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PERFORMANCE OF A 5MW(e) BINARY GEOTHERMAL-ELECTRIC POWER PLANT

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Abstract

A 5MW(e) Pilot Geothermal Power Plant was built by the Idaho National Engineering Laboratory (INEL), at Raft River, Idaho, as an integral part of the Department of Energy's plan for commercial development of geothermal energy. The purpose of the plant was to investigate the technical feasibility of utilizing a moderate temperature hydrothermal resource (275 to 300°F) to generate electrical power in an environmentally acceptable manner. The plant used a dual-boiling binary cycle with isobutane as the working fluid, and drew thermal energy from a 280°F liquid-dominated resource. This paper presents the results of that testing, and compares both component and system performance to the performance predicted prior to operation.

INTRODUCTION

Work on geothermal programs at the Idaho National Engineering Laboratory (INEL) has focused on using low- and moderate-temperature hydrothermal resources. A major portion of the work was the design, construction, and operation of a binary-cycle pilot power plant with a nominal gross rating of 5MW(e), located in the Raft River Valley of Southern Idaho. Figure 1 shows the location of the plant. RRGE-1, 2, and 3 represent the production wells used, and RRGI-6 and 7, the injection wells used for the plant.

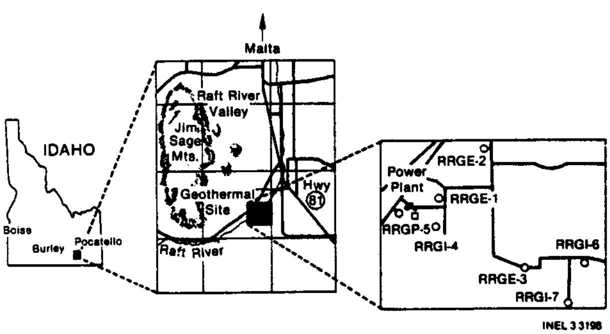

Figure 1. Location of the Raft River 5MW(e) Pilot Geothermal Power Plant

The purpose of building this plant was to gain operational experience and demonstrate the technical feasibility of generating electric power from a moderate-temperature (275-300°F) dual-boiling power cycle in an environmentally acceptable manner using isobutane as the working fluid and using state-of-the-art components. The information and general operational experience would be applicable to any binary cycle plant including geothermal, solar, and waste heat bottoming cycles. The plant was designed to take maximum advantage of the low ambient temperatures occurring in the Intermountain region by operating in a floating power mode, thereby enabling the plant to produce more power in the winter months than at the summer design condition. It was also designed to use treated geothermal was for plant heat rejection in the wet cooling towers to gain experience for geothermal plants located in environments where water is scarce.

When the project was conceived, the plant was to be run for a five-year period of testing and operational evaluation. References 1 and 2 describe the test plan in detail. When the Department of Energy (DOE) shifted its goals from demonstration projects to more basic research, plant operations were first cut back to two years and later to a start-up and shake-down run in the fall of 1981, continued shakedown and a sequence of performance tests in the spring of 1982, and a final shutdown June 15, 1982. Reference 8 gives a more detailed description of the plant, performance analysis, and operational experience.

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POWER CYCLE SELECTION AND DESCRIPTION

A variety of working fluids and cycles were initially studied for this moderate temperature resource application. It was found that the dual-boiling cycle had a significantly better performance than either the single boiling cycle or the supercritical cycle with isobutane working fluid when the resource temperature was below 300°F. Figure 2 shows a simplified schematic diagram of the plant including state point numbers. In this figure, the three primary systems are shown, but with bypass, recirculation, makeup, blowdown, vent, and fill lines omitted.

Based on a 290°F liquid geothermal resource at the plant, a design base case was established;

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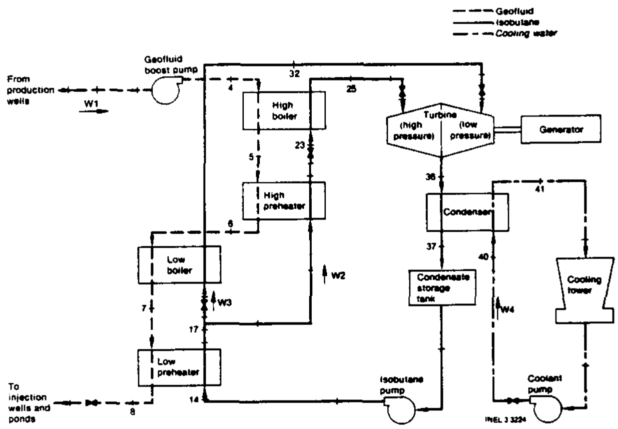

Figure 2. Schematic Diagram of the Plant

Tables 1 and 2 give the nominal state point and flow values and a heat-power balance for the design ambient condition (65°F wet bulb temperature). Experimental results are also shown in these tables.

The pressure of the geofluid entering the plant was increased using a geothermal boost pump to account for the pressure losses within the plant as the geofluid flowed through the heat exchangers and associated piping and valves. The geofluid flowed in series through the high pressure boiler, the high temperature preheater, the low pressure boiler and the low temperature preheater.

In the isobutane loop, slightly subcooled liquid was taken from the condensate storage tank and pumped to the pressure of the high pressure boiler. The entire isobutane flow passed through the low temperature preheater exiting at around 180°F. At this point the flow was split; approximately two-thirds went through the high temperature preheater and the high pressure boiler, and the other third went through the low pressure boiler after passing through a control valve which decreased its pressure to the proper magnitude. This control valve operated to maintain the liquid level in the boiler. The high temperature preheater heated the liquid isobutane to approximately 240°F. The liquid was vaporized in the high pressure boiler and the vapor flowed to the high pressure turbine wheel. Similarly, the liquid vaporized in the low pressure boiler flowed to the low pressure turbine wheel. No effort was made to recover the available energy lost by throttling the liquid flow into the low pressure boiler. The two vapor streams mixed within the turbine casing before they went to the condenser. In the condenser the condensed vapor was slightly subcooled before it was returned to the condensate storage tank.

The cooling water which received the energy given up by the condensing isobutane vapor flowed through the condenser with approximately a 20°F temperature rise. The cooling water then flowed

Table 1. Flow and State Point Data

Design			Baseline Run (Test 1A)		
Geofluid	Isobutane	Cooling Water	Geofluid	Isobutane	Cooling Water
Mass Flow Rates (lbm/hr)					
$W1 = 1.04 \times 10^6$	$W2 = 6.13 \times 10^5$ $W3 = 3.21 \times 10^5$	$W4 = 7.53 \times 10^6$	$W1 = 1.00 \times 10^6$	$W2 = 5.37 \times 10^6$ $W3 = 3.36 \times 10^5$	$W4 = 5.94 \times 10^6$
Temperatures °F (saturation pressure, psia)					
4 290	14 105	40 75	4 279	14 91	41 57
5 250	17 180	41 95	5 247	17 168	42 78
6 222	23 240		6 215	23 443	
7 190	25 240 (382)		7 185	25 236 (373)	
8 144	32 180 (203)		8 146	32 178 (202)	
	36 128			36 102	
	37 101 (78)			37 88 (61)	

Note: For Design Case: wet bulb temperature was 65°F
For Baseline Case: wet bulb temperature was 36°F

Table 2. Power Balances

Power Balance in Megawatts	Design	Baseline Run (Test 1A)
Heat Addition		
Low temperature preheater	14.0	11.7
Low pressure boiler	10.0	8.8
High temperature preheater	8.5	9.8
High pressure boiler	12.5	10.0
TOTAL	45.0	40.3
Heat Rejection		
Condenser	40.7	36.9
Turbine Power	5.0	4.0
Parasitic Power		
Feed pump	0.7	0.6
Cooling tower fan and pump	0.6	0.5
Geofluid boost pump	0.1	0.1
TOTAL	1.4	1.2
Net Plant Power	3.6	2.8
Production - Well Pumps	0.8	0.8
Injection - Well Pumps	0.4	0.4
NET POWER	2.4	1.6

through a wet cooling tower in which the energy was rejected to the atmosphere. Treated geothermal water was used for cooling water make-up.

COMPONENT DESCRIPTIONS

Pumps

The working fluid pumping was provided by two parallel vertical turbine pumps at 1514 ft and 1747 gpm each. Each pump had six stages and a 500 hp motor. The pump efficiency at rated conditions was specified at 78 percent. The pumps were sized for the minimum condenser pressure of 42 psia.

The geothermal boost pumps provided the head required to pump the geofluid through the heat exchangers and through the transmission lines to the injection pumps. Two parallel, vertical-split case centrifugal pumps (each with a head of 272 ft at a flow of 1115 gpm, a design efficiency of 80.5 percent, and driven by a 125 hp electric motor) provided this capability.

The pumping required to move the cooling water through the condenser and cooling tower was provided by two parallel vertical turbine pumps. At rated conditions each pump provided 7700 gpm of water at 125 ft head. At these conditions the efficiency was specified as 83 percent. Each pump was driven by a 300 hp motor.

Heat Exchangers

The heat exchanger characteristics are summarized in the following table:

Heat Exchanger	Surface Area (ft^2)	Length (ft)	Diameter (in)	Weight (T)
Low temperature preheater	30,039[a]	49	50	43
Low pressure boiler	5,938	42	33/68	20
High temperature preheater	15,059[a]	50	35	22
High pressure boiler	5,938	42	33/68	20
Condenser	59,996	50	88	140

[a]Extended surface.

The tube material for all geothermal fluid heat exchangers was admiralty brass. The tube sheets were aluminum bronze clad carbon steel. The geothermal side fouling factor was assumed to be 0.0015 hr ft^2 F/Btu, and 0.0005 hr ft^2 F/Btu was used on the isobutane side. The condenser was made of carbon steel throughout, including the tubes. For design of the condenser, the cooling water side fouling factor was taken as 0.0010 hr ft^2 F/Btu, and an isobutane side fouling factor of 0.0005 hr ft^2 F/Btu was used.

Cooling Tower

The cooling tower was a crossflow, two-cell, mechanical draft, wet unit. Each of the 40 by 70-ft cells was equipped with a fan which had an 80 hp motor. The tower was 53 ft high and was constructed of treated Douglas fir and redwood.

Turbine-Generator

The turbine utilized the barrel design. This design was easy to seal for high-pressure service, and facilitates disassembly and reassembly for maintenance. The rotor had two radial inflow wheels, and operated at 8000 rpm. Because the flows from the low and high pressure inlets were combined to a common outlet, the aerodynamic thrust load was low.

The generator was rated at 7200kW, 7579 kVA, 1200 rpm synchronous speed, and electrical conditions of three-phase, 60 Hz and 4160 V. The generator design power factor was 0.9.

Supply and Injection System

Geofluid was supplied to the operating plant from three production wells, RRGE-1, 2, and 3. The spent geofluid was reinjected into wells RRGI-6 and 7. All of the lines in the supply and injection system were made of cement-asbestos pipe with transition to steel pipe at the wells, at the plant, and at a manifold into which the individual production-well pipelines joined. The pipe was buried to a depth of about 2-1/2 ft. The supply lines were insulated with urethane foam to limit the temperature drop to less than 1.5°F per mile. Figure 1 shows the location of the wells relative to the plant. The pipeline for the production wells to the plant covered about one mile in length, and the line from the plant to the injection wells was about 1.8 miles.

Line-shaft pumps were installed in each production well. At each injection well, the line dumped into a pond, and then the geofluid was pumped from the pond and injected with individual pumps.

PERFORMANCE ANALYSIS

The plant was tested over a period of three months. The tests consisted primarily of varying the geothermal inlet and cooling water conditions to determine system performance.[1,2] In addition to the system performance, the behavior of the individual components was investigated. The changes in input conditions allowed for a wide range of operating conditions for the individual components.

The geothermal fluid and cooling water properties were taken to be those of pure water because the concentration of impurities in both of these systems was sufficiently low. The 1967 ASME Steam Tables[3] was the source of the thermodynamic properties of water. The isobutane properties were obtained from Reference 4 which uses Starling's modification of the Benedict-Webb-Rubin equation of state. The viscosities and thermal conductivities needed in the heat exchanger analyses were obtained from a computer program developed by Ely and Hanley of the National Bureau of Standards. This program used a variation of the law of corresponding states and methane properties, and is described in Reference 5.

Component Performance

Pumps. The data from the 17 different tests indicated some deficiencies in the performance of the pumps. The isobutane feed pumps produced a head rise approximately five to six percent lower than the manufacturer's test curves indicated for a given flow. This was a critical deviation because a higher than expected pressure drop was found to exist in the piping between the pump and the high pressure boiler. The result was the inability to supply the boiler with the desired amount of isobutane at the rated geofluid flow; the impact will be discussed under System Performance.

The geofluid boost pump operated as specified, but the cooling water pumps were able to supply only 78 percent of the rated cooling water flow. This caused a large reduction in power produced by the plant. The reason for the poor performance of these pumps was found to be improper installation. The pump pit in which the cooling water pumps operated was found to be too shallow to accommodate the complete pump inlet. The inlets were shortened and strainers reduced in size and placed on the bottom of the pit. The pumps were installed at an inappropriate distance from the back wall and appreciable vortexing was noted. It is felt that if the pumps had been installed correctly, no flow reduction would have resulted.

Turbine-Generator. The measured performance of the turbine-generator was approximately as expected from the manufacturer's predictions, but one fact complicated the performance assessment. The boilers were entraining and carrying over some liquid and even after passing through the turbine throttle valve liquid entered the turbine in many cases. When this was accounted for as a penalty on the expected turbine efficiency, the measured performance agreed quite well with prediction.

One deviation which was noted during testing was the fact that each turbine stage passed a somewhat larger flow at a particular inlet pressure than was expected. Calculations indicated that the nozzle throat area for the high pressure stage was approximately three percent larger than specified and that the low pressure nozzle was approximately 10 percent larger than specified. These areas would have been corrected had a longer period of time been available for testing.

Cooling Tower. Measurements on the cooling water leaving the cooling tower indicated that when the tower fans were operated at full speed, the temperature was within 2 to 3°F of the manufacturer's predicted value. The temperature was always higher than predicted, however. Because of problems with the cooling water treatment facility, the fans were not run on the high speed for many of the operating conditions, resulting in an increased condensing temperature and reduced turbine power for those tests.

Heat Exchangers. The performance of each heat exchanger for the various test conditions was evaluated by use of the proprietary Heat Transfer Research, Inc. (HTRI) computer codes. The members of HTRI are heat exchanger manufacturers, architect-engineering firms and heat exchanger users in industry. Hence, the benefits of comparing the test data with performance evaluation by using HTRI codes and other information will benefit the heat exhanger industry and utlimately the future design of geothermal heat exchangers. These codes consisted of shell-and-tube, condenser, and boiler programs used in a rating mode for this study. The approach used was to input the measured fluid temperatures and flow rates, and then to vary the required fouling resistance so that the heat duty matched that from the test data.

The results of this evaluation are shown in Figure 3. The performance of each heat exchanger is shown as the difference in overall resistance from that predicted by the manufacturer, expressed as a fraction of the total design resistance. Note that a positive value of performance decrease is equivalent to operation at a condition with more thermal resistance than design.

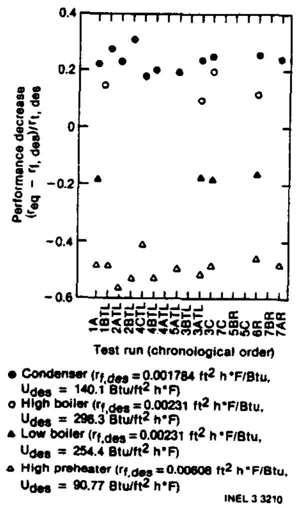

Figure 3. Heat Exchanger Performance

The low temperature preheater results are not shown on this plot because the HTRI codes were unable to predict the performance. That preheater was somewhat overdesigned for this application, resulting in an approach temperature for the two streams well within the experimental uncertainty of the measurements. This also resulted in an extremely high thermal leakage across the longitudinal baffle contributed to the closeness of the approach. The combination of these effects made the analysis impracticable using the HTRI codes.

The tests are listed on Figure 3 in chronological order. Generally, there was no apparent trend to indicate increase in fouling as the time proceeded. Additional details concerning the heat exchangers are given in Reference 8 and are the subject of a paper which is presently being prepared.

System Performance

State Point Data. Experimental data taken during the test were used to calculate thermodynamic properties at state points throughout the system for each test. Test 1A was taken as the baseline case for the system. The geofluid temperature was 10°F lower than the design temperature resulting in a decrease in output power of approximately 500kW. This was, however, the highest temperature obtained during the testing period. A summary of the reduced state point data of Test 1A is presented in Table 1; the mass flow rates and energy balances for the boilers, heat exchangers, and condenser are shown in Table 2. The state points correspond to points in the system as indicated in Figure 1. These are the best estimates of the cycle state point data for the test which was nearest the design point. The test that produced the maximum power was not used because the liquid levels in both the high- and low-pressure boiler were so high that if was not possible to estimate the amount of moisture that was being carried from the boilers.

Generally, mass and energy balances were good. The greatest difference in the calculated parameters was between the calculated heat transfer rate from the geofluid compared with that into the isobutane working fluid in the high pressure boiler. The calculated heat from the geofluid was 8.5 percent lower than that calculated to be transferred to the isobutane. For the 17 tests which were examined, the calculated heat transfer rate from the geofluid was consistently lower than the calculated rate into the isobutane, averaging 10.4 percent lower for the turbine powered cases and 9.9 lower for the thermal loop (without turbine) cases. This difference is attributed to some sensor errors or the possibility of isobutane leakage through return lines to the condensate storage tank.

In addition to the test data shown in the paper, tests which showed the effects of geofluid and cooling water flow and temperature changes on system power output were conducted. Reference 8 discusses these tests in detail.

Availability-Irreversibility Analysis. The ideas associated with an availability-irreversibility analysis allow the performance of the system to be considered in the perspective of the thermodynamic ideal and assess the losses in thermodynamic performance attributable to the individual components. Figure 4 presents the results of such a study on the baseline case (Test 1A). If the plant itself is considered to be this system of interest, there are a number of things external to the system that are affected by it. The geofluid leaving the plant has a lower thermodynamic availability than that entering the plant, creating a decrease in availability of things external to the plant. The cooling water increases in availability as it flows through the plant condenser. These processes create increases in availability external to the plant. (The remainder of the cooling water loop (pumps and cooling tower) were not included in the system because the state points in the cooling tower were not known with

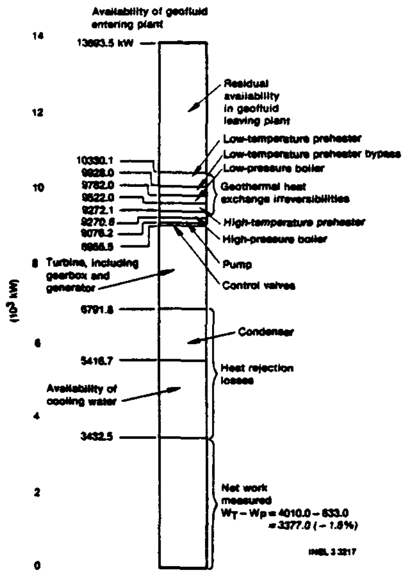

Figure 4. Availability Analysis

sufficient accuracy.) The algebraic sum of all of the changes in availability external to the system is equal to the sum of the irreversibilities of the components within the system. The irreversibilities of each of the components within the system were calculated separately along with the availability of each flow into or out of the system. The dead (atmospheric) state was taken as the wet bulb temperature, 35°F, and atmospheric pressure, 12.5 psia.

Table 2 shows the other parasitic power requirements of the plant. If these power requirements were subtracted from the net plant power of 3.4MW (Figure 4) from the availability analysis, the net power produced during Test 1A would have been 1.6MW. This number may be abnormally low because the power expended in the geothermal supply and injection system was relatively high. The supply and injection system was not designed for the purpose of supplying the plant only and expends more power than a properly designed and matched system. Therefore, the more typical value to consider is that for the plant without the supply and injection system. For Test 1A the plant produced 2.9MW exclusive of any supply and injection system parasitic power losses.

Plant Output with Major Problems Corrected

The deviations from design of the plant component performance and system operability have been noted earlier. The effect of correcting these deficiencies is illustrated by considering their effects the baseline run from the performance test series. Table 3 indicates the power for the baseline case with the major deficiencies corrected. Note that pretest estimates of the plant power with

Table 3. Baseline Performance of System with Major Deficiencies Corrected

	Power (kW) (% of possible power)
Generator output	4010
Increment in power caused by defect	
1. Failure to utilize design geofluid flow	110 (2)
2. Moisture in turbine	144 (3)
3. Cooling water pumps not able to produce specified flow	380 (7)
4. Cooling tower unable to produce specified cold water temperature	454 (9)
5. Other components including heat exchangers, turbine-generator	125 (2)
POWER POSSIBLE WITHOUT DEFECTS	5224 (100)

design fouling, design flows, 278°F inlet geofluid and 35°F wet bulb temperature were 5347kW, as compared to the 5224kW for the "corrected" baseline test performance. Had the component performance deficiencies been corrected to design specifications, the plant would have performed generally as predicted.

CONCLUSIONS AND RECOMMENDATIONS

The following summarizes the primary conclusions of the plant performance tests and makes recommendations concerning design of a new plant.

1. The performance of the system, when corrected for the component performance values which were below specifications, was approximately as predicted.

2. The system sensitivity to changes in geofluid inlet temperature and flow rate was essentially that calculated prior to operation of the plant.

3. Thermodynamic and transport properties used appear to be adequate in describing the performance of components and system.

4. The HTRI computer codes appear generally to be adequate in determining overall performance of the heat exchangers. Some small problems were noted but they did not change overall conclusions.

The following recommendations are made after the experience with this plant.

1. In design of a new facility, the feed pump should be designed with a comfortable margin because it may need to overcome higher than design pressure drops, and any excess pressure can be handled by a control valve.

2. Proper design and execution of the heat rejection system is mandatory. This is where the greater share of the loss in power production from design occurred in this plant.

3. Care should be taken to ensure that any liquid entrained in boiler vapor flow is separated prior to removal of the vapor from the boiler.

4. The initial predictive methods were successful and could be used for any type of binary power cycle.

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