PERFORMANCE AND OPERABILITY STUDY OF THE RAFT RIVER
5 MW(e) PILOT GEOTHERMAL POWER PLANT STEADY-STATE BEHAVIOR

MASTER

C. J. Bliem

November 13, 1979

Conversion Technology and Engineering

EG&G Idaho, Inc.

IDAHO NATIONAL ENGINEERING LABORATORY

DEPARTMENT OF ENERGY

IDAHO OPERATIONS OFFICE UNDER CONTRACT DE-AC07-76IDO1570

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Introduction and Summary

Estimates of the steady-state performance of the Raft River 5 MW(e) Pilot Geothermal Power Plant have been made using an existing digital computer simulation which has been updated to reflect the individual performance of the actual components in the plant. The plant output (gross and net) has been obtained as a function of geothermal fluid inlet temperature and flow rate and isobutane preheater bypass (boiler inlet subcooling). The plant performance has been determined for two ambient conditions (summer and winter extremes), and different conditions of heat exchanger fouling. These results along with transient simulations, which will be made in the near future, are needed to establish operating procedures and set point, check out control system, determine control gains, and set alarm points.

During the design phase of the 5 MW project the initial model using the Time and Frequency (TAF) code was developed. For the present work, the original model has been updated to reflect actual components such as the pumps, turbine, cooling tower and heat exchangers as they are being built. Some additional modeling including a new cooling tower model have been incorporated. The digital model predicts both steady-state and transient response of the plant. The model represents the components of the geothermal fluid, isobutane and cooling water loops in terms of mass and energy balances, thermodynamic and transport properties, flow resistances, convection heat transfer correlations, and valve and pump characteristics. Analog controls are incorporated in the model. The plant boundary conditions (geothermal fluid conditions at the plant inlet and ambient air conditions), operating parameters (heat exchanger fouling and isobutane preheater bypass), and valve positions may be represented in a static or dynamic fashion. Appendix A contains a brief description of the executive code (TAF), the mathematical model of the plant, and the input-output requirements to obtain steady-state and transient solutions.

This report contains a description of the comparison made between code calculations and specified performance of the components and system, the plant performance for steady-state plant operation with varying geothermal fluid inlet temperatures and flow rates is presented in graphical form along with an example of plant performance with the existing geothermal field. The gross and net output of the plant have been determined for two fouled conditions: design fouling and ten (10) percent of design fouling (a condition which approaches clean operation) and two ambient conditions: the summer and winter extreme conditions. The plant output was determined as a function of geothermal fluid flow rate, plant inlet temperature and desired isobutane subcooling entering the boilers. Preheater bypass flows are necessary to achieve a certain subcooling for the fluid entering each of the boilers to prevent excessive cavitation in the boiler feed valves and damage to the piping and possibly the boilers. (For example, 5°F subcooling on each boiler is achieved with design geothermal fluid flow at 280°F in the summer with ten (10) percent fouling with 6.4 percent of the flow bypassing the low preheater and none bypassing the high preheater). Methods are given to obtain the amount of bypass necessary to achieve a given boilers subcooling. Corrections on the plant output with no bypass may then be calculated. The amount of subcooling necessary is discussed in a subsequent section of this report.

Limits on plant steady-state operation were investigated. These limits include a limit resulting from the pressure head performance of the isobutane feed pump, limits on geothermal fluid velocities in the heat exchangers get by fouling considerations, and limits on the boiler subcooling discussed above.

In summary, the results of this study to date are:

1. The digital simulation model has been checked out by comparison with component data from independent sources. The only significant discrepancy is the condenser performance. This difference will not impact predicted plant

operation or operating procedures. The power outputs, however, are approximately 8% lower than those predicted with condenser performance nearer the manufacturer's specified values.

2. The turbine efficiency is lower than that assumed in the original analyses thereby giving lower generator output than estimated earlier.

3. Curves of system performance have been generated for various geothermal fluid inlet flow rates and temperatures under summer and winter extreme ambient conditions and clean and design fouled condition.

4. Constraints on system performance have been identified. They include high boiler pressure limit caused by the isobutane fuel pump capacity, geothermal fluid flow rate limits required to minimize scaling, corrosion and erosion of heat exchanger tubes and boiler inlet subcooling limits to minimize flashing and cavitation problems.

Code Verification

The present computer model has been assessed by comparison of calculated results with data from other sources. The major items addressed in the code verification are the heat exchanger performance and overall cycle performance at the nominal design conditions. The heat exchangers calculations in the code have been compared with two independent sources: 1) the manufacturer's specifications at design condition, and 2) prediction of Heat Transfer Research, Inc. (HTRI) computer codes for both the design conditions and for four typical off-design conditions at fouling of ten (10) percent of the design value, for the boilers and preheaters, RKH for boilers and CST for the condenser. No off-design values were calculated for the condenser because of uncertainties in the condenser code uncovered during the design case modeling. (See Reference 1)

Tables 1 and 2 summarize the results of these comparisons. The results of these comparisons were within typical uncertainties associated with heat transfer phenomenon; the differences in both h 's and U 's for the design case

were all within twenty percent of the HTRI calculated values with four exceptions. Comparison between the code at the manufacturer's specifications reveal four cases in which the differences exceed twenty percent. Two of the discrepancies are the shellside "h" in the low preheater and the tubeside "h" in the high preheater, each of which (not being the controlling "h" have small enough effect on the overall heat transfer coefficient to keep the disagreement in "U" below fifteen percent. It is noted that the other tubeside "h" values compare favorable with the code calculated values. It is felt that the differences here are with the method used by the manufacturer in his calculation. The shellside "h" in the condenser has the largest difference (89 percent) when compared with manufacturer's data which make the "U" lower than the manufacturer's value by thirty-eight percent. It is noted that this "h" is in good agreement with the HTRI value quoted in Reference 1 which leads to the conclusion that the manufacturer's value may be high. An HTRI run with a new version of the CST code, however, gives results closer to the manufacturer's value (approximately twenty (20) percent lower). When the system code was run with this "h" increased by eighty-nine percent, the differences in statepoints from the code run with the lower "h" were minor. The high boiler temperature was decreased by 1°F ; the low boiler temperature by 2°F ; and the condenser temperature by 6°F . The subcooling margins entering the high pressure and low pressure boilers respectively were increased by 0.1°F and 0.2°F . It is felt, therefore, that the difference in condenser performance will have negligible effects on the operating characteristics of the system and the operational statepoints generated. Because the condenser pressure was lowered, the generator power increases by about 8%. The method of computing "h" for isobutane condensation will be revised to more closely correspond to the new HTRI estimate.

A comparison was made between the nominal design statepoints from the thermodynamic analysis of Reference 3 and the code calculation of the same basic

conditions. Figure 1 and the associated table summarize the calculated state-points at nominal design conditions (nominal fouling; geofluid inlet temperature, 290°F; geofluid flow rate 1.04×10^6 lb/hr; summer ambient conditions, 92°F drybulb temperature, 65°F wetbulb temperature. The initial thermodynamic cycle analysis (nominal design of Ref. 3) assumed the following: 10°F approach temperature difference in the cooling tower and boilers/heaters no subcooling entering the boilers, 4°F subcooling leaving the condenser and a cooling water flow rate about thirteen percent lower than the currently anticipated rate with the specified equipment. The current computer simulation allows the pinch points to be determined by the heat exchanger performance, the boiler subcooling determined by the preheater performance, but no subcooling leaving the condenser.

Experience indicates that the 4°F subcooling leaving the condenser will have negligible effect on the other system statepoint. The case mentioned above in which the condenser outlet changed 6°F caused negligible changes in subcooling and changes of only 1 to 2°F in boiler temperatures. Considering all state-points the maximum temperature difference between solutions is 4°F; the isobutane flow rates are all within 3% of each other. (The boiler pinch points in the simulation are 9°F; condenser pinch is 11°F; and the cooling tower approach is 8°F). One difference noted between the thermodynamic analysis and the code calculation is the power produced. The thermodynamic analysis gives a generator power of 5.0 MW(e) while that predicted by the code is 4.54 MW(e). The reason for this difference is the efficiency differences in the turbine. The prior thermodynamic analysis assumed 81 percent overall engine efficiency. It should be noted that when the turbine was specified, it was not required to produce 5 MW(e) at the nominal design conditions, but to maximize the power produced overall engine efficiency used in the current simulation represents the approximate combined efficiency of the turbine-generator.

Plant Steady-State Performance

The steady-state plant performance has been estimated for a range of geothermal plant inlet temperatures from 270°F to 290°F and geothermal fluid flow rates from 80% to 120% of the nominal design value of 1.04×10^6 lb/hr. Cases have been run for 10% and 100% design fouling. Design fouling resistance on the geothermal side of heat exchangers - $0.0015 \text{ hr.}^\circ\text{F ft}^2/\text{Btu}$ on isobutane side of heat exchanger $0.005 \text{ hr.}^\circ\text{F ft}^2/\text{Btu}$ on cooling water side of condenser - $0.0010 \text{ hr.}^\circ\text{F ft}^2/\text{Btu}$ and summer ambient conditions which are exceeded 1% of the time (drybulb temperature 92°F, wetbulb temperature, 65°F), and the 1% winter ambient conditions (drybulb temperature 8°F, wetbulb temperature 8°F). Cases have been run with 1% of design fouling. The results indicated that the generator power increases approximately 1% over a comparable case at 10% fouling; the high pressure boiler was the same. It is, therefore, felt that 10% fouling solutions are typical of the unfouled plant performance.

For the cases presented in this section, all valves were set at fixed positions with the exception of the boiler feed valves which moved to provide the proper match between the isobutane feed pump and the demand of the boilers. Turbine throttle valves were set to give approximately 5 psi drop from boiler pressure. Cooling water flow rate was maintained constant and both fans were used in the cooling tower.

Each of the cases mentioned above is presented in a separate set of curves.

For example: Figure 2, 3, and 4 depict the performance of the plant at the summer extreme condition with 10% of design fouling.

Figure 2 gives the generator output and net plant effectiveness parasitic loads include isobutane feed pump, geofluid booster pump, cooling water pump and cooling tower fans. Vendor data was used to represent these loads in the program as a function of geothermal fluid inlet temperature and mass flow rate.

These results are for no preheater bypass. A subsequent section will discuss the choice of boiler inlet subcooling. In the event that it is necessary to produce a given subcooling by bypassing isobutane around the preheaters, Figures 3 and 4 give the amount of bypass necessary to achieve a given amount of boiler subcooling and small corrections may be made on the values read from Figure 2. (The corrections should not change the trends indicated in the Figure.) Figure 3 allows the determination of the low pressure boilers subcooling at now bypass and 10% bypass (percentages are of mixed flow leaving the preheater) for a given geothermal fluid temperature and flow rate. Linear interpolation will give the bypass for any subcooling desired.

Figure 4 gives the high pressure boiler subcooling with no bypass. Notes on the Figure give the additional subcooling resulting from each percent low preheater bypass (which is now known) and that for each part of high preheater bypass. With this the high preheater bypass may be determined. Figures 5, 6, and 7 give the same information for the winter extreme ambient conditions. Figures 8, 9, 10, 11, 12, and 13 give the information described above for nominal design fouling for summer and winter ambient conditions respectively.

Having determined the high and low preheater bypass percentages, the impact on the power may be assessed. For all conditions of fouling and all ambient conditions, the generator output is decreased by 0.17% for each percent of low preheater bypass and 0.005 MW(e) for each percent of high preheater bypass. (These are empirical corrections which best fit the data.) The net plant output is decreased by the same amount. An example of the use of these curves to find the plant operating point is given in Appendix B.

Figure 14 gives the cooling water evaporation in summer and winter for no preheater bypass and/or 10% fouling conditions. Corrections for preheater bypass at other fouling conditions are indicated on the Figure.

System Operation Constraints

There are certain constraints on the operation of the power plant. They include preheater isobutane bypass to give boiler inlet subcooling, isobutane pump-turbine operating limit, and scaling and corrosion limitations on the geothermal fluid velocities in the heat exchangers. These are all discussed in this section.

The primary reason for specifying a subcooling margin at the outlet of the boiler feed valves is to prevent cavitation and flashing in the valves or the lines between the valves and the boilers. Current simulation indicates that the high boiler feed valve will not in general cavitate. The low boiler feed valve, however, will always be operating in a cavitation regime. Both valves have been fitted with cavitation control trim so that each should be able to function with the flow cavitating inside the valve. Actual measurements have been obtained from the manufacturer (Ref. 5). Calculations indicated that this valve will operate between the pressure drop for incipient cavitation and the choked flow point at any positive subcooling. The ISA Handbook For Control Valves (Ref. 6) states that if a valve is operated below the "Km" point (Δp for choked flow) cavitation damage does not seriously affect the life expectancy of the valve. Therefore, it is expected that this will present no serious problems if there is a positive subcooling (no flashing in the pipe downstream of the valve).

It is felt that for conservative operation initial operation should require 2°F to 3°F of subcooling. At a later time, this margin may be reduced to more closely approach the choked condition. Reference 6 states that as the choked flow point is reached, the noise level at the valve increases substantially. The noise level can then be used to adjust the subcooling to less conservative levels during plant operation.

On the basic performance curves (Figures 2, 5, 8, and 11) inherent system limits are shown. The isobutane pump limit is a system limiting condition caused

by the flow rate and pressure interaction between the pump and the high pressure boilers and its turbine stage. Conditions which would increase the high boiler pressure (increased geothermal fluid temperature or flow rate) ultimately reach the condition where the pump cannot supply the flow which the boiler-turbine required, hence the limit. This limit occurs when the boiler level, it would be more expedient to operate with some controllable pressure drop across the valve. The line which represents 25 psi pressure drop across the valve is shown also.

Because of scaling, corrosion, and erosion, the geothermal fluid flow velocity in the tubes of the heat exchangers should be kept within certain limits. Figure 14 shows the velocity in the tubes of each geothermal heat exchanger as a function of fluid flow. It is noted that in the range from 80% to 120% of design flow, the maximum velocity is less than 7 ft./sec. and greater than 4.5 ft./sec. The velocities are judged to be within the limits set by scaling and corrosion constraints. To limit erosion of brass, the velocity is usually kept less than 7 ft./sec.

Raft River Plant-Field Performance Prediction

As examples of the use of the previous results, the performance of the plant was predicted for a number of cases representing the pumps and field as described in Reference 4. At the time that Reference 4 was written there has been no stimulation of existing wells. RRG1-7 data is represented by the original data. RRG2-5 data and status were not known at the time so that well was not included in the examples. Two cases were considered. The first case assumed that the wells as indicated in the reference. Under these conditions, the injection capacity is the limiting condition. The second case assumed that there was sufficient increase in injection capacity to approach a limiting condition in the plant (i.e., the pump limit as mentioned in the previous section).

The following assumptions were made:

1. Year round experiments on the site require 200 gpm of hot geofluid; in the summer an additional 180 gpm will be required for an irrigation experiment.
2. Production well pumps are those described by Jacoby in Appendix F of Reference 4 (the large pumps; RRFE-1, 10 stage Reda N-1500; RRGE-2, 17 stage Reda N-1500; RRFE-3, 12 stage Centerlift LA-600).
3. Injection pumps are those determined by Jacoby, Reference 4 (7 stage, Johnson 9DHC with a 700 psi injection limit on the wells).
4. Well drawdowns assume 85% operating factor.
5. Preheater bypasses produce at least 5°F subcooling entering each boiler.
6. Make up water requirements are calculated by the code.

Under these assumptions, the system is limited by the capacity of the injection wells. For the first case, the amount of geofluid produced was exactly the amount which could be injected plus the evaporation and irrigation requirements. The amount of produced fluid going to the plant was the total produced minus that going to the experiments. These results are summarized in Table 3. It is noted that for none of the conditions listed is the plant operating at the pump limit as discussed in the previous section. Lower production flow is required in the winter than in the summer because of lower make up (evaporation) for the plant cooling tower and no irrigation requirements. The best match is wells RRGE-1 and RRGE-2 in the summer and RRGE-1 and RRGE-3 in the winter. When flow was reduced to match the injection flow, the cooler well was throttled (consistent with production pump specification for minimum flow). The table shows the generator output, plant parasites, net plant power, production and injection pumping powers, and power to the grid are shown for summer, winter, 10%, 100% fouling, six months and five years of operation.

A separate set of conditions is presented in Table 4 assuming sufficient increases in injection capacity (at same pressure) to allow for enough plant flow to reach the inherent plant pump limit. It will be noted that although the power to the grid is not substantially different, the net plant power is increased in each case.

Conclusions and Next Planned Effort

The TAF digital simulation model of the 5 MW(e) Geothermal Pilot Plant has been updated, checked out and performance for various geothermal fluid temperature and flow rates. The constraints on the system have been analyzed and examples presented which relate the plant and geothermal field interaction with field data updated to June, 1979, data. The following conclusions are noted:

1. The digital simulation model has been checked out by comparison to independent data. The only component which is questionably modeled is the condenser, and the heat transfer coefficient of the condensing isobutane appear low. This model will be revised.
2. The turbine has a lower efficiency than that used in earlier analyses thereby giving lower generator output than originally estimated in Reference 3.
3. Curves have been generated to predict the system performance under summer and winter extreme ambient conditions for design fouling and 10% of design fouling (approximately clean performance).
4. Constraints on performance include:
 - a) an upper limit of geothermal fluid flow at a given temperature resulting in a maximum generator output) resulting from a high boiler pressure at the limit at which the isobutane feed pump can produce the required flows.
 - b) boiler inlet subcooling margins to insure no problem with cavitation and flashing in the supply valves.
 - c) geothermal flow rate limits to remain with the velocities required for scaling and corrosion minimization.

State Point	Temp. ($^{\circ}\text{F}$)		Flow Rate ($\text{lbm/hr} \times 10^{-5}$)	
	Ref. 3	Code	Ref. 3	Code
1	105	109	9.34	9.02
2	180	179	6.13	5.88
3	240	233	6.13	5.88
4	240	237	6.13	5.88
5	130	134	6.13	5.88
6	128	131	9.34	9.02
7	101	104	9.34	9.02
8	180	179	3.21	3.15
9	180	179	3.21	3.15
10	123	126	3.21	3.15
11	290	290	10.40	10.40
12	250	246	10.40	10.40
13	222	221	10.40	10.40
14	190	188	10.40	10.40
15	144	149	10.40	10.40
16	75	73	76.20	77.40
17	95	93	67.20	77.40

Figure 1 Nominal Design Conditions From Reference 3 and Computer Simulation

Figure 2.
Five MW (ca) Pilot Plant Performance - Summer - 10% Fouling

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18 X 25 CM

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Figure 3. Low Boiler Inlet Subcooling Margin - 10% Fouling - Summer Ambient

Design Geothermal Fluid mass flow rate - 1.04×10^6 lb/hr

Note: 1. For each percent of low pressure bypass, add 0.4 F to the number read from this curve.

2. High pressure bypass will increase the subcooling by 0.36 F for each percent of bypassed flow.

Figure 4 - High Boiler Inlet Subcooling Margin - 10% Forged - Summer Ambient

Figure 5
Five MWes Pilot Plant Performance - Winter - 10% Fouling

Design GeoFluid mass flow rate = 1.04×10^6 lb/hr

GeoFluid Temperature (F)

Figure 6 Low Boiler Inlet Subcooling Margin - 10% Focled - Winter Ambient

Note: 1. For each percent of Low Preheater Bypass add 0.4 F to the value read from this curve for subcooling margin with no high preheater bypass

2. High Preheater Bypass will increase the subcooling margin by 0.36 F for each percent of high preheater bypass

Design Geothermal Fluid mass flow rate = 1.04×10^6 lb/hr

Figure 7 High Boiler Inlet Subcooling Margin = 10% Forced - Water Ambient

Figure 8 Five MW (e) Pilot Plant Performance - Design Failing - Summer Ambient

Figure 9 - Low Boiler Inlet Subcooling Margin - Design Faulted - Summer Ambient

- Notes: 1) For Each percent of Low pressure bypass, add .07 F to the Subcooling from this graph.
2) High Pressure Bypass will increase the subcooling margin by 0.24 F per percent of bypass flow.

Figure 10 - High Boiler Inlet Subcooling Margin Design Factor - Summer Ambient

Figure 11-5 MW(e) Pilot Plant Performance - Design - Fouled - Water Ambient

Figure 12 - Low Boiler Inlet Subcooling Margin - Design Fluid - Water Ambient

- Notes: 1. For each percent of Low pressure bypass, add 0.09 F to the water need Formulae curve.
2. High pressure bypass will increase the subcooling margin by $\frac{F}{100}$ for each percent of bypassed flow.

Figure 1b - High Boiler Inlet Subcooling Margin Design Fouled - Water Ambient

Figure 15 Velocities in tubes of Heat Exchanger

TABLE 1

Difference Between Code Calculation and HTRI and
Manufacturer's Specification at Design Conditions

Heat Exchanger	Percentage* Difference Between Code and:					
	HTRI			Manufacturer's		
	tubeside h	shellside h	overall U	tubeside h	shellside h	overall U
Low Preheater	-4	+8	+3	-20	+34	+13
High Preheater	+10	-19	0	-23	+10	+9
Low Boiler	-2	+17	+3	-3	+15	+3
High Boiler	+3	+3	+1	-2	+9	+4
Condenser	-8 **	+10 **	+9 **	+1	-89	-38

*Positive percentage signifies code calculation greater than other value.

**New results are now available from HTRI indicating a shellside "h" which is close to the manufacturer's value. The code will be revised to account for this. The plant statepoint results will change slightly; turbine power will increase approximately 8%.

TABLE 2

Difference Between Code Calculation and HTRI Calculation
For Four Typical Cases at 10% Fouling For Each Heat Exchanger

Heat Exchanger	Percentage* Difference Between Code at HTRI:						
	tubeside h		shellside h		overall u		
	min	max	min	max	min	max	avg
Low Preheater	-18	-4	+2	+6	+2	+7	+4
High Preheater	-1	+6	-19	-19	-6	-2	-4
Low Boiler	-4	-2	+17	+21	+7	+8	+8
High Boiler	+2	+4	+10	+27	+4	+10	+6

*Positive number indicates code calculation greater than HTRI value.

TABLE 3 Predicted Plant-Field Interaction With Existing Field Condition (as outlined in Ref. 4)

	Prod. Wells (Flow in gpm)	Inj. Wells (Flow in gpm)	Geofluid Plant Inlet Condition		Generates Output (MW)	Plant Parasities (MW)	Net Plant Output (MW)	Prod. Pumping Power (MW)	Inj. Pumping Power (MW)	Power to Ext. Grid (MW)
			Temp. (F)	Flow rate (% design)						
1. Summer - 6 mo. 10% Fouled	#1(1562) #2(1120)	#6(1490) #7(560)	277	110	4.74	1.43	3.31	1.05	0.66	1.60
2. Summer - 5 yr. 10% Fouled	#1(1415) #2(1065)	#6(1400) #7(460)	277	102	4.52	1.42	3.10	1.05	0.63	1.42
3. Summer - 6 mo. 100% Fouled	#1(1562) #2(1120)	#6(1490) #7(560)	277	110	4.12	1.41	2.71	1.05	0.66	1.00
4. Summer - 5 yr. 100% Fouled	#1(1415) #2(1065)	#6(1400) #7(460)	277	102	3.93	1.40	2.53	1.05	0.63	0.85
5. Winter - 6 mo. 10% Fouled	#1(1600) #3(700)	#6(1490) #7(560)	280	94	6.01	1.43	4.58	0.68	0.66	3.24
6. Winter - 5 yr. 10% Fouling	#1(1590) #3(650)	#6(1400) #7(460)	280	84	5.58	1.40	4.18	0.68	0.63	2.87

TABLE 4 Predicted Plant Field Interaction With Injection Capacity Increased to Allow Plant To Operate At Plant Limit

	Prod. Wells (Flow in gpm)	Inj. Wells (Flow in gpm)	Geofluid Plant Inlet Condition		Generator Output (MW)	Plant Parasities (MW)	Net Plant Output (MW)	Prod. Pumping Power (MW)	Inj. Pumping Power (MW)	Power to Ext Grid (MW)	
			Temp. (F)	Flow rate (% design)							
7. Summer - 6 mo. 10% Fouled	#1(1170) #2(900) #3(660)	#6(1490) #7(608)	280	113	5.08	1.44	3.64	1.22	0.67	1.75	9% increase in Well #7 capacity
8. Summer - 5 yr. 10% Fouled	#1(1170) #2(910) #3(650)	#6(1340) #7(698)	280	113	5.08	1.44	3.64	1.22	0.74	1.68	52% increase in Well #7 capacity
9. Winter - 6 mo. 10% Fouled	#1(1170) #2(900) #3(660)	#6(1490) #7(951)	280	113	6.62	1.46	4.76	1.22	0.78	3.16	70% increase in Well #7 capacity
10. Winter - 5 yr. 10% Fouled	#1(1170) #2(910) #3(650)	#6(1340) #7(1041)	280	113	6.62	1.46	4.76	1.22	0.83	3.11	126% increase in Well #7 capacity

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Appendix A

DIGITAL SIMULATION OF THE 5 MW(e) PILOT GEOTHERMAL POWER PLANT

Summary

The static and dynamic behavior of the 5 MW(e) Pilot Geothermal Power Plant is predicted by the Plant Transient Model (Subroutine DERTRU) used in conjunction with the Time and Frequency (TAF) Simulation Code on the INEL CDC digital computer.

The model represents the physical components of the geothermal fluid, isobutane and cooling water loops in terms of mass and energy balances, thermodynamic and transport properties, flow resistances, convection heat transfer correlations, and valve and pump characteristics. Analog controls are incorporated in the model. The plant boundary conditions (geothermal fluid conditions at the plant inlet and ambient air conditions), operating parameters (heat exchanger fouling and isobutane preheater bypass), and valve positions may be represented in a static or dynamic fashion.

Time and Frequency (TAF) Digital Simulation Code

The Time and Frequency Digital Simulation Code (TAF) is an executive code which reads input data; finds steady-state and transient solutions, and frequency response; and lists output or plots it on microfilm for a system which is defined in a FORTRAN subroutine (DETURU) of the main program. It was developed at Los Alamos in 1966 to add capabilities for direct steady state and frequency response calculations to the transient calculation capability available in other simulation languages. The user codes two Fortran subroutines: (1) INPUT providing for input-output and (2) DERTRU providing the mathematical description of the user's problem in terms of tabulated functions, transfer functions and algebraic and first-order differential equations, which may be non-linear. (The current TAF code is of 100th order.) The code is run on the INEL CDC Computer, with batch entry. A plot package provides for microfilm plots of a maximum of 24 channels with a maximum of 5 channels per graph.

Plant Transient Model

A schematic diagram of the plant as modeled in SUBROUTINES DERTRU and INPUT of the TAF program is shown in Figure 1. The geothermal fluid, isobutane and cooling water loops are shown along with the major components simulated. On the diagram, dots refer to points at which energy balances are made (e.g., in a heat exchanger at each fluid and tube node), squares refer to mass balances (e.g., flow and storage of mass in isobutane side of boilers and condensers) and X is to points at which pressure is calculated from valve and pump characteristics and line hydraulic resistances.

The geothermal fluid loop at the right of the figure contains the boost pump and the heat exchanger passes. The recirculation line and control valves are incorporated to simulate the geothermal temperature control system. The isobutane loop is modeled with feed pump, heat exchangers, turbines, condenser and storage tank. The preheater bypass lines and turbine bypass lines are modeled. The cooling water loop consists of parallel pumps, condenser, cooling tower and sump along with make-up and blowdown connections.

The pumps have been modeled with typical pump characteristics. As actual pump characteristics are known, they are incorporated into the model. The turbines are simulated using manufacturer's values for flow coefficient as a function of pressure ratio and efficiency as a function of velocity ratio. The control valves have been updated to reflect vendor C_v data.

The cooling tower was modeled using Merkel's theory for cooling tower performance along with vendor supplied diffusion unit data. Each preheater is subdivided into seven sections. Each section consists of a

geothermal fluid node, a tube node and an isobutane node (each with its separate energy balance). Heat transfer from the geothermal fluid to the tube considers the convection from fluid to surface, the fouling resistance and the conduction through half of the tube. Similarly, on the isobutane side, half the tube conduction, fouling resistance and convection to the isobutane are considered. In the boilers and condenser where the isobutane undergoes a phase change, there is only one isobutane node. There are three tube nodes and three geothermal fluid (or cooling water) nodes in each boiler and condenser. The convection heat transfer coefficients are determined from standard correlations. The predicted overall heat transfer coefficients compare favorably with manufacturer's specifications at the design point and with predictions using a proprietary code developed by Heat Transfer Research, Inc. (HTRI) at other conditions.

Code Input and Output

The static (steady-state) solution of the plant model with the TAF code involves an iterative procedure. Starting with initial guesses of the solution variables along with a given set of boundary conditions, the code iterates until the solution variables converge to the steady state solution. The set of boundary conditions specified by the user would include inlet geothermal fluid temperature and flow rate, ambient air wet and dry bulb temperatures, heat exchanger fouling and the setting of trim-type valves within the system (e.g., settings for the preheater bypass and turbine throttle valves).

A baseline data deck has been generated to produce the original design steady-state conditions: nominal heat exchanger fouling, geothermal fluid inlet conditions, 290°F and 1.04×10^6 lb/hr and 1% summer ambient condition, 92°F dry bulb at 65°F wet bulb. This solution is used as the initial guess for the first solution with changed boundary conditions. Subsequent changes in boundary conditions use the previous solution as the initial guess. The output is a matrix of all important (approximate)

mately 800) parameters. Values are labeled with their FORTRAN names used in DERTRU.

Transient solutions are obtained with different control cards. A typical transient starts from a calculated set of steady state conditions. The transient (valve movement or temperature transient) is programmed into SUBROUTINE DERTRU. Any number of variables may be printed out at given time increments, and a plotting capability is available for these same variables.

Figure 1. Schematic Diagram of Plant Transient Model

Appendix B

Example of Use of Performance Curves

For plant operation with geothermal fluid at 280.5°F at summer extreme ambient conditions with 10% of design fouling determine the geothermal fluid flow rate, generator output, net plant effectiveness, and preheater bypass flows to maintain 5°F subcooling entering each boiler.

Solution:

- Figure 2 gives the base-line (no bypass) values for given input conditions.

For $T_{\text{geo}} = 280.5^{\circ}$ at the pump limit

Geothermal Fluid Flow Rate = 113% design

Generator Output = 5.08 MW

Net Plant Effectiveness = 3.12 w hr/lb geofluid

The Design Geothermal Fluid Flow Rate = $1.04 \times 10^6 \text{ lb/hr}$

$m_{\text{geo}} = 1.13 (1.04 \times 10^6 \text{ lb/hr approximately } 2540 \text{ gpm})$

- Figure 3 allows the computation of the bypass fraction around the low preheater.

For $T_{\text{geo}} = 280.5^{\circ}\text{F}$ $mgw = 113\%$

At: no bypass Subcooling entry low boiler = 1.25°F

10% bypass = 7.05°F

Interpolation for subcooling of 5°F given 6.4% low preheater bypass

- Figure 4 give high boiler subcooling with no high preheater bypass. From the curve with no bypass the subcooling is 4.6°F , but note 1 indicates additional subcooling caused by the low preheater bypass of $0.4^{\circ}\text{F}/\%$ bypass.

High boiler subcooling = $4.6 + 0.4(6.4) = 7.1^{\circ}\text{F}$ which is greater than 5°F , therefore, no high preheater bypass is necessary.

- Corrections for the bypass. The generator power (and net power) are decreased 0.17% for each percent of low preheater bypass;

Power correction = $0.0017 (6.4) (5.08) = 0.06 \text{ MW}$

Generator output = $5.08 - 0.06 = 5.02 \text{ MW}$

Net plant effectiveness = $3.12 \text{ w hr/lb} - \frac{0.06 \times 10^6 \text{ w}}{1.175 \times 10^6 \text{ lb/hr}}$

= 3.07 w hr/lb