the first scheme would be preferable, since the binary plant will return all produced fluid to the subsurface whereas the separated steam fraction for the backpressure plant is lost to the atmosphere. As a result, the backpressure plant will experience more rapid reservoir pressure decline and have a shorter useful lifetime.

9.3 Proof-of-Concept Experiments

One troubling aspect of the present study is the degree to which it relies on engineering estimates of the properties and capabilities of various types of equipment which has so far not been built or tested. This is certainly true for slim-hole downhole geothermal pumps. Experience with small-scale (< 1000 kWe) wellhead geothermal power plants is also sparse. Three general approaches to power plant design appear most promising: single-flash backpressure steam turbines (owing mainly to their low cost and portability), single-flash condensing steam turbines (particularly low-pressure and sub-atmospheric designs) and small binary plants. While the other design concepts are not ruled out at this stage, the above three deserve the highest priority.

Design, prototype construction, and field testing would do much to reduce the ambiguities in the various design studies employed in the present project, and would also provide much-needed practical data on costs, reliability, maintenance requirements, and performance of the integrated well/pump/plant systems. A field-testing program of this type will also be essential to demonstrate the feasibility of small-scale slim-hole geothermal development for rural electrification to the investment community.

10

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