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IDAHO NATIONAL ENGINEERING LABORATORY

ANALYSIS OF THE BINARY
CYCLE FOR GEOTHERMAL POWER GENERATION

PREPARED FOR THE
ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
IDAHO OPERATIONS OFFICE UNDER CONTRACT E (10-1)-1375

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Geothermal Energy

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ANALYSIS OF THE BINARY
CYCLE FOR GEOTHERMAL POWER GENERATION

W. W. Madsen

I. J. Ingvarsson

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ABSTRACT

Exchanging heat from hot geothermal fluid to a secondary working fluid which is then expanded through a turbine appears to be the best way of producing electrical power from a medium temperature geothermal resource. A study has been made to determine the conceptual design state points for a dual boiling system which utilizes isobutane as the working fluid. This paper presents the results of this work and the sensitivity of plant performance to variations from nominal design point. In addition, variations due to tolerances applied to thermodynamic properties and other key factors are included.

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TABLE OF CONTENTS

Summary	1
Results	2
1.0 Development of the Design Case	4
1.1 Boiler Temperature	4
1.2 Condenser Temperature	6
1.3 Flow Split	7
1.4 Superheaters	8
2.0 Assumptions for the Determination of the Design Case for the Geothermal Binary Cycle	9
3.0 Sensitivity Study	11
3.1 Brine Inlet Temperature	11
3.2 Condenser Temperature	14
3.3 Pinch Point	16
3.4 Flow Ratio	16
3.5 Amount of Superheat on the LP Leg	19
3.6 Mechanical Efficiencies	22
4.0 Tolerance Study	24
4.1 Turbine Entropies	24
4.2 Specific Heats	24
4.3 Mechanical Efficiencies	25
4.4 Overall Heat Transfer Coefficients for the Heat Exchangers (U's) and Brine Inlet Temperature	25
4.5 Discussion	29
5.0 Off-Design Operational State Points	33
5.1 The Effects of Variation in Brine Inlet Temperature on Plant Parameters with Constant Power Output	33
5.3 The Effects of Variation in Brine Side Fouling Factor and Coolant Water Temperature	37

6.0 Conclusions and Recommendations	42
7.0 References	44

Appendix A - Calculation of the Dual Boiling Cycle

Appendix B - Geothermal Program Listing

Figures

1-1	Binary Cycle Geothermal Power Plant Systems Analysis Model	3
1.1-1	Isobutane Dual Boiling Cycle	5
3.1-1	Design Case Sensitivity to Brine Inlet Temperature Variations	12
3.2-1	Design Case Sensitivity to Variations in Isobutane Condensing Temperature	13
3.3-1	" " " " Pinch Point Variations	15
3.4-1	" " " " Flow Split Variations	17
3.4-2	" " " " Variations in Flow Split	18
3.5-1	" " " " " " Superheat on the Low Pressure Boiler	20
3.5-2	" " " " " " " " " " " " " " " "	21
3.6-1	" " " " " " Pump Efficiency	23
3.6-2	" " " " " " Turbine Efficiency	23
3.6-3	" " " " " " Electrical Generator Efficiency	23
5.1-1	Binary Cycle Geothermal Power Plant Systems Analysis Model	31
5.1-2	" " " " " " " " " "	32
5.2-1	" " " " " " " " "	34
5.2-2	" " " " " " " " "	35
5.3-1	" " " " " " " " "	38
5.3-2	" " " " " " " " "	39
5.3-3	" " " " " " " " "	40
6-1	" " " " " " " " "	41

Tables

I. Overall Heat Transfer Coefficients Used for Heat Exchanger Calculations	10
II. Base Case Tolerances	26

Summary

A series of analyses was conducted to determine the preliminary configuration and nominal design parameters for a geothermal binary system, for a 10 MW(e) plant operating from the Raft River Idaho Geothermal Source. In this context, a binary system is one with the power cycle in a secondary loop, employing a second fluid as the working fluid. Detailed studies were made on the following critical parameters:

1. Number of boilers
2. Flow split to the double boilers
3. Pinch points of the various boilers
4. Superheater desirability and requirements
5. Regenerator desirability and requirements
6. Temperature of the various boilers
7. Condensing temperature and therefore, turbine back pressure
8. Brine inlet temperature variations on cycle performance.

A simplified computer model of the geothermal system capable of running superheaters, regenerators, turbines, pumps, preheaters, and multiple boilers was developed (see Appendix). With this computer program, a design case was developed, with two boilers and two preheaters, and with no superheaters or regenerators.

Sensitivity studies were run on the design case to determine the effect of individual parameters on plant operation. The parameters considered were brine inlet temperature, condenser temperature, pinch point, flow split, the amount of superheat added to the LP boiler, and mechanical efficiencies. A tolerance study was made for the design case to determine the effects of uncertainties upon plant performance. Finally, off-design operational state points were examined. These encompassed possible best and worst case operating conditions that could occur during plant life.

Results

From previous studies on working fluids,^(1,2) it was decided that isobutane would be used for the working fluid. These studies included a supercritical cycle, a boiling cycle, and a boiling-superheat cycle, utilizing other fluids as well as isobutane. It was found that the pumping power required to raise the working fluid to the pressure necessary for the supercritical cycle penalized the system too severely. Isobutane was chosen because the isobutane boiling cycle was the most efficient and further studies were performed to determine the number of boilers, the temperatures of the boilers, the desirability of using superheaters or regenerators, overall plant size and general development of a preliminary design.

Based upon the results of the studies, a two boiler system was developed. The entire isobutane flow is initially preheated, then split into two flow paths. Approximately four-tenths of the flow is directed to the low pressure boiler (160°F) and is then run through the low pressure turbine. The remainder of the flow is passed through a second preheater raising the isobutane temperature to 230°F, or the temperature of the high pressure boiler. From there, it is run through the high pressure turbine. The two turbine exhaust streams are then mixed and condensed.

The plant is designed to produce 10MW(e), with brine utilization (effectiveness) of 3.93 watts produced pound of brine circulated for 2.5×10^6 lbm/hr of brine. The total isobutane flow is 2.7×10^6 lbm/hr and coolant water flow is 1.9×10^7 lbm/hr from the cooling towers. The combined turbine shaft output is 15.05MW, with parasitic losses as follows: electrical losses, .75MW, brine pumping power, 1.43MW, isobutane pumping power, 1.40MW, and coolant water pumping power, 1.47MW.

BINARY CYCLE GEOTHERMAL POWER PLANT SYSTEMS ANALYSIS MODEL

ANC-A-5398

STATE POINT	TEMPERATURE (°F)	PRESSURE (psia)	FLOW RATE (lbm/hr)
1	105	396	2.7×10^6
2	160	368	1.9×10^6
3	230	354	1.9×10^6
4	230	347	1.9×10^6
5	128	80	1.9×10^6
6	120	80	2.7×10^6
7	101	77	2.7×10^6
8	160	169	7.3×10^5
9	160	162	7.3×10^5
10	116	80	7.3×10^5
11	300	238	2.5×10^6
12	243	213	2.5×10^6
13	202	201	2.5×10^6
14	170	188	2.5×10^6
15	130	138	2.5×10^6
16	75	22	1.9×10^7
17	95	12	1.9×10^7

NET POWER = 10MW (e)

TURBINE OUTPUT = 15.05MW

PLANT POWER REQUIREMENTS = 5.05MW

Brine Pump Power = 1.43 MW*

Isobutane Pump Power = 1.40MW

Coolant Water Pump Power = 1.47MW

ELECTRICAL LOSSES = 0.75MW

EFFECTIVENESS = 13.43 BTU/LB of BRINE

*Includes Brine Supply & Reinjection Pumps

Figure 1-1

1.0 Development of the Design Case

The design case state point is shown in Figure 1-1. The two-boiler system was chosen to help alleviate the pinch point problem. For a one-boiler system, the pinch point (which was assumed to be 10°F in all cases) controlled the amount of brine flow required, which resulted in rejecting brine flow (out of the preheaters) at approximately $150\text{--}160^{\circ}\text{F}$. This represents wasted energy which can be utilized by the second boiler.

As more boilers are added, another cycle benefit is noted: The system becomes less sensitive to brine inlet conditions. Since brine is rejected at $120\text{--}130^{\circ}\text{F}$ with the double boiler system, a third boiler was tried in an effort to utilize more brine energy. However, the increase in system effectiveness (Btu's produced per pound of brine) was marginal compared to the corresponding increase in the total system heat transfer area.

1.1 Boiler Temperature

For the two-boiler design, the system will optimize at basically two points. At the first optimization point, the boilers operate at the relatively high temperatures of approximately 270° and 215°F . A superheater is required on the 270°F boiler to insure that the turbine does not expand into the two-phase region. This superheater pulls energy from the brine and, therefore, extenuates the problem of the pinch point in the boilers for a given plant output. The pinch point in the 270°F boiler controls the brine flow rate by defining the brine temperature at the boiler exit. Additional brine flow is therefore required to accommodate the superheater. The 215°F boiler does not require a large amount of brine energy (compared to the higher temperature leg) and therefore, brine is exhausted at higher temperatures, reducing brine utilization. These high temperature

Figure 1.1-1 Isobutane Dual
Boiling Cycle

boilers do have an advantage in that the latent heat of evaporation is small at the higher temperature and, therefore, the boilers will be smaller. However, the preheaters tend to be larger and the overall system pressure is higher.

The second optimization point occurs at lower boiling temperatures, namely 230°F for the first boiler and 160°F for the second boiler. The reason why the 230°F boiler is desirable can be seen from the pressure-enthalpy diagram for isobutane (see Figure 1.1-1). At 230°F, the saturated vapor is at the highest enthalpy point, and expansion through the turbine to any given low pressure state will give maximum power output per pound of flow without expanding through the two-phase portion of the dome. Turbine efficiency can, therefore, be quite high and turbine design is facilitated. The pinch point of 10°F occurs in the second boiler and, therefore, the boiler temperature has a primary effect on the brine exit temperature. Since maximum utilization of the brine energy is desirable, a low boiler temperature is also desirable to reduce brine exit temperatures, but low boiler temperatures yield lower turbine inlet temperatures and pressures and thus, less specific power output is obtained from the low pressure turbine. When the second boiler temperature was varied, the optimum temperature was found to be near 160°F. This lower temperature optimization point yields several advantages. First, it gives better utilization of the brine than the high temperature optimization point. Second, it reduces the size of the preheaters, which are large. Third, it reduces the system overall pressure. Fourth, it makes the overall system much less sensitive to fluctuations of brine supply temperatures.

1.2 Condenser Temperature

The temperature, and therefore pressure, of the condenser has large effects upon system performance as well as on heat transfer area. System performance

is affected primarily due to the back pressure effects on the turbines resulting from an increase or decrease of condensing pressure. Although turbine output is maximized with low condensing pressures and, therefore, condensing temperatures, the size of the condenser grows dramatically. This problem is even more prominent with the utilization of a cooling tower since the cooling water supply temperature can be no lower than approximately 75°F to assure plant output will not drop below 10 MW for design conditions selected. In order to maintain reasonable condenser area sizes and condensing water flow rates, a minimum water temperature difference of 20°F was specified across the condenser. This means the lowest practical condensing temperature of the isobutane is approximately 105°F (10° condenser pinch point assumed).

1.3 Flow Split

In order to determine the optimum flow split between the high and low pressure boiler, a series of cases were run varying the flow ratio (the ratio of flow through the low pressure boiler to the flow through the high pressure boiler) from 0.1 to 3. See Section 3.4 for a more detailed discussion of the sensitivity analysis. It would appear that for maximum isobutane utilization, the maximum amount of flow should be channeled to the high pressure boiler since its higher operating temperature gives a turbine which produces more Btu's per lb of flow. However, as more isobutane flow is run through the high pressure boiler, more brine energy is required to boil it and the pinch point limitations of the low pressure boiler are made more severe. The performance of the cycle at a flow split of 0.1 is very low due to these pinch point problems, but as the flow split increases to approximately 0.35 to 0.4, the overall cycle efficiency is doubled. As the flow split increases from this value, the curve tends to be fairly flat with a slight decrease in overall performance as flow ratios increase. When the flow ratios become very high, approximately 1, the performance drops off rapidly. The flow ratio selected for the nominal design point was .38, which is the optimum point for system performance.

1.4 Superheaters

Studies were made to determine the desirability of superheaters for both boilers. The performance of the system was degraded when any superheat was added to the high temperature boiler. Any superheat added to the isobutane required additional energy from the brine, and the pinch point limitation in the low temperature boiler demanded more brine flow to produce the same power. Thus, system performance suffers. Superheating after the low temperature boiler would improve the system performance since the isobutane flow rate through this leg is approximately four-tenths of that through the high temperature leg and very little energy is required from the brine flow to superheat as high as 50°F. This superheat would improve the performance of the low temperature turbine significantly. However the low pressure turbine exhaust carries considerable superheat, on the order of 8-10 Btu/lb, which is not much less than the energy output of the low pressure turbine itself. A regenerator would reclaim much of this wasted energy, as well reducing the condensing requirements.

A more efficient cycle than the design case therefore, consists of no superheater on the high pressure side, 50°F superheat on the low pressure side and the use of a regenerator. This requires extra equipment, lines, valves, and associated spares, which complicates the system considerably. A system which has no regenerator and no superheater would result in about a 10% loss in system performance and in an increase in heat transfer area of approximately 10%. However, it was decided to exclude the superheater and regenerator for the preliminary design case in order to make layout and costing easier and, in general, to simplify the overall system. Later, corporations of these features could then be evaluated on their ability to pay for any cost increase that might occur.

2.0 Assumptions for the Determination of the Design Case for the Geothermal Binary Cycle

The basic assumptions are as follows:

1. All pump efficiencies were assumed to be 80%. Turbine efficiency was assumed to be 85% and electrical efficiency was assumed to be 95%. For these studies, a power fraction of 1 was also assumed.
2. The limiting pinch point was assumed to be 10°F for either boiler.
3. The overall pressure drop for each leg for the isobutane flow was set at 40 psid. Three psid was assumed across the isobutane condenser.
4. The brine inlet temperature was assumed to be 300°F and each brine supply well will deliver 1500 gal/min. It was also assumed that twice as many supply wells as reinjection wells were required.
5. The total brine pump pressure rise was set equal to 480 psid due to system design requirements.
6. Both cooling tower and aquifer supply systems were considered for use in the condenser. Since the number of aquifer supply wells, reinjection wells, and the drawdown of the supply wells due to the large amounts of cooling water required is unknown at this time, the aquifer was not considered as an alternative to supply cooling water to the condenser for this study. The state points thus generated show the use of a wet cooling tower. The power shown for the condensing water pump reflects a 10 psid pressure drop in the condenser, the line pressure drops, a 40-ft head rise required by the cooling tower, and tower fan power requirements.
7. The condensing water temperature inlet from the cooling tower was set at 75°F. This was obtained from cooling tower performance curves and the climatic conditions existing at Raft River. The water outlet of the condenser was set 10°F less than the condensing temperature, or 95°F.

8. The overall heat-transfer coefficients (U's) that were used for the design case were derived from manufacturers data corresponding to a fouling factor of $0.003 \text{ Btu/hr-ft}^2$. The U values corresponding to a brine-side fouling factor of 0.001 are also shown in the Table I below.

TABLE I

Overall Heat Transfer Coefficients Used for Heat Exchanger Calculations

<u>Heat Exchanger Description</u>	<u>Design Case U's (Btu/hr-ft²-°F) Fouling Factor = 0.003</u>	<u>Alternate Case U's (Btu/hr-ft²-°F) Fouling Factor = 0.001</u>
HP Boiler	157	229
HP Preheater	120	158
LP Boiler	145	204
LP Preheater	110	141
Condenser	133	133*

*Since the condenser does not see the brine, the fouling factor, and therefore the U value, remains unchanged.

3.0 Sensitivity Study

After the design case was fixed, sensitivity studies were run to determine plant response to variation in primary plant parameters. This is accomplished by varying external plant parameters to ascertain their effect on plant size and brine utilization. These sensitivity studies are performed on a variable plant size (holding power constant) and not on a fixed plant configuration. Six parameters were considered: brine inlet temperature, condenser temperature, pinch point, flow split, the amount of superheat added to the LP boiler, and mechanical efficiencies. These parameters were varied separately to determine individual, not cumulative, effects upon the plant performance. Two good indicators of the plant response are effectiveness (Btu's of work produced per pound of brine) and heat transfer area (HTA). HTA is defined as the sum of the heat exchanger areas required for the preheaters, boilers, superheaters (if any), condenser, and regenerators (if any). The percent change from the design condition for these two indicators is plotted versus the change in the variable parameter. It should be noted that the plant has been optimized at the design condition, and that by varying different combinations of interrelated parameters, different optimization points can be reached.

3.1 Brine Inlet Temperature (Figure 3.1-1)

As brine inlet temperature increases from the design case, effectiveness increases due to the higher temperature heat source. There is a larger log mean temperature difference which indicates better heat transfer and that more heat is available from the brine, requiring less brine flow. The pinch point of the second boiler controls at higher brine inlet temperatures, thus, the brine temperature at the exit of the Low Pressure (LP) boiler (brine side exit) is fixed. This means that as brine inlet temperature increases, the LP boiler area decreases proportionally. However, as brine inlet temperature increases, the area of the LP preheater must increase to accommodate the reduced brine flow and still provide the necessary preheat to the temperature of the low pressure boiler. There is a tradeoff point where the area of the LP preheater increases faster than the area of the LP boiler decrease, and

Figure 3.1-1 Design Case Sensitivity
to Brine Inlet Temperature
Variations

Figure 3.2-1 Design Case Sensitivity
to Variations in Isobutane
Condensing Temperature

the total heat transfer area curve turns upward.

As brine inlet temperature decreases, effectiveness decreases, since the log mean temperature differences is reduced, and the brine flow must increase. At lower temperatures, approaching the High Pressure (HP) boiler temperature, the pinch point of the HP boiler controls. This fixes the brine temperature at the outlet (brine side) of the HP boiler and thereby fixes the brine flow rate and the amount of brine energy available to the rest of the system. The area of the HP boiler must increase to accomodate the increased brine flow rate, and more heat is available to the rest of the system and other HX areas go down. Therefore, the area of the HP boiler increases, and the areas of the HP preheater, and the LP boiler and preheater decrease. There is a tradeoff where the net decrease is greater than the increase in the HP boiler area, and total heat transfer area decreases.

The condenser area proportionally increases as brine inlet temperature decreases due to slightly elevated isobutane flow rates. This increase is due to the extra working fluid mass flow rate required to obtain the amount of extra power to pump the additional brine. The percentage change is small.

3.2 Condenser Temperature (and therefore turbine back pressure) (Figure 3.2-1)

As condenser temperature increases from the design case, effectiveness decreases due to the reduced turbine pressure ratio. Isobutane flow must increase to provide the same power from the lower pressure turbine, which puts out less Btu/lb of isobutane. HTA increases due to the increased amount of superheat that is required to be extracted before condensation.

As condenser temperature decreases, HTA again increases due to a sharp increase in the area of the LP preheater (liquid isobutane has to be preheated from a lower temperature). Effectiveness again decreases due to the higher pumping power required to raise the isobutane from a lower pressure.

ANC-A-5405

Figure 3.3-1 Design Case Sensitivity
to Pinch Point Variations

3.3 Pinch Point (Figure 3.3-1)

As the pinch point is increased from the design case, effectiveness goes down because less of the available brine energy is utilized. Heat transfer area also decreases due to the larger terminal differences for the heat exchangers. Conversely, when the pinch point is decreased, effectiveness goes up due to utilization of more available brine energy, and heat transfer area goes up due to small terminal temperature differences. (Note: If the pinch point goes to zero, the size of the heat exchanger becomes infinite.) Since the heat transfer area is proportional to the log of the mean temperature difference, the heat transfer curve is exponential and the heat transfer area rises sharply as pinch points get very small.

3.4 Flow Ratio (wf_2/wf_1)(Figures 3.4-1, 3.4-2)

As the flow ratio is increased, more flow is diverted to the low pressure (and lower temperature) working fluid path. This has three effects. First, it shifts control of the pinch points; second, less fluid must be heated to higher temperatures; and third, more fluid runs through the LP turbine, which has a lower pressure ratio, therefore, less possible power output on a per pound basis. The first effect shifts control from the HP pinch point to the LP pinch point at a flow split of approximately .35. For a more detailed discussion of the effects of pinch point control, see Section 3.1. The second effect reduces heat transfer area and increases effectiveness. The third effect reduces effectiveness. These three effects are additive and result in a net decrease in heat transfer area and effectiveness. There is a sharp break in both curves when pinch point control switches from the HP pinch point to the LP pinch point, since they are, in effect, two separate curves.

Figure 3.4-1 Design Case Sensitivity
to Flow Split Variations

Figure 3.4-2 Design Case Sensitivity
to Variations in Flow
Split

3.5 Amount of Superheat on the LP Leg (Figures 3.5-1, 3.5-2)

Superheat was varied from 0 to 40° for the LP boiler. As superheat increases, effectiveness decreases. Ordinarily, superheat would increase effectiveness due to the increased power output of the LP turbine. In this case however, the pinch point on the LP boiler presents a problem. When the amount of heat required for the superheat is extracted from the brine, the temperature of the brine exit of the boiler remains fixed with a fixed pinch point. Therefore, more brine flow is required to provide the additional energy. The increase in LP turbine output is not large enough to offset the increased brine flow and corresponding increase in brine pumping power requirements. In order to maintain the same net power output, isobutane flow rate must also increase, which in turn, increases isobutane pumping power, etc. The flow rates and pumping powers reach another higher equilibrium level, at a reduced effectiveness value. If, however, the flow split were also varied, it would be possible to increase effectiveness with superheat. If more isobutane flow were diverted to the HP boiler, less energy would be required from the brine in the LP boiler and more would be available to the LP superheater. The HP boiler pinch point would decrease and the LP pinch point would increase, allowing superheat without substantial brine flow increase.

Heat transfer area decreased, then increases. This trend is also a result of the LP boiler pinch point problem. As brine flow increases to accommodate superheat, the pinch point on the HP boiler increases, requiring less area (larger log mean temperature difference) for the HP boiler and preheater. At the same time, the superheat area is increasing. Eventually, the increase in superheater area is greater than the decrease in HP boiler and preheater areas and total heat transfer area increases.

Figure 3.5-1 Design Case Sensitivity
to Variations in Superheat
on the Low Pressure Boiler

Figure 3.5-2 Design Case Sensitivity
to Variations in Superheat
on the Low Pressure Boiler

It should be noted that this comparison was performed for a system without regeneration. The superheat remaining after expansion through the LP turbine is not utilized. If regeneration were added to the cycle, this waste heat would be used in the preheating process. Thermal efficiency would increase but effectiveness is still down due to the increased brine flow. The use of a regenerator also makes the system less sensitive to fluctuations in brine inlet temperature. If brine inlet temperature increases, regenerator flow can be decreased, so that major brine flow rate changes are not necessary and brine is not exhausted at high temperatures. At low brine inlet temperatures, the pinch point of the HP boiler fixes brine temperature at the boiler outlet. Increased brine flow is required to maintain downstream system temperatures. The regenerator can supply a large portion of the preheating, requiring less energy from the brine. Also, for lower brine temperatures, more of the isobutane flow can be diverted to the LP turbine, requiring less flow required to the HP leg of the system. This means that less isobutane needs to be heated to the higher temperature, providing more brine energy for the rest of the system without increasing brine flow rate.

In spite of the advantages cited above, a regenerator was not included in the design case. Although a system with superheaters and regenerators is more thermodynamically efficient, a less complex (and less expensive) system is desirable as a baseline design.

3.6 Mechanical Efficiencies (Figures 3.6-1, -2, -3)

Turbine, electrical, and pump efficiencies directly affect effectiveness and heat transfer area. As the mechanical efficiency of the component increases, effectiveness increases because less flow is required for the same net power output. Due to the reduced flow rates, heat transfer area decreases. Conversely, as mechanical efficiency decreases, effectiveness decreases and heat transfer area increases due to increased flow rates required. 22

Figure 3.6-1 Design Case Sensitivity to Variations in Pump Efficiency

Figure 3.6-2 Design Case Sensitivity to Variations in Turbine Efficiency

Figure 3.6-3 Design Case Sensitivity to Variations in Electrical Generator Efficiency

4.0 Tolerance Study

The design case state point was derived using the reasoning described in Section 1.0 and the sensitivities discussed in Section 3.0. A tolerance study was run on the base case estimating tolerances of various parameters affecting system performance. For parameters that were input into the computer model, the tolerances used, and the methods used to determine the tolerance effect on plant performance are as follows:

4.1 Turbine Entropies

Lacking a consistent properties deck, linear interpolation was applied was applied to obtain data from available tables. It was assumed that entropy, and corresponding enthalpies were accurate to $\pm 1/2\%$. Since the specific output of the turbines is low (on the order of 20 Btu/lb), it was felt that the most significant effect of entropy deviations would occur at the turbine inlet. Since the computer code performs a table lookup to determine the enthalpy change for an isentropic expansion, the turbine outlet entropies and enthalpies would be biased no more than the values at the turbine inlet. Two cases were run, changing the inlet entropies by $\pm 1/2\%$ from the base case. The effects on various plant parameters is shown in Table II. The maximum tolerance swing affected the coolant water flow rate and its pumping power, and was found to be + 28.1%, -18.7%.

4.2 Specific Heats:

The assumption of unvarying specific heats at constant pressure (C_p) results in a large error margin. Therefore, the input for C_p was split into five areas: C_{p1} @ the inlet to the HP turbine, C_{p2} @ the inlet to the LP turbine, C_{p3} @ the turbine exhausts, C_{p4} @

the condenser, and Cp_5 @ the HP preheater. Values of $Cp \pm 5\%$ were input to the base case for two additional cases to determine plant response. See Table II. The effect of variation in these Cp 's is not large and is not of major concern.

4.3 Mechanical Efficiencies

Pump efficiencies, turbine efficiencies, and electrical efficiencies were considered separately. Pump and turbine efficiencies were varied $\pm 6\%$, and the electrical efficiency was varied $\pm 1\%$ from the base case. For a manufactured turbine, an efficiency tolerance of $\pm 6\%$ is impossibly large. However, since the isobutane turbines have not been designed yet, it was felt that $\pm 6\%$ is a more realistic estimate than the manufacturers usual $\pm 1/2\%$. Therefore, for each parameter, two additional cases were run to establish tolerance limits. Table II summarizes the effects on plant performance. The effect of pump and electrical efficiency uncertainty was small compared to the effect of turbine efficiency variation. This is due to the fact that turbine specific output is small, and therefore sensitive to the variations in mechanical efficiency.

4.4 Overall Heat Transfer Coefficients for the Heat Exchangers (U's): and Brine Inlet Temperature:

The effect of uncertainty in the U-values dominated the tolerance study, due to the large uncertainties involved, which will not be resolved until studies have been completed to determine fouling factors for the brine side of the heat exchangers. The U and brine inlet temperature tolerances were derived indirectly by combining the effects of variations in brine inlet temperature with the U-variations. When the brine inlet temperature was varied from the 300°F design case to 280°F and 320°F , the effectiveness varied $+16.9\%$, -28.5% . These variations were applied to the design case net power output to obtain tolerances to minimize

TABLE II

BASE CASE TOLERANCES

	Turbine Entropies		Specific Heats (Cp)		Pump Efficiencies		Turbine Efficiencies		Electrical Efficiencies		Dirty U's @TB _{in} =280°F	Clean U's @TB _{in} =320°F	Total + Tolerances	Total - Tolerances
	+½%	-½%	+5%	-5%	+6%	-6%	+6%	-6%	+1%	-1%	-0 to -24%	+10%	%	%
Effectiveness	-20.4	+20.5	-1.5	+1.5	+8	-9	+8.1	-8.1	+1.5	-1.5	-35.5	+40.5	+46.2	-41.8
Net Power	0	0	0	0	0	0	0	0	0	0	-28.5	+16.9	+16.9	-28.5
Isobutane Flow Rate 1	+17.1	-17.1	+4	-.4	-.8	+9	-7.5	+8.8	-1.5	+1.5	-13.5	+12.7	+23.1	-24.2
Isobutane Flow Rate 2	+25.6	-9.4	-1.2	+1.2	-.8	+9	-7.5	+8.8	-1.5	+1.5	+19.1	-7.4	+33.2	-14.3
Brine Flow Rate	+25.6	-17.1	+1.5	-1.5	-.8	+9	-7.5	+8.8	-1.5	+1.5	+10.9	-16.8	+29.3	-25.2
Coolant water Flow Rate	+28.1	-18.7	0	0	-.8	+9	-8.2	+9.7	-1.5	+1.5	-7.7	+2	+29.8	-21.9
Brine Pumping Power	+25.6	-17.1	+1.5	-1.5	-.8	+9	-7.5	+8.8	-1.5	+1.5	+10.9	-16.8	+29.3	-25.2
Coolant Water Pumping Power	+25.6	-17.1	0	0	-.8	+9	-8.2	+9.7	-1.5	+1.5	-7.7	+2	+27.4	-20.5
Isobutane Pumping Power	+28.1	-18.7	+2	-.2	-6.6	+7.7	-7.5	+8.8	-1.5	+1.5	-27.2	+15.5	+34.2	-34.5
Brine Inlet Temperature	0	0	0	0	0	0	0	0	0	0	-6.7	+6.7	+6.7	-6.7
Brine Temp @ Outlet of Boiler 1	0	0	+3	-.3	0	0	0	0	0	0	+3.8	-5.2	+3.8	-5.2
Brine Temp @ Outlet of PH 1	0	0	-.4	+5	0	0	0	0	0	0	+15.1	-17.3	+15.1	-17.3
Brine Temp @ Outlet of Boiler 2	0	0	0	0	0	0	0	0	0	0	+2.9	-5.8	+2.9	-5.8
Brine Temp @ Outlet of RH2	0	0	+1.9	-2	0	0	0	0	0	0	+8.8	-13.2	+9.0	-13.4
Total Isobutane Flow Rate	+25.6	-17.1	-.1	+1	-.8	+9	-7.5	+8.8	-1.5	+1.5	-6.6	0	+27.1	-19.9

brine flow rate fluctuations. These power values were input into the program with the extreme U-values, the pinch points were set to maintain heat exchanger area constant. Although net power output varied (+16.9%, - 28.5%), total isobutane flow did not vary significantly. Instead, the flow ratio varied considerably, changing the relative output of the two turbines, so that more isobutane was run through the HP turbine (larger specific power output) for the higher net power level, and vice versa.

BINARY CYCLE GEOTHERMAL POWER PLANT
SYSTEMS ANALYSIS MODEL

ANC-A-5398

STATE POINT	TEMPERATURE (°F)	PRESSURE (psia)	FLOW RATE (lbm/hr)
1	105+2%, -1%	396+43%, -23%	$2.7 \times 10^6 + 27\%, -20\%$
2	160+16%, -8%	368+39%, -22%	$1.9 \times 10^6 + 23\%, -24\%$
3	230+14%, -8%	354+35%, -20%	$1.9 \times 10^6 + 23\%, -24\%$
4	230+14%, -8%	347+33%, -19%	$1.9 \times 10^6 + 23\%, -24\%$
5	128 ± 3%	80+3%, -2%	$1.9 \times 10^6 + 23\%, -24\%$
6	120 ± 2%	80+2%, -1%	$2.7 \times 10^6 + 27\%, -20\%$
7	101 ± 1%	77 ± 1%	$2.7 \times 10^6 + 27\%, -20\%$
8	160+16%, -8%	169+45%, -16%	$7.3 \times 10^5 + 33\%, -14\%$
9	160+16%, -8%	162+33%, -12%	$7.3 \times 10^5 + 33\%, -14\%$
10	116 ± 3%	80+3%, -2%	$7.3 \times 10^5 + 33\%, -14\%$
11	300 ± 7%	238+45%, -23%	$2.5 \times 10^6 + 29\%, -25\%$
12	243+7%, -5%	213+42%, -19%	$2.5 \times 10^6 + 29\%, -25\%$
13	202+15%, -17%	201+40%, -17%	$2.5 \times 10^6 + 29\%, -25\%$
14	170+3%, -6%	188+38%, -15%	$2.5 \times 10^6 + 29\%, -25\%$
15	130+9%, -13%	138+30%, -7%	$2.5 \times 10^6 + 29\%, -25\%$
16	75 ± 1%	22+31%, -19%	$1.9 \times 10^7 + 30\%, -22\%$
17	95 ± 1%	12 ± 1%	$1.9 \times 10^7 + 30\%, -22\%$

NET POWER = 10MW (e) + 17%, -29%

TURBINE OUTPUT = 15.05MW

PLANT POWER REQUIREMENTS = 5.05MW

BRINE PUMP POWER = 1.43MW* + 29%, -25%

ISOBUTANE PUMP POWER = 1.40MW + 34%, -3

COOLANT WATER PUMP POWER = 1.45MW + 27%, -21%

ELECTRICAL LOSSES = 0.75MW ± 2%

BRINE REINJECTION PRESSURE = 300 psia

EFFECTIVENESS = 13.43 BTU/LB OF BRINE

*Includes Brine Supply and Reinjection Pumps

Figure 4.5-1

4.5 Discussion

These various tolerances were summed up by taking the square root of the sum of the squares of the component tolerances. All like effects were combined, regardless of whether the tolerances in the varying parameters were positive or negative. These final tolerances are shown in the last two columns in Table II, and were applied to the design state point. (See Figure 4.5-1)

Additional tolerances were required for pressure and temperature values. These fluctuations are due to the variations in flow rate in different parts of the system. The pressure drop through a section of line (or component) is assumed proportional to the square of the mass flow through the line. The change in mass flow is obtained from the previous tolerance studies.

For the tolerances on pressure,

$$\% \Delta P = \frac{[\Delta P_0 \times (\% \dot{W})^2] - \Delta P_0}{P}$$

where: $\% \Delta P$ = the + or - pressure tolerance corresponding to the flow rate tolerance

ΔP_0 = the nominal pressure drop

$\% \dot{W}$ = the minimum or maximum value of flow rate, expressed in % of nominal flow rate

P = the nominal pressure

This method was applied to the brine, the isobutane, and the coolant water.

Temperature tolerances for the isobutane were found by comparing the temperature corresponding to the tolerance saturation pressures. Brine temperature tolerances were computed by the program. Coolant water temperature tolerances were small, since they are regulated

by the cooling tower system.

It is important to note that these tolerances (pressure, temperature, flow rate, power, pumping requirements) do not together represent a state point. They only represent largest + or - swings possible due to combinations of favorable or adverse factors. Each tolerance value is independent of the others in that the conditions that would cause a given variation do not necessarily correspond to the conditions that are represented by another tolerance value.

**BINARY CYCLE GEOTHERMAL POWER PLANT
SYSTEMS ANALYSIS MODEL**

ANC-A-8398

	TEMP. (°F)	PRESS. (psia)	FLOW RATE (LBM/HR)	
1	105	396	5.8×10^6	NET POWER = 22.1 MW (e)
2	160	368	4.2×10^6	PUMPING POWER
3	230	354	4.2×10^6	BRINE = 2.7 MW
4	230	347	4.2×10^6	ISOBUTANE = 3.0 MW
5	128	80	4.2×10^6	COOLANT WATER = 3.2 MW
6	120	80	5.8×10^6	
7	101	77	5.8×10^6	BEST U'S
8	160	169	1.6×10^6	
9	160	162	1.6×10^6	
10	116	80	1.6×10^6	
11	320	238	4.8×10^6	BRINE REINJECTION PRESSURE = 300 psia
12	254	213	4.8×10^6	BRINE INLET TEMP = 320°F
13	207	201	4.8×10^6	EFFECTIVENESS = 15.8 BTU/LB OF BRINE
14	170	188	4.8×10^6	
15	124	138	4.8×10^6	
16	75	22	4.1×10^7	
17	95	12	4.1×10^7	

FIGURE 5.1-1

BINARY CYCLE GEOTHERMAL POWER PLANT
SYSTEMS ANALYSIS MODEL

ANC-A-5398

	TEMP (°F)	PRESS (psia)	FLOW RATE (LBM/HR)
1	105	396	2.5×10^6
2	160	368	1.7×10^6
3	230	354	1.7×10^6
4	230	347	1.7×10^6
5	128	80	1.7×10^6
6	120	80	2.5×10^6
7	101	77	2.5×10^6
8	160	169	8.4×10^5
9	160	162	8.4×10^5
10	116	80	8.4×10^5
11	280	238	3.2×10^6
12	240	213	3.2×10^6
13	212	201	3.2×10^6
14	182	188	3.2×10^6
15	152	138	3.2×10^6
16	75	22	1.8×10^7
17	95	12	1.8×10^7

NET POWER = 8.6 MW (e)

PUMPING POWER

BRINE = 1.8 MW

ISOBUTANE = 1.3 MW

COOLANT WATER = 1.4 MW

WORST U'S

BRINE REINJECTION PRESSURE = 300 psia

BRINE INLET TEMP = 280°F

EFFECTIVENESS = 9.33 BTU/LB OF BRINE

Figure 5.1-2

5.0 Off-Design Operational State Points

Several additional cases were run to determine the effects of variable operating conditions. A variation of the geothermal plant code was used to establish the effect of overall heat transfer coefficients (U's) variation on plant performance (See Appendix). This modified version iterated to hold the heat transfer area constant, but let the power vary for different U values. In this way, the effect of running "clean" and "dirty" heat exchangers, which were sized on the basis of nominal U's, can be determined.

5.1 The Effects of Variation in Brine Inlet Temperature on Power Output

The first set of cases were run to determine the effect on power output of variations in brine inlet temperature and U's. Tolerances of $\pm 10\%$ were placed on the "clean" U values, and $+10\%$, -0 to -24% on the "dirty" U values. Both the U's and the tolerances were derived from a combination of manufacturers' information and preliminary fouling data. Variations in brine inlet temperature ($\pm 7\%$) were incorporated into these calculations, providing the worst and best cases. The worst case (largest negative power swing) considered a minimum brine inlet temperature of 280°F , with "dirty" U's. The best case considered a minimum brine inlet temperature of 320°F and "clean" U's.

Figures 5.1-1 and 5.1-2 show the state points which are represented by these two cases. In Figure 5.1-1, using "clean" U's $+10\%$ and a brine inlet temperature of 320°F , effectiveness is increased by 18% and net power is increased by 121% . This increase is achieved by increasing the isobutane and brine flow rates. These elevated flow rates could be accommodated by the heat exchangers without varying the areas because of the increased U's. In other words, net power output can be changed only by raising isobutane flow through the turbines. In reality, turbines do not have the flexibility to run with flow variations of this magnitude. However, a spare turbine could be valved into service to utilize the

BINARY CYCLE GEOTHERMAL POWER PLANT
SYSTEMS ANALYSIS MODEL

ANC-A-5398

	TEMP (°F)	PRESS (psia)	FLOW RATE (LBM/HR)
1	105	396	2.6×10^6
2	160	368	1.9×10^6
3	230	354	1.9×10^6
4	230	347	1.9×10^6
5	128	80	1.9×10^6
6	120	80	2.6×10^6
7	101	77	2.6×10^6
8	160	169	7.0×10^5
9	160	162	7.0×10^5
10	116	80	7.0×10^5
11	320	238	2.2×10^6
12	253	213	2.2×10^6
13	206	201	2.2×10^6
14	170	188	2.2×10^6
15	124	138	2.2×10^6
16	75	22	1.9×10^7
17	95	12	1.9×10^7

NET POWER = 10 MW (e)

PUMPING POWER

BRINE = 1.2 MW

ISOBUTANE = 1.4 MW

COOLANT WATER = 1.4 MW

BRINE INLET TEMP = 320°F

NOMINAL U'S

BRINE REINJECTION PRESSURE = 300 psia

EFFECTIVENESS = 15.7 BTU/LB OF BRINE

Figure 5.2-1

BINARY CYCLE GEOTHERMAL POWER PLANT
SYSTEMS ANALYSIS MODEL

ANC-A-5398

	TEMP. (°F)	PRESS. (PSIA)	FLOW RATE (LBM/HR)
1	105	396	2.8×10^6
2	160	368	1.9×10^6
3	230	354	1.9×10^6
4	230	347	1.9×10^6
5	128	80	1.9×10^6
6	120	80	2.8×10^6
7	101	77	2.8×10^6
8	160	169	9.0×10^5
9	160	162	9.0×10^5
10	116	80	9.0×10^5
11	280	238	3.6×10^6
12	240	213	3.6×10^6
13	212	201	3.6×10^6
14	182	188	3.6×10^6
15	152	138	3.6×10^6
16	75	22	2.0×10^7
17	95	12	2.0×10^7

NET POWER = 10MW (e)

PUMPING POWER

BRINE = 2.0MW

ISOBUTANE = 1.4MW

COOLANT WATER = 1.6MW

NOMINAL U's

BRINE REINJECTION PRESSURE = 300 psia

BRINE INLET TEMP = 280°F

EFFECTIVENESS = 9.6 BTU/LB OF BRINE

Figure 5.2-2

additional flow. For the nominal U's, a fixed amount of brine is required to provide the necessary cycle energy for a fixed amount of isobutane and for fixed heat exchanger areas. With the increase in heat exchanger effectiveness, more isobutane can be heated by more brine in the same heat exchangers, therefore, net power output increases. Coolant water flow rate also increases to provide the necessary heat sink for the increased isobutane flow.

Figure 5.1-2 illustrates the 280°F brine inlet case with the "dirty" U's, -0% to -24%. Power output is reduced by 14% and effectiveness is down 30.5%. Isobutane flow is down and brine flow is up due to the added inefficiencies of the heat exchangers. (More brine is required to heat less isobutane.) Due to the increased brine flow and reduced heat transfer, brine is exhausted at 152°F, a 14.5% increase in temperature over the base case.

5.2 The Effects of Variation in Brine Inlet Temperature on Plant Parameters With Constant Power Output

The second set of cases describe the effect on plant size due to variations in brine inlet temperature maintaining a constant 10 MW(e) plant output. Figures 5.2-1 and -2 illustrate these state points. Figure 5.2-1 represents a brine inlet temperature of 320°F, with nominal U's. Brine flow has decreased by 12%, due to the more efficient energy transfer in the heat exchangers. (See Section 3.1, the sensitivity study on brine inlet temperature). Isobutane flow is also down slightly since brine pumping power requirements are decreased (less brine flow) and therefore a smaller percentage of gross power is lost internally.

Figure 5.2-2 represents a state point corresponding to a brine inlet temperature of 280°F with nominal U's. Brine exit temperature is increased to 152°F, so that brine flow has increased 44% due to the decrease in available heat energy. Effectiveness is down to 9.6 Btu/lb of brine, a 29% decrease from the design case. Isobutane flow has increased

4% to provide the additional brine pumping power.

5.3 The Effects of Variation in Brine Side Fouling Factor and Coolant Water Temperature

The third set of cases deals with a new nominal design point. The design base case was based upon the philosophy that the plant must produce 10 MW(e) at all times. If plant performance were allowed to vary, producing 10 MW(e) most of the time, a new nominal design case may be defined which would reduce overall plant size.

Recent research and experimental data have indicated that the brine side fouling factor may be reduced from 0.003 to 0.001. Atmospheric data⁽³⁾ indicate that a wet-bulb temperature of 60°F will not be exceeded 10% of the time during the four hottest months of the year, so that cooling tower water may be available at 65°F for a substantial amount of plant operating time. If the plant were designed with these new criteria, the effects of adverse conditions (fouling factor returning to 0.003 from brine scale and coolant water inlet temperature up to 75°F during very hot periods) must be determined.

A brine side fouling factor of 0.003 was used to determine nominal U's for the design state point. Reduction of the fouling factor to 0.001 would reduce brine-exposed heat transfer area by 26% from the design case and total heat transfer area by 15%. Holding these reduced areas constant, and returning to a fouling factor of 0.003, net power output is decreased 6% to 9.4 MW for the design case brine flow rate of 2.5×10^6 lbm/hr. See Figure 5.3-1.

The use of coolant water with an inlet temperature of 65°F (design case = 75°F) and outlet temperature of 90°F would allow condensing of the isobutane at 100°F (72 psia). This results in a 7% decrease from the design case in brine flow and a 24% decrease in coolant water flow for 10 MW(e) net power output. See Figure 5.3-2. Condenser area is down 18%, with a decrease in overall heat exchanger area of 11%. For the same reduced heat exchanger areas and brine flow rate, if the coolant water temperature is

BINARY CYCLE GEOTHERMAL POWER PLANT
SYSTEMS ANALYSIS MODEL

ANC-A-5398

	TEMP. (°F)	PRESS. (psia)	FLOW RATE (LBM/HR)	
1	105	396	2.5×10^6	NET POWER = 9.4MW (e)
2	160	368	1.8×10^6	
3	230	354	1.8×10^6	
4	230	347	1.8×10^6	PUMPING POWER BRINE = 1.4MW ISOBTANE = 1.3MW COOLANT WATER = 1.4MW
5	128	80	1.8×10^6	
6	125	80	2.5×10^6	
7	101	77	2.5×10^6	U'S CORRESPONDING TO A FOULING FACTOR OF 0.003 HEAT EXCHANGER AREAS CORRESPONDING TO A FOULING FACTOR OF 0.001
8	160	169	6.9×10^5	
9	160	162	6.9×10^5	
10	116	80	6.9×10^5	BRINE INLET TEMP. = 300°F BRINE REINJECTION PRESSURE = 300 psia EFFECTIVENESS = 12.7 BTU/LB OF BRINE
11	300	238	2.5×10^6	
12	246	213	2.5×10^6	
13	208	201	2.5×10^6	
14	177	188	2.5×10^6	
15	139	138	2.5×10^6	
16	75	22	1.8×10^7	
17	95	12	1.8×10^7	

FIGURE 5.3-1

BINARY CYCLE GEOTHERMAL POWER PLANT

SYSTEMS ANALYSIS MODEL

ANC-A-5398

	TEMP. (°F)	PRESS. (psia)	FLOW RATE (LBM/HR)	
1	100	396	2.5×10^6	NET POWER = 10MW (e)
2	160	368	1.8×10^6	
3	230	354	1.8×10^6	PUMPING POWER
4	230	347	1.8×10^6	BRINE = 1.3MW
5	125	77	1.8×10^6	ISOBUTANE = 1.3MW
6	122	77	2.5×10^6	COOLANT WATER = 1.1MW
7	96	72	2.5×10^6	
8	160	169	6.8×10^5	U'S CORRESPONDING TO A FOULING FACTOR OF
9	160	162	6.8×10^5	0.003
10	113	77	6.8×10^5	ISOBUTANE CONDENSING TEMPERATURE = 100°F
11	300	238	2.4×10^6	BRINE INLET TEMP. = 300°F
12	243	213	2.4×10^6	BRINE REINJECTION PRESSURE = 300 psia
13	202	201	2.4×10^6	EFFECTIVENESS = 14.4 BTU/LBM OF BRINE
14	170	188	2.4×10^6	
15	127	138	2.4×10^6	
16	65	22	1.4×10^7	
17	90	12	1.4×10^7	

FIGURE 5.3-2

BINARY CYCLE GEOTHERMAL POWER PLANT
SYSTEMS ANALYSIS MODEL

ANC-A-5398

	TEMP. (°F)	PRESS. (psia)	FLOW RATE (LBM/HR)
1	100	396	2.5×10^6
2	160	368	1.8×10^6
3	230	354	1.8×10^6
4	230	347	1.8×10^6
5	125	77	1.8×10^6
6	122	77	2.5×10^6
7	96	72	2.5×10^6
8	160	169	6.8×10^5
9	160	162	6.8×10^5
10	113	77	6.8×10^5
11	300	238	2.5×10^6
12	247	213	2.5×10^6
13	209	201	2.5×10^6
14	179	188	2.5×10^6
15	139	138	2.5×10^6
16	65	22	1.4×10^7
17	90	12	1.4×10^7

NET POWER = 9.9MW (e)

PUMPING POWER

BRINE = 1.4MW

ISOBUTANE = 1.3MW

COOLANT WATER = 1.1MW

U'S CORRESPONDING TO A FOULING FACTOR OF 0.003

HEAT EXCHANGER AREAS CORRESPONDING TO A FOULING FACTOR OF 0.001

ISOBUTANE CONDENSING TEMPERATURE = 100°F

BRINE INLET TEMP. = 300°F

BRINE REINJECTION PRESSURE = 300 psia

EFFECTIVENESS = 13.27 BTU/LB OF BRINE

FIGURE 5.3-3

BINARY CYCLE GEOTHERMAL POWER PLANT
SYSTEMS ANALYSIS MODEL

ANC-A-5398

STATE POINT	TEMPERATURE (°F)	PRESSURE (psia)	FLOW RATE (lbm/hr)
1	105+2%, -1%	396+43%, -23%	$2.7 \times 10^6 + 27\%, -20\%$
2	160+16%, -8%	368+39%, -22%	$1.9 \times 10^6 + 23\%, -24\%$
3	230+14%, -8%	354+35%, -20%	$1.9 \times 10^6 + 23\%, -24\%$
4	230+14%, -8%	347+33%, -19%	$1.9 \times 10^6 + 23\%, -24\%$
5	128 ± 3%	80+3%, -2%	$1.9 \times 10^6 + 23\%, -24\%$
6	120 ± 2%	80+2%, -1%	$2.7 \times 10^6 + 27\%, -20\%$
7	101 ± 1%	77 ± 1%	$2.7 \times 10^6 + 27\%, -20\%$
8	160+16%, -8%	169+45%, -16%	$7.3 \times 10^5 + 33\%, -14\%$
9	160+16%, -8%	162+33%, -12%	$7.3 \times 10^5 + 33\%, -14\%$
10	116 ± 3%	80+3%, -2%	$7.3 \times 10^5 + 33\%, -14\%$
11	300 ± 7%	238+45%, -23%	$2.5 \times 10^6 + 29\%, -25\%$
12	243+4%, -5%	213+42%, -19%	$2.5 \times 10^6 + 29\%, -25\%$
13	202+15%, -17%	201+40%, -17%	$2.5 \times 10^6 + 29\%, -25\%$
14	170+3%, -6%	188+38%, -15%	$2.5 \times 10^6 + 29\%, -25\%$
15	130+9%, -13%	138+30%, -7%	$2.5 \times 10^6 + 29\%, -25\%$
16	75 ± 1%	22+31%, -19%	$1.9 \times 10^7 + 30\%, -22\%$
17	95 ± 1%	12 ± 1%	$1.9 \times 10^7 + 30\%, -22\%$

NET POWER = 10MW (e) + 17%, -29%

TURBINE OUTPUT = 15.05MW

PLANT POWER REQUIREMENTS = 5.05MW

BRINE PUMP POWER = 1.43MW* + 29%, -25%

ISOBUTANE PUMP POWER = 1.40MW + 34%, -31%

COOLANT WATER PUMP POWER = 1.45MW + 27%, -21%

ELECTRICAL LOSSES = 0.75MW ± 2%

BRINE REINJECTION PRESSURE = 300 psia

EFFECTIVENESS = 13.43 BTU/LB OF BRINE

*Includes Brine Supply and Reinjection Pumps

Figure 6-1.

increased to 75°F, and if the isobutane condensing temperature is raised to 105°F, the coolant water flow rate is increased 66% and net power output is down 9.5% to 9.05 MW. If, however, the coolant water inlet temperature is lowered to 55°F, and the isobutane condensing temperature is lowered to 92.5°F, coolant water flow rate is down 29%, and net power is increased 11% to 11.1 MW(e) for the same heat exchanger areas.

Combining both effects, if the coolant water temperature and fouling factors are lowered as above, brine-related heat exchanger area is decreased 29% from the design case, with overall heat transfer area decreasing 25%. For the same heat exchanger areas and brine flow rate (2.4×10^6 lbm/hr) net power output is down 1% to 9.9 MW if the fouling factor is increased to 0.003. See Figure 5.3-3.

If the plant is designed for the new nominal case, it can be made smaller and more versatile. The effect of variations in coolant water inlet temperature and U values can be minimized by allowing the isobutane condensing temperature to float, accomodating the different heat exchanger temperature parameters.

6.0 Conclusions and Recommendations

The design case is shown in Figure 6-1. The tolerances included are discussed in Section 4.0. It should be noted that although tolerances are computed to aid in plant design, these preliminary values are high, and do not indicate system requirements. For a preliminary estimate, they indicate that more work must be done to reduce plant uncertainty. The major uncertainties, dealing with consistent, accurate properties values, the range of overall heat transfer coefficients, and brine inlet temperature, will be reduced after more data is acquired. The tolerances will then be recalculated, and will be in a workable range, so that system and component design considerations can be facilitated.

The sensitivity studies (Section 3.0) show the effects of varying certain primary plant parameters, and are useful for determining trends for plant cost and effectiveness tradeoffs.

The off-design operational state points are divided into two basic groups: those dealing with the design case, which is designed to produce 10 MW(e) always, and those dealing with a new nominal case, which is designed to produce 10 MW(e) most of the time.

For the design case, if brine inlet temperature and U's are varied, and the heat exchanger area remains constant, the net power output savings are -30.5% and +121% for the worst and best cases, respectively. If the net power output is restricted to 10 MW(e), and brine inlet temperature is varied, brine flow varies from -12% to +44% for the worst and best cases.

For the new nominal case, these changes were incorporated:

1. The coolant water inlet temperature was decreased to 65°F, with the exit temperature at 90°F.
2. The isobutane condensing temperature was lowered to 100°F, at 72 psia.
3. The brine-side fouling factor of 0.003 was lowered to 0.001.

The results of these changes were:

1. Brine-related heat exchanger area was lowered 29% from the design case.
2. Overall heat transfer area decreased 25%.
3. Coolant water flow rate is down 11%, and isobutane flow rate is down 7% for the design case brine flow.

The new nominal case plant will exhibit the same sensitivity trends as the design case, except that plant performance will be less sensitive to variations in parameters such as coolant water temperature and U's.

7.0 References:

1. W. W. Madsen and I. J. Ingvarsson, Preliminary Thermal Analysis of Binary Cycle for Geothermal Pilot Plant, ANCR-1191, February 1975.
2. J. F. Whitbeck, Review and Tentative Selection of a Working Fluid for Use with a Medium Temperature (300°F) Geothermal Heat Service, ANCR-1224, to be published.
3. The Marley Company, Engineering Weather Data, Mission, Kansas.

APPENDIX A

CALCULATION OF THE DUAL BOILING CYCLE

CALCULATION OF THE DUAL BOILING CYCLE

The dual boiling binary cycle shown on temperature-entropy coordinates is described below. WF1 is the isobutane flow through the high pressure leg of the system, and WF2 is the flow through the low pressure leg.

State Point	Temperature (°F)	Pressure (psia)
a	230	346
b	128	78
c	116	78
d	105	78
e	105	78
f	101	78
g	105	346
h	160	346
i	160	161
j	160	161
k	230	346

<u>Process</u>	<u>Comments</u>
a-b	High Pressure Turbine (WF1)
j-c	Low Pressure Turbine (WF2)
k-a	High Pressure Boiler (WF1)
i-j	Low Pressure Boiler (WF2)
f-g	Pump (WF1 + WF2)
g-h	Low Pressure Preheater (WF1 + WF2)
h-i	WF2 is valved down to 161 psia
h-k	High Pressure Preheater (WF1)
b-d	Condenser (WF1)
c-d	Condenser (WF2)
d-e	Condenser (WF1 + WF2)
e-f	Subcooling (WF1 + WF2)

Starting at f, the entire flow of isobutane is pumped to the pressure of the HP boiler (346 psia) at g. It is then heated to the temperature of the LP boiler (160°F), where WF2 is valved off and expanded to the pressure of the LP boiler (161 psia). WF2 is then boiled and run through the LP turbine. The remainder of the flow, WF1, is further heated to the temperature of the HP boiler (230°F), boiled, and expanded through the HP turbine. The two turbine exhausts are mixed, condensed, and subcooled, returning to point f, the pump inlet.

INPUT PARAMETERS

Variable Name	Definition	Base Case Value
1. TF1	HP boiler temperature	230°F
2. TF2	LP boiler temperature	160°F
3. PF1	HP boiler pressure	345.8 psia
4. PF2	LP boiler pressure	160.6 psia
5. HFG1	HP boiler latent heat	75.6 BTU/LB
6. HFG2	LP boiler latent heat	112.3 BTU/LB
7. HG1	HP boiler outlet enthalpy	-615.6 BTU/LB
8. HG2	LP boiler outlet enthalpy	-631.8 BTU/LB
9. TCOND	Condensing temperature	105°F
10. VFCOND	Liquid specific volume @ TCOND	0.03 FT ³ /LB
11. PCOND	Condensing pressure	77.5 psia
12. HFCOND	Condensing latent heat	131.6 BTU/LB
13. CPF1	Cp @ HP turbine inlet	0.63 BTU/LB-°F
14. XK	Ratio of specific heats	1.11
15. CONDH	Saturated vapor enthalpy @ condensing temperature	-648.3 BTU/LB
16. TB1	Brine inlet temperature	300°F
17. DLTPP1	HP boiler pinch point	10°F
18. DLTPP2	LP boiler pinch point	10°F
19. DLTSC	Degrees of subcooling in condenser	4°F
20. DTRG	Minimum terminal difference required for a regenerator	10°F
21. CPB	Cp of the brine	1 BTU/LB-°F
22. NP	Pump efficiency,	0.8
23. CPCW	Cp of the coolant water	1 BTU/LB-°F
24. TCWO	Coolant water temperature at condenser exit	95°F

Variable Name	Definition	Base Case Value
25. TCWIN	Coolant water temperature at condenser inlet	75°F
26. PNET	Net power output	10 MW
27. WBW	Brine well flow rate	2.5×10^6 LB/HR
28. DLTPF1	Pressure drop through HP leg of isobutane system	40 psid
29. DLTPF2	Pressure drop through LP leg of isobutane system	40 psid
30. UPH1	Overall heat transfer coefficient (U) for HP preheater	120 BTU/HR-FT ² -°F
31. UBOIL1	U for HP boiler	157 BTU/HR-FT ² -°F
32. UPH2	U for LP preheater	110 BTU/HR-FT ² -°F
33. UBOIL2	U for LP boiler	145 BTU/HR-FT ² -°F
34. UCOND	U for condenser	133 BTU/HR-FT ² -°F
35. XRAT	Flow split	0.5
36. NEL	Electrical efficiency,	0.95
37. NGT1	HP turbine efficiency,	0.85
38. NGT2	LP turbine efficiency,	0.85
39. URG1	U for the HP regenerator	0.0
40. URG2	U for the LP regenerator	0.0
41. ST1	HP turbine inlet entropy	1.1784 BTU/LB-°R
42. ST2	LP turbine inlet entropy	1.172 BTU/LB-°R
43. DLTSH1	Degrees of superheat for HP boiler	0.0
44. DLTSH2	Degrees of superheat for LP boiler	0.0
45. USH1	U for HP superheater	81 BTU/HR-FT ² -°F
46. USH2	U for LP superheater	74.2 BTU/HR-FT ² -°F
47. CPF2	Cp @ LP turbine inlet	0.51 BTU/LB-°F
48. CPF3	Cp @ turbine exhausts	0.46 BTU/LB-°F
49. CPF4	Cp @ condenser (liquid)	0.649 BTU/LB-°F
50. CPF5	Cp @ HP preheater	0.762 BTU/LB-°F

*Indicates equations used in the program

I. Brine

For the high pressure (HP) boiler, the energy added to the Isobutane (Q_{ISO}) is equal to the energy extracted from the urine (Q_{BRINE}) or $Q_{ISO} = Q_{BRINE}$.

$$Q_{ISO} = (WF1)(HFG1)$$

$$Q_{BRINE} = (WB)(CPB)[TB1P - (TF1 + DLTPP1)]$$

where: $WF1$ = Isobutane flow rate for HP path (lbm/hr)
 $HFG1$ = Latent heat of vaporization for Isobutane HP boiler (BTU/lbm)
 WB = Brine flow rate (lbm/hr)
 CPB = Cp of brine (BTU/lbm-°F)
 $TB1P$ = Brine temperature at the exit of the HP superheater (°F)
 $TF1$ = HP boiler temperature (°F)
 $DLTPP1$ = HP boiler pinch point (°F)

$$(1) \text{ or: } \frac{WF1}{WB} = \frac{(CPB)(TB1P - TF1 - DLTPP1)}{HFG1}$$

For the HP preheater, $Q_{ISO} = Q_{BRINE}$

$$Q_{ISO} = (WF1)(CPF5)(TF1 - TF2)$$

$$Q_{BRINE} = (WB)(CPB)[(TF1 + DLTPP1) - TB3]$$

where: $CPF5$ = Cp of Isobutane in the HP preheater (BTU/lbm-°F)
 $TF2$ = Low pressure (LP) boiler temperature (°F)
 $TB3$ = Brine temperature at the exit of the HP preheater (°F)

$$*(2) \text{ or: } TB3 = TF1 + DLTPP1 - \frac{(WF1)(CPF5)(TF1 - TF2)}{(WB)(CPB)}$$

Combining (1) and (2),

$$(3) \quad TB3 = TF1 + DLTPP1 - \frac{(CPF5)(TF1 - TF2)(TB1P - TF1 - DLTPP1)}{(HFG1)}$$

For the LP boiler, $Q_{ISO} = Q_{BRINE}$

$$Q_{ISO} = (WF2)(HFG2)$$

$$Q_{ISO} = (WB)(CPB)[TB3P - (TF2 + DLTPP2)]$$

where: $WF2$ = Isobutane flow rate for LP path (lbm/hr)

$HFG2$ = Latent heat of vaporization for Isobutane LP boiler (BTU/lbm)

$TB3P$ = Brine temperature at the exit of the LP superheater (°F)

$DLTPP2$ = LP boiler pinch point (°F)

$$(4) \quad \text{or: } WF2 = \frac{(WB)(CPB)(TB3P - TF2 - DLTPP2)}{HFG2}$$

If pinch point two controls, pinch point one must be calculated:

Combining (1) and (4)

$$(5) \quad DLTPP1 = TB1P - TF1 - \frac{(HFG1)(TB3P - TF2 - DLTPP2)}{(HFG2)(XRAT)}$$

$$\text{where } XRAT = \frac{WF2}{WFT}$$

For the HP superheater, $Q_{ISO} = Q_{BRINE}$

$$Q_{ISO} = (WF1)(CPF1)(DLTSH1)$$

$$Q_{BRINE} = (WB)(CPB)(TB1 - TB1P)$$

where: $CPF1$ = Cp of Isobutane in the HP superheater (BTU/lbm-°F)

$DLTSH1$ = Degrees of HP superheat (°F)

$TB1$ = Brine inlet temperature (°F)

$$(6) \quad \text{or: } TB1P = TB1 - \frac{(WF1)(CPF1)(DLTSH1)}{(WB)(CPB)}$$

Substituting (1) into (6),

$$*(7) \quad TB1P = \frac{\left[TB1 + \frac{(CPF1)(DLTSH1)(TF1 + DLTPP1)}{(HFG1)} \right]}{\left[1 + \frac{(CPF1)(DLTSH1)}{(HFG1)} \right]}$$

Substituting (7) into (5),

$$(8) \quad DLTPP1 = TB1 - TF1 - \frac{(HFG1) \left[1 + \frac{(CPF1)(DLTSH1)}{(HFG1)} \right] (TB3P - TF2 - DLTPP2)}{(HFG2)(XRAT)}$$

For the LP superheater, $Q_{ISO} = Q_{BRINE}$

$$Q_{ISO} = (WF2)(CPF2)(DLTSH2)$$

$$Q_{BRINE} = (WB)(CPB)(TB3 - TB3P)$$

where: $CPF2$ = Cp of Isobutane in the LP superheater (BTU/lbm-°F)

$DLTSH2$ = Degrees of LP superheat (°F)

$$(9) \quad \text{or: } TB3P = TB3 - \frac{(WF2)(CPF2)(DLTSH2)}{(WB)(CPB)}$$

Substituting (4) into (9), and rearranging,

$$*(10) \quad TB3P = \frac{\left[TB3 + \frac{(CPF2)(DLTSH2)(TF2 + DLTPP2)}{(HFG2)} \right]}{\left[1 + \frac{(CPF2)(DLTSH2)}{(HFG2)} \right]}$$

Substituting (3) into (10), and rearranging,

$$(11) \quad TB3P = \frac{\left[\frac{(CPF5)(TF1 - TF2)}{(HFG1)} + 1 \right] (TF1 + DLTPP1)}{\left[1 + \frac{(CPF2)(DLTSH2)}{HFG2} \right]} -$$

$$\frac{\frac{(CPF5)(TF1 - TF2)(TB1P)}{(HFG2)} + \frac{(CPF2)(DLTSH2)(TF2 + DLTPP2)}{(HFG2)}}{\left[1 + \frac{(CPF2)(DLTSH2)}{HFG2} \right]}$$

Combining Equations (11), (7), and (8),

$$*(12) \quad DLTPP1 = \frac{TB1 \{ (CPF)(TF1 - TF2) + XRAT[(CPF2)(DLTSH3) + HFG2] \}}{(CPF)(TF1 - TF2) + XRAT[(CPF2)(DLTSH2) + HFG2] + HFG1 + (CPF1)(DLTSH1)} +$$

$$\frac{[HFG1 + (CPF1)DLTSH1](TF2 + DLTPP2)}{(CPF)(TF1 - TF2) + XRAT[(CPF2)(DLTSH2) + HFG2] + HFG1 + (PF1)(DLTSH1)} - TF1$$

If pinch point one controls, pinch point two must be calculated:

Rearranging Equation (10),

$$(13) \quad TB3P = TB3 - \frac{(CPF2)(DLTSH2)(TB3P - TF2 - DLTPP2)}{(HFG2)}$$

Combining Equations (4) and (1), and substituting into Equation (13),

$$(14) \quad TB3P = TB3 - \frac{(XRAT)(CPF2)(DLTSH2)(TB1P - TF1 - DLTPP1)}{HFG1}$$

Combining Equations (13) and (14) and rearranging,

$$*(15) \quad DLTPP2 = TB3 - TF2 - \left\{ \frac{(XRAT)[HFG2 + (CPF2)(DLTSH2)](TB1P - TF1 - DLTPP1)}{HFG1} \right\}$$

For the brine temperature at the exit of the LP preheater, a heat balance is performed for the heating portion of the Isobutane cycle.

$$Q_{ISO} = Q_{BRINE} \text{ (total)}$$

$$Q_{ISO} = WF1[HG1 + (CPF1)(DLTSH1) - CONDH + HFCOND + (CPF4)(DTRG1) - DELR1] + \\ WF2[HG2 + (CPF2)(DLTSH2) - CONDH + HFCOND + (CPF4)(DTRG2) - DELR2]$$

$$Q_{BRINE} = (WB)(CPB)(TB1 - TB5)$$

where: HG1, HG2 = HP, LP boiler exit enthalpies for the Isobutane (BTU/lbm)

CONDH = Isobutane enthalpy at the vapor saturation line at the turbine exhaust pressure (BTU/lbm)

HFCOND = Isobutane latent heat of condensation at condensing temperature (BTU/lbm)

CPF4 = C_p of Isobutane in the liquid phase (BTU/lbm-°F)

DLTSC = Degrees of subcooling (°F)

DTRG = Terminal difference required for regenerator(s) (°F)

DTRG1, DTRG2 = DTRG when regeneration is used (°F), or
= DLTSC for no regeneration (°F)

DELR1, DELR2 = Enthalpy used for regeneration for HP, LP fluid path (BTU/lbm)

TB5 = Brine temperature at the exit of the LP preheater (°F)

$$*(16) \quad TB5 = TB1 - \left\{ \frac{WF1[HG1 + (CPF1)(DLTSH1) - CONDH + HFCOND + (CPF4)(DTRG1) - DELR1] + \\ WF2[HG2 + (CPF2)(DLTSH2) - CONDH + HFCOND + (CPF4)(DTRG2) - DELR2]}{(WB)(CPB)} \right\}$$

For brine flow rate

From Equation (1)

$$* (1) \text{ RATIO } 1 = \frac{WF1}{WB} = \frac{(CPB)(TB1P - TF1 - DLTPP1)}{HFG1}$$

$$\text{RATIO } 2 = \frac{WF2}{WFT} = \text{XRAT}$$

Combining Equations (1) and (4)

$$*(17) \frac{WF2}{WFT} = \frac{(HFG1)(TB3P - TF2 - DLTPP2)}{(HFG2)(TB1P - TF1 - DLTPP1)}$$

$$(18) \text{ RATIO } 3 = \frac{QNET}{WB}$$

QNET = net plant power output including pumping power requirements

$$(19) \text{ QNET} = (WF1)(QNET1) + (WF2)(QNET2) - (WB)(BP)$$

where: QNET1, QNET2 = Net power from HP, LP turbine including Isobutane and coolant water pumping power requirements (BTU/lbm) (See Equations 29 and 30 for derivation)

BP = Brine pumping power requirement (BTU/lbm)

For the assumed 480 psid brine pressure drop through the system, and including 80% pumping efficiency,

$$BP = \int_1^2 \frac{dp}{\rho \eta_p} = \frac{\Delta p}{\rho \eta_p} = \frac{480 \times 144}{57.77 \times 778 \times .8}$$

$$BP = 1.92 \text{ BTU/lbm}$$

$$\text{where } J = 778 \frac{\text{ft-lb}}{\text{BTU}}$$

$$\rho = 57.77 \frac{\text{lbm}}{\text{ft}^3}$$

Then Equation (19) becomes:

$$(20) \text{ QNET} = (WF1)(QNET1) + (WF2)(QNET2) - 1.92(WB)$$

Then:

$$(18) \text{ RATIO3} = \frac{\text{QNET}}{\text{WB}} = \frac{(\text{WF1})(\text{QNET1})}{(\text{WB})} + \frac{(\text{WF2})(\text{QNET1})}{(\text{WB})} - 1.92$$

Substituting Equations (1) and (17) into Equation (18),

$$*(21) \text{ RATIO3} = \text{RATIO1}[\text{QNET1} + (\text{RATIO2})(\text{QNET2})] - 1.92$$

The power required from the plant is PNET. Then, the ratio of $\frac{\text{PNET}}{\text{QNET}}$ should equal one (with the appropriate conversion factors).

$$*(22) \text{ Then WB} = \frac{(\text{PNET})(3.413 \times 10^6 \frac{\text{BTU}}{\text{MW-HR}})}{\text{RATIO3}}$$

where: $\text{RATIO3} =$

$$\frac{(\text{CPB})(\text{TB1P} - \text{TF1} - \text{DLTPP1})(\text{QNET1})}{\text{HFG1}} + \frac{(\text{HFG1})(\text{TB3P} - \text{TF2} - \text{DLTPP2})(\text{QNET2})}{(\text{HFG2})(\text{TB1P} - \text{TF1} - \text{DLTPP1})} - 1.92$$

II. Isobutane

Turbine exhaust enthalpies:

Isentropic enthalpies are found by a table look-up. Adding mechanical efficiencies:

$$*(23) \quad HG1TO = HI1 - NGT1(HI1 - HE1)$$

where: $HG1TO, HG2TO$ = HP, LP turbine exit enthalpy (BTU/lbm)

$HI1, HI2$ = HP, LP turbine inlet enthalpy; $HI1 = HG1 + (CPF1)(DLTSH1)$ (BTU/lbm)

$NGT1, HGT2$ = Mechanical efficiency of HP, LP turbine

$HE1, HE2$ = HP, LP turbine exit enthalpy (isentropic) from look-up (BTU/lbm)

Similarly,

$$*(24) \quad HG2TO = HI2 - NGT2(HI2 - HE2)$$

The net power from each turbine is equal to the turbine output minus isobutane and coolant water pumping power requirements. The brine pumping power is computed separately (See Equation 20).

$$*(25) \quad QT1 = (NEL)(HI1 - HG1TO) \text{ and}$$

$$*(26) \quad QT2 = (NEL)(HI2 - HG2TO)$$

where: $QT1, QT2$ = HP, LP turbine output (BTU/lbm)

NEL = Electrical efficiency

The pumping powers are:

$$W_p = \int_1^2 \frac{v dp}{\eta_p} = \frac{v \Delta p}{\eta_p}$$

For the Isobutane,

$$\Delta p = PF1 - PCOND + DLTPF1 \quad \text{for the HP leg}$$

$$\Delta p = PF2 - PCOND + DLTPF2 \quad \text{for the LP leg}$$

where: PF1, PF2 = HP, LP turbine inlet pressure (psia)
 PCOND = Condensing pressure (psia)
 DLTPF1, DLTPF2 = Line pressure losses for HP, LP leg (psia)
 $v = VFCOND$ = Specific volume of the liquid @ PCOND
 (FT³/lbm)
 $\eta_p = NP$ = Pump efficiency

$$*(27) \therefore QPW1 = \frac{144}{778} \frac{(VFCOND)(PF1 - PCOND + DLTPF1)}{NP}$$

and

$$*(28) QPW2 = \frac{144}{778} \frac{(VFCOND)(PF2 - PCOND + DLTPF2)}{NP}$$

where: $\frac{144}{778}$ is a conversion factor for units

The coolant water pumping power is also split into the two flow paths, where QCW1, QCW2 are the coolant water pumping power requirements.

Then, the net power from each turbine is:

$$*(29) QNET1 = QT1 - QPW1 - QCW1$$

$$*(30) QNET2 = QT2 - QPW2 - QCW2$$

For the Isobutane regenerator(s)

The enthalpy used for regeneration is:

$$*(31) DELR1 = HG1TO - CONDH - CPF3(DTRG - DLTSC)$$

and

$$*(32) DELR2 = HG2TO - CONDH - CPF3(DTRG - DLTSC)$$

where: CPF3 = Isobutane Cp through the lower pressure superheat remaining after expansion (BTU/lbm-°F)

$$*(33) \quad \text{DELRG} = \frac{(\text{DELRI})(\text{WF1}) + (\text{DELRI})(\text{WF2})}{(\text{WF1} + \text{WF2})}$$

where: DELRG = Mixed regeneration enthalpy (BTU/lbm)

Then, the temperature of the Isobutane at the regenerator exit (going to LP preheater) is:

$$*(34) \quad \text{TRGO} = (\text{TCOND} - \text{DLTSC}) + \frac{\text{DELRG}}{\text{CPF4}}$$

where: TRGO = Isobutane temperature at the exit of the regenerator(s) (°F)

TCOND = Isobutane condensing temperature (°F)

Flow Rates

$$*(35) \quad \text{WF1} = (\text{RATIO1})(\text{WB}) \quad \text{Equation (1)}$$

$$*(36) \quad \text{WF2} = (\text{RATIO2})(\text{WF1}) \quad \text{Equation (17)}$$

III. Coolant Water

For the condenser and the coolant water, $Q_{ISO} = Q_{CW}$

$$Q_{ISO} = (WF1)[DELHR1 + (CPF3)(DTRG) + HFCOND - DELR1] + (WF2)[DELHR2 + (CPF3)(DTRG) + HFCOND - DELR2]$$

$$Q_{CW} = (WCW)(CPCW)(TCWO - TCWIN)$$

where: DELHR1, DELHR2 = HP, LP leg enthalpy available for regeneration (BTU/lbm)

DELR1, DELR2 = HP, LP leg enthalpy used for regeneration

WCW = Coolant water flow rate (lbm/hr)

CPCW = Cp of coolant water (BTU/lbm-°F)

TCWO = Exit temperature of the coolant water (°F)

TCWIN = Inlet temperature of the coolant water (°F)

Let $Q_{ISO} = Q_{COND}$, then

$$*(37) \quad WCW = \frac{Q_{COND}}{(CPCW)(TCWO - TCWIN)}$$

Coolant water pumping power is

$$W_p = \int_1^2 \frac{dp}{\rho n_p} = \frac{\Delta p}{\rho n_p}$$

For the coolant water $\Delta p = 71$ psid, including 10 psid drop through the condenser, 40 ft head rise to the cooling towers, and line losses.

$$\text{Then } W_p = \frac{70}{62.4 \times .8} \times \frac{144}{778} = .263 \text{ BTU/lbm}$$

Splitting the coolant water requirements

$$(38) \quad QCW1 = \frac{(.263)[DELHR1 + (CPF3)(DTRG) + HFCOND - DELR1]}{(CPCW)(TCWO - TCWIN)}$$

$$(39) \quad QCW2 = \frac{(.263)[DELHR2 + (CPF3)(DTRG) + HFCOND - DELR2]}{(CPCW)(TCWO - TCWIN)}$$

If regeneration is used,

$$DELHR1 = DELR1 \text{ and } DELHR2 = DELR2$$

and the equations reduce to:

$$*(40) \quad QCW1 = \frac{(.263)[HFCOND + (CPF3)(DTRG)]}{(CPCW)(TCWO - TCWIN)}$$

$$*(41) \quad QCW2 = \frac{(.263)[HFCOND + (CPF3)(DTRG)]}{(CPCW)(TCWO - TCWIN)}$$

If no regeneration is used, or if the energy available for regeneration is not sufficient, according to the input minimum temperature difference DTRG:

$$DELR1 = DELR2 = 0 \text{ and}$$

$$DTRG1 = DTRG2 = DTRG = DLTSC$$

and the equations reduce to:

$$*(42) \quad QCW1 = \frac{(.263)[DELHR1 + HFCOND + (CPF4)(DLTSC)]}{(CPCW)(TCWO - TCWIN)}$$

$$*(43) \quad QCW2 = \frac{(.263)[DELHR2 + HFCOND + (CPF4)(DLTSC)]}{(CPCW)(TCWO - TCWIN)}$$

IV. Heat Exchanger Areas

Superheater Area

$$q = W C_p \Delta T = h A \Delta T$$

$$A = \frac{q}{h \Delta T} \quad \Delta T = \frac{\Delta T_{IN} - \Delta T_{OUT}}{\log \left(\frac{\Delta T_{IN}}{\Delta T_{OUT}} \right)}$$

$$q_1 = (W F_1)(C P F_1)(D L T S H_1)$$

$$h_1 = U S H_1$$

$$q_2 = (W F_2)(C P F_2)(D L T S H_2)$$

$$h_2 = U S H_2$$

$$\Delta T_{IN_1} = (T B 1 P - T F 1)$$

$$\Delta T_{OUT_1} = T B 1 - (T F 1 + D L T S H_1)$$

$$\Delta T_{IN_2} = (T B 3 P - T F 2)$$

$$\Delta T_{OUT_2} = T B 3 - (T F 2 + D L T S H_2)$$

$$*(44) \quad A S H_1 = \frac{(W F_1)(C P F_1)(D L T S H_1) \log \left[\frac{T B 1 P - T F 1}{T B 1 - T F 1 - D L T S H_1} \right]}{U S H_1 (T B 1 P - T B 1 + D L T S H_1)} \quad \text{Area of HP Superheater}$$

$$*(45) \quad A S H_2 = \frac{(W F_2)(C P F_2)(D L T S H_2) \log \left[\frac{T B 3 P - T F 2}{T B 1 - T F 2 - D L T S H_2} \right]}{U S H_2 (T B 3 P - T B 3 + D L T S H_2)} \quad \text{Area of LP Superheater}$$

Boiler Areas

$$q = WC_p \Delta T = hA \Delta T$$

$$A = \frac{q}{h \Delta T}$$

$$\Delta T = \frac{\Delta T_{IN} - \Delta T_{OUT}}{\log \left(\frac{\Delta T_{IN}}{\Delta T_{OUT}} \right)}$$

$$q_1 = (WB)(CPB)(TB1P - TF1 - DLTPP1)$$

$$q_2 = (WB)(CPB)(TB3P - TF2 - DLTPP2)$$

$$\Delta T_{IN_1} = DLTPP1$$

$$\Delta T_{OUT_1} = (TB1P - TF1)$$

$$\Delta T_{IN_2} = DLTPP2$$

$$\Delta T_{OUT_2} = (TB3P - TF2)$$

$$AB1 = \frac{(WB)(CPB)(TB1P - TF1 - DLTPP1) \log \left[\frac{DLTPP1}{TB1P - TF1} \right]}{(UBOIL1)(DLTPP1 - TB1P + TF1)}$$

Rearranging,

$$*(46) \quad AB1 = \frac{(WB)(CPB) \log \left[\frac{TB1P - TF1}{DLTPP1} \right]}{UBOIL1} \quad \text{Area of HP Boiler}$$

Similarly,

$$*(47) \quad AB2 = \frac{(WB)(CPB) \log \left[\frac{TB3P - TF2}{DLTPP2} \right]}{UBOIL2} \quad \text{Area of LP Boiler}$$

Preheater Areas

$$q = WC_p \Delta T = hA \Delta T$$

$$A = \frac{q}{h \Delta T}$$

$$\Delta T = \frac{\Delta T_{IN} - \Delta T_{OUT}}{\log \left(\frac{\Delta T_{IN}}{\Delta T_{OUT}} \right)}$$

$$q_1 = (WB)(CPB)(TF1 + DLTPP1 - TB3)$$

$$q_2 = (WB)(CPB)(TF2 + DLTPP2 - TB5)$$

$$\Delta T_{IN_1} = DLTPP1$$

$$\Delta T_{OUT_1} = (TB3 - TF2)$$

$$\Delta T_{IN_2} = DLTPP2$$

$$\Delta T_{OUT_2} = (TB5 - TRG0)$$

$$APH1 = \frac{(WB)(CPB)(TF1 + DLTPP1 - TB3) \log \left[\frac{DLTPP1}{TB3 - TF2} \right]}{(UPH1)(DLTPP1 - TB3 + TF2)}$$

Rearranging,

$$*(48) \quad APH1 = \frac{(WB)(CPB) \log \left[\frac{TB3 - TF2}{DLTPP1} \right]}{(UPH1) \left[\frac{(TF1 - TF2)}{TF1 + DLTPP1 - TB3} - 1 \right]} \quad \text{Area of HP Preheater}$$

Similarly,

$$*(49) \quad APH2 = \frac{(WB)(CPB) \log \left[\frac{TB5 - TRG0}{DLTPP2} \right]}{(UPH2) \left[\frac{(TF2 - TRG0)}{TF2 + DLTPP2 - TB5} - 1 \right]} \quad \text{Area of LP Preheater}$$

Regenerator Area

$$q = W\Delta h = hA\Delta T$$

$$A = \frac{q}{h\Delta T}$$

$$\Delta T = \frac{\Delta T_{IN} - \Delta T_{OUT}}{\log \left(\frac{\Delta T_{IN}}{\Delta T_{OUT}} \right)}$$

$$*(50) \quad QRG = (WF1)(DEL R1) + (WF2)(DEL R2)$$

$$h = \frac{URG1 + URG2}{2}$$

where: URG1, URG2 = U for regenerators (BTU/HR-FT²-°F)

QRG = Heat transferred in the regenerator (BTU/HR)

For the mixing of the HP & LP turbine exhaust streams,

$$*(51) \quad HMIX = \frac{(WF1)(HG1T0) + (WF2)(HG2T0)}{(WF1 + WF2)}$$

$$TMIX = \frac{HMIX}{C_p} + 1467 \quad (^\circ R) \quad \text{Absolute Scale}$$

$$*(52) \quad TMIX = \frac{HMIX}{C_p} + 1007 \quad (^\circ F)$$

$$\Delta T_{IN} = TMIX - TRGO$$

$$\Delta T_{OUT} = DTRG + DLTSC$$

$$\Delta T = \frac{(TMIX - TRGO - DTRG - DLTSC)}{\log\left(\frac{TMIX - TRGO}{DTRG + DLTSC}\right)} \quad \text{Area of the Regenerator}$$

$$*(53) \quad ARG = \frac{QRG(2)}{(URG1 + URG2)} \cdot \frac{(DTRG + DLTSC + TRGO - TMIX)}{\log\left(\frac{DTRG + DLTSC}{TMIX - TRGO}\right)}$$

Condenser Area

$$q = W\Delta h = hA\Delta T$$

$$A = \frac{q}{h\Delta T}$$

$$\Delta T = \frac{\Delta T_{IN} - \Delta T_{OUT}}{\log\left(\frac{\Delta T_{IN}}{\Delta T_{OUT}}\right)}$$

$$\begin{aligned} *(54) \quad Q_{COND} = & (WF1)[DELHR1 - DELR1 + HFCOND + (CPF3)(DTRG)] + \\ & (WF2)[DELHR2 - DELR1 + HFCOND + (CPF3)(DTRG)] \end{aligned}$$

where: Q_{COND} = The energy transferred in the condenser (BTU/HR)
 $DELHR1 = DELR1$ for regeneration

$$\Delta T_{IN} = TMIX - TCWO$$

$$\Delta T_{OUT} = TCOND - DLTSC - TCWIN$$

$$h = U_{COND} = U \text{ for condenser (BTU/HR-FT}^2\text{-}^\circ\text{F)}$$

$$*(55) \quad AC = \frac{Q_{COND} \log\left(\frac{TMIX - TCWO}{TCOND - DLTSC - TCWIN}\right)}{U_{COND}(TMIX - TCWO - TCOND + DLTSC + TCWIN)} \quad \text{Area of the Condenser}$$

IV. Miscellaneous Parameters

Effectiveness - Power output per lbm of brine

From Equation (19),

$$*(19) \frac{Q_{NET}}{WB} = \text{Effectiveness} = \text{RATIO 3} = \text{RATIO1}[Q_{NET1} + (\text{RATIO2})(Q_{NET2})] - 1.92 \frac{\text{BTU}}{\text{lbm of brine}}$$

$$*(56) \frac{P_{NET}}{WB} = \text{Effectiveness} = \frac{\text{RATIO3}}{3.413} \frac{\text{WATT-HR}}{\text{lbm of brine}}$$

$$\text{Thermal efficiency} = \frac{Q_{NET}(\text{OUTPUT})}{Q_{ADDED}}$$

From (20),

$$(20) Q_{NET} = (WF1)(Q_{NET1}) + (WF2)(Q_{NET2}) - (WB)(1.92)$$

$$(57) Q_{ADDED} = (WB)(CPB)(TB1 - TB5)$$

$$(58) \frac{Q_{NET}}{Q_{ADDED}} = \frac{(WF1)(Q_{NET1}) + (WF2)(Q_{NET2}) - (WB)(1.92)}{(WB)(CPB)(TB1 - TB5)}$$

Combining Equation (56) with Equation (19)

$$*(59) \frac{Q_{NET}}{Q_{ADDED}} = \eta_{th} = \frac{\text{RATIO3} \times 100}{(CPB)(TB1 - TB5)} \quad (\%)$$

Overall Effective Use of Energy Supplied (OEU)

By extracting all of the brine energy from the source temperature to the sink temperature, the maximum energy utilization is obtained. The source temperature is TB1 and the sink is TCWIN.

Then the maximum energy available is:

$$(WB)(CPB)(TB1 - TCWIN)$$

The actual energy output of the plant is QNET.

$$(60) \text{ Then, } OEU = \frac{QNET}{(WB)(CPB)(TBI - TCWIN)}$$

Combining with Equation (19)

$$*(61) \text{ } OEU = \frac{RATIO3 \times 100}{(CPB)(TBI - TCWIN)} \quad (\%)$$

V. Program Options

In the case of a perfect gas, the isentropic expansion through the turbine may be calculated in lieu of using the table lookup:

$$h = u + pv$$

$$dh = du + pdv + vdp$$

$$Tds = du + pdv$$

$$\therefore dh = Tds + vdp$$

$$\text{For } ds = 0, dh = vdp$$

$$pv = RT \text{ or } v = \frac{RT}{p}$$

$$dh = RT \frac{dp}{p}$$

$$c_p = \left(\frac{dh}{dT} \right)_p$$

$$c_p \frac{dT}{T} = \frac{Rdp}{p}$$

$$\ln T \Big|_1^2 = \frac{R}{c_p} \ln p \Big|_1^2 + \text{const}$$

$$\ln T_2 - \ln T_1 = \frac{R}{c_p} [\ln p_2 - \ln p_1]$$

$$\ln \left(\frac{T_2}{T_1} \right) = \frac{R}{c_p} \ln \left(\frac{p_2}{p_1} \right)$$

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^{\frac{R}{c_p}}$$

$$R = c_p - c_v$$

$$\frac{R}{C_p} = 1 - \frac{C_v}{C_p} = 1 - \frac{1}{K} = \frac{K-1}{K}$$

$$\text{Then } \frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^{\frac{K-1}{K}}$$

$$\text{or } T_2 = T_1 \left(\frac{p_2}{p_1} \right)^{\frac{K-1}{K}}$$

$$\Delta h \text{ through the turbine} = C_p \Delta T$$

$$\Delta T = T_1 - T_2 = T_1 \left[1 - \left(\frac{p_2}{p_1} \right)^{\frac{K-1}{K}} \right]$$

$$\Delta h = C_p T_1 \left[1 - \left(\frac{p_2}{p_1} \right)^{\frac{K-1}{K}} \right]$$

Adding the turbine mechanical efficiency,

$$\Delta h = \eta_t C_p T_1 \left[1 - \left(\frac{p_2}{p_1} \right)^{\frac{K-1}{K}} \right]$$

and the exhaust enthalpy,

$$h_2 = h_1 - \eta_t C_p T_1 \left[1 - \left(\frac{p_2}{p_1} \right)^{\frac{K-1}{K}} \right]$$

Substituting the variables used in the program,

$$*(62) \quad HG1TO = HI1 - (NGT1)(CPFF)(TI1 + 460) \left[1 - \left(\frac{PCOND}{PFI} \right)^{\frac{XK-1}{XK}} \right]$$

$$\text{where: } CPFF = \frac{CPF1 + CPF3}{2}$$

$$*(63) \quad \text{and } HG2TO = HI2 - (NGT2)(CPFH)(TI2 + 460) \left[1 - \left(\frac{PCOND}{PFI} \right)^{\frac{XK-1}{XK}} \right]$$

$$\text{where } CPFH = \frac{CPF2 + CPF3}{2}$$

If pinch points are input, the flow split may be calculated by Equation (17)

$$*(17) \quad XRAT = RATIO2 = \frac{WF2}{WF1} = \frac{(HFG1)(TB3P - TF2 - DLTPP2)}{(HFG2)(TB1P - TF1 - DLTPP1)}$$

For the supercritical cycle, a heater is used rather than the preheater-boiler-superheater combination. There is only one fluid stream.

$$Q_{ISO} = Q_{BRINE}$$

$$Q_{ISO} = (WF1)(HFG1 - DELR1)$$

where: $WF1$ = Total Isobutane flow (lbm/hr)

$HFG1$ = $HI1 - HF1$ (input $HFG1$) (lbm/hr)

$HF1$ = Isobutane enthalpy of saturated liquid at the condensing temperature

$$Q_{BRINE} = (WB)(CPB)(TB1 - TB5)$$

$$(64) \therefore TB5 = TB1 - \frac{(WF1)(HFG1 - DELR1)}{(WB)(CPB)}$$

$$\text{or } TB5 = TB1 - \frac{(RATIO1)(HFG1 - DELR1)}{(CPB)}$$

Although there is no real pinch point (no boiler), there is the possibility of temperature overlap between the Isobutane and the brine in the heater. Temperatures are checked along the length of the heater, and RATIO1 is changed to compute TB5 from the largest value of RATIO1 which will not cause temperature overlap in the heater. RATIO1 can be decreased below this value, but the resulting brine outlet temperature reaches a high level.

The heater area is calculated from:

$$q = WC_p \Delta T = hA \Delta T$$

$$q = (WB)(CPB)(TB1 - TB5)$$

$$\Delta T = \frac{\Delta T_{IN} - \Delta T_{OUT}}{\log \left(\frac{\Delta T_{IN}}{\Delta T_{OUT}} \right)}$$

The HP boiler is used as the heater, $\therefore h = UBOIL1$

$$\Delta T = \frac{TB1 - TF1 - TB5 + TRG0}{\log \left(\frac{TB1 - TF1}{TB5 - TRG0} \right)}$$

$$*(65) \quad A = \frac{q}{h \Delta T} = \frac{(WB)(CPB)(TB1 - TB5) \log \left(\frac{TB1 - TF1}{TB5 - TRG0} \right)}{(UBOIL1)(TB1 - TB5 - TF1 + TRG0)} \quad \text{Heater Area}$$

Ordinarily, the program adjusts flow rates, temperatures, and heat exchanger areas to produce the required net power output. There is an option, however, to keep the heat transfer areas constant, and to vary the power output to accommodate variable operating conditions. The program iterates, changing the power level, to close on the heat exchanger area called for.

There is also an option to run a single boiler cycle. Both the HP and LP legs of the plant are input identically, with a flow split of zero. The HP preheater and the LP boiler are zeroed out. In other words:

TF1 = TF2
PF1 = PF2
HFG1 = HFG2
HG1 = HG2
CPF1 = CPF2
DLTPP1 = DLTPP2
DLTPF1 = DLTPF2
NGT1 = NGT2
ST1 = ST2
DLTSH1 = DLTSH2
XRAT = 0.0

VI. Input to the Program

- A. First set of cards: Isobutane enthalpy-temperature data for supercritical case table lookup. Enthalpies are input in 8F10.4 format; the number of entries is NUN. (NUN must be defined on DATA card.) Corresponding temperatures are also input in 8F10.4, NUN entries.
- B. Second set of cards: Isobutane entropy-enthalpy data for turbine lookup routine. Entropies are input in 8F10.4 format; the number of entries is LIP. (LIP must be defined on DATA card.) Corresponding enthalpies are input in the same way. This set of data is applicable to one turbine exhaust pressure only. If the pressure is to be changed, see section on NSTOP options. If the turbine expansion is to be calculated, inserting blank cards for the enthalpy data will cause the lookup turbine exit enthalpies to be zeroed out.
- C. Third set of cards: Case information-cases may be stacked.

1. First Card: Title card containing any pertinent information.
2. Second Card: Heat exchanger data. This card flags the option for closing on heat exchanger areas and varying the net power output. The format is (1X, E10.4, I5). The heat transfer area in square feet is input in E10.4. IFLG is input in I5. This flags which area is to be kept constant:

IFLG = 1 - flag out this option
2 - total heat transfer area
3 - preheaters, boilers and superheaters
4 - HP boiler
5 - LP boiler
6 - HP preheater
7 - LP preheater
8 - condenser
9 - regenerator

3. Third Card: Starts data set for the case. For the first case, all 50 data must be input. The first column of a case data card is NSTOP, in I1 format, which calls several options:

NSTOP = 0 Go on to read the next card

- = 1 Placed on last card of last case run to indicate the end of case data.
- = 2 Placed on last card of a base case. All information input for a base case will be saved until it is replaced by another base case.
- = 3 Placed on the last card of a stacking case. The information contained in the stacking case cards will replace base case information for one run only, then will be scratched, and the base case information is retrieved.

- = 4 Placed on the last card of a supercritical case. This flags the supercritical option in the program.
- = 5 Placed on the last card of a case which changes the turbine exhaust pressure. An additional set of entropy-enthalpy data cards is also read in for the new pressure.

The next two columns of the card are blank followed by an index number (INDEX1) in the I2 format. The input parameters are placed in an array, numbered 1-50. The index number is the array number of the first data piece on the card. INDEX1 is followed by two spaces, then INDEX2, the number of pieces of input data following on that card is read in (I2 format). Starting in column 11, data is input in E10.4 format. Therefore, 1 to 7 pieces of consecutively numbered data may be read in on one card.

Then, for each case input, there must be a title card, a heat exchanger data card, and one or more data cards. The last data card of each case must be flagged with NSTOP equal from 1 to 5.

APPENDIX B

GEOHERMAL PROGRAM LISTING


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C**** PROGRAM FOR CUAL BOILING CYCLE POWER GENERATION SYSTEM
0001 DIMENSION A(50),B(7),TITLE(20),C(50)
0002 DIMENSION ENTH(10), XENTH(40), TEMP(10), T(40), TBRIN(10), S(11),
1F(11),HE(42),SCPP(10),TBRINS(10)
0003 EQUIVALENCE (A(1),TF1),(A(2),TF2),(A(3),PF1),(A(4),PF2),(A(5),HFG1
1 ),(A(6),HFG2),(A(7),HG1),(A(8),HG2)
0004 EQUIVALENCE (A(9),TCCND),(A(10),VFCOND),(A(11),PCOND),
1 (A(12),HFCOND),(A(13),CPF1),(A(14),XK),(A(15),CONDH),
2 (A(16),TBI),(A(17),CLTPP1),(A(18),CLTPP2),(A(19),CLTSC)
0005 EQUIVALENCE (A(20),CTRG),(A(21),CPB),(A(22),AP),(A(23),CPCW),
1 (A(24),TCWC),(A(25),TCWIN),(A(26),PNET1),(A(27),WBW),
2 (A(28),CLPF1),(A(29),CLPF2)
0006 EQUIVALENCE (A(30),UPH1),(A(31),UBOIL1),(A(32),UPH2),
1 (A(33),UBOIL2),(A(34),UCOND),(A(35),XRAT),(A(36),NEL),
2 (A(37),NGT1),(A(38),NGT2),(A(39),URG1),(A(40),URG2)
3,(A(41),ST1),(A(42),ST2),(A(43),CLTSH1),(A(44),CLTSH2),(A(45),USH1
4),(A(46),USH2),(A(47),CPF2),(A(48),CPF3),(A(49),CPF4),(A(50),CPF5)
0007 DATA NUN,LIP,DRATIC,URATIC/40,6,.1,.01/
0008 REAL NGT1,NGT2,NEL,AP
0009 CALL ECHO
C READ IN VARIOUS DATA FOR TABLES
0010 READ(5,1601)(XENTH(J),J=1,NUN)
0011 READ(5,1601)(T(I),I=1,NUN)
0012 2 READ(5,1601)(S(N),N=1,LIP)
0013 READ(5,1601)(H(N),N=1,LIP)
0014 IF(NSTCP .EQ. 5)GC TC 8
0015 1601 FORMAT(8F10.4)
C**** READ TITLE AND PRIMARY INPUT DATA
0016 5 READ(5,559)TITLE
0017 ICT3=0
0018 ICT4=0
0019 ICT5=0
0020 ICT6=0
0021 LCPC=0
0022 999 FORMAT(20A4)
C**** READ AHTX AND IFLG
0023 READ(5,130)AHTX,IFLG
0024 130 FORMAT(1X,E10.4,I5)
0025 IF(IFLG .NE. 10) GC TC 100
0026 READ(5,131) ABCL1,ABCL2,ATH1I,ATH2I
0027 131 FORMAT(1X,4F10.3)
0028 100 READ(5,989)NSTCP ,INDEX1,INDEX2,(B(I),I=1,INDEX2)
0029 989 FORMAT(11,2X,I2,1X,I2,2X,7E10.4)
0030 II=1
0031 NALL=INDEX1+INDEX2-1
0032 DO 112 I=INDEX1,NALL
0033 A(I)=B(II)
0034 112 II=II+1
0035 IF(NSTCP .EQ. 5)GC TC 2
0036 IF(NSTCP .GE. 1) GC TC 6
0037 GCTC 100
0038 6 IF(NSTCP .NE. 2) GC TC 8
0039 DO 7 I=1,50
0040 C(I)=A(I)
0041 7 CONTINUE
0042 8 ICK1=0

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0043      ICK2=C
0044      FNET = PNET1
0045      WRITE(6,800)TITLE
0046      800  FORMAT(1H1,20A4)
0047      WRITE(6,802)
0048      802  FORMAT(1H0,60X,'INPLT ECIT THIS CASE')
0049      WRITE(6,700) (A(I),I=1,50)
0050      700  FORMAT(1H0,1P10E13.4)
0051      WRITE(6,130)A+TX,IFLG
0052      IF(IFLG .EQ. 10) WRITE(6,131)ABCL1,ABCL2,ATH11,ATH21
0053      CTRG2 = CTRG
0054      CTRG1 = CTRG
0055      IF((CLTSC .GT. CTRG) .CR. ((URG1 .LE. 0.0) .AND. (URG2 .LE. 0.0)))
      ICTRG=DLTSC
0056      CPF=(CPF1+CPF2)/2.
0057      ICTM=0
0058      ARG=C.C
0059      T11=TF1+CLTSH1
0060      H11=HFG1+CPF1*CLTSH1
0061      T12=TF2+CLTSH2
0062      H12=HFG2+CPF2*CLTSH2
0063      CC 770 L=41,42
0064      CC 720 M=1,LIP
0065      IF(M .EQ. LIP) GC TC 704
0066      IF(A(L)-S(M))701,710,720
0067      701  IF(M .EQ. 1)GC TC 704
0068      RIFT = (A(L)-S(M-1))/(S(M)-S(M-1))
0069      HE(L)=H(M-1)+ RIFT*(H(M)-H(M-1))
0070      GC TC 770
0071      704  WRITE(6,8050)A(L),H(M)
0072      8050  FORMAT(1X,F10.5,' IS OUTSIDE ENTR TABLE,USE F = ',F10.5)
0073      710  HE(L)=H(M)
0074      CC TC 770
0075      720  CONTINUE
0076      770  CONTINUE
0077      HE1 = HE(41)
0078      HE2 = HE(42)
0079      IF(INSTOP .EQ. 4)GC TC 41
C
C *** COMPLETE BRINE SUPERHEATER ONE CUTLET
0080      ZZ5=CPF1*CLTSH1/HFG1
0081      TB1P=(TB1+ZZ5*(TF1+CLTPP1))/(1.+ZZ5)
0082      10  IF(XRAT.LT. 0.0) GC TC 40
0083      ICK1=C
0084      ICK2=C
C *** ASSUME PINCH POINT ONE CONTRCLS
0085      TB1P=(TB1+ZZ5*(TF1+CLTPP1))/(1.+ZZ5)
0086      TB3=TF1+CLTPP1-CPF5/HFG1*(TF1-TF2)*(TB1P-TF1-CLTPP1)
C
C *** COMPUTE PINCH POINT TWO
0087      ZZ6=HFG2/HFG1*XRAT
0088      ZZ7=CPF2*CLTSH2/HFG1*XRAT
0089      CLTPP4=TB3-(ZZ6+ZZ7)*(TB1P-TF1-CLTPP1)-TF2
C      CHECK FOR CONTRCL
0090      IF(DLTPP4-CLTPP2) 20,22,22
C      PINCH POINT TWO CONTRCLS

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0091      20 ZZ1=CPF*(TF1-TF2)
0092      ZZ2=XRAT*(CPF2*CLTSH2+HFG2)
0093      ZZ3=HFG1+CPF1*CLTSH1
0094      ZZ11=ZZ1+ZZ2+ZZ3
0095      ZZ22=ZZ1+ZZ2

C
C *** CCNPLTE PINCH POINT ONE
0096      CLTPP1=TB1+ZZ22/ZZ11+ZZ3/ZZ11*(TF2+CLTPP2)-TF1
0097      ICK2=ICK2+1
0098      TB1P=(TB1+ZZ5*(TF1+CLTPP1))/(1.+ZZ5)
0099      GO TO 40

C
C 22 SET PINCH POINT TWO
0100      CLTFF2=CLTPP4
0101      ICK1=ICK1+1
C**** SOLVE EQ. 1 (RATIC1=WF1/WB)
0102      40 RATIC1=(TB1P-TF1-CLTPP1)*CPB/HFG1
0103      ICT5=C

C
C**** SOLVE EQUATION 2
C *** SOLVE FOR BRINE OUTLET PREHEATER ONE
0104      TB2=(TF1+CLTPP1)-(CPF5*RATIC1*(TF1-TF2)/CPB)

C
C *** SOLVE FOR BRINE INLET BOILER TWO
0105      ZZ5=CPF2*CLTSH2/HFG2
0106      TB3P=(TB3+ZZ9*(TF2+CLTPP2))/(1.+ZZ9)
0107      CLTPP1M=TB1P-TF1
0108      CLTPP2M=TB3P-TF2

C
C**** SOLVE EQUATION 3 (WF2/WF1=RATIC2)
0109      RATIC2=HFG1/HFG2*((TB3P-TF2-CLTPP2)/(TB1P-TF1-CLTPP1))
0110      IF(XRAT.EQ.0.0) RATIC2 = 0.0

C
C IF((H1.EQ. 0.0).OR. (H2.EQ. 0.0)) GO TO 42
0111      41 H1TC=H11-NGT1*(H11-H1)
0112      HG2TC=H12-NGT2*(H12-H2)
0113      GO TO 43
0114      42 CPFF=(CPF1+CPF3)/2.
0115      CPFH=(CPF2+CPF3)/2.
0116      H1TC=H11-NGT1*CPFF*(H11+460.)*(1.-(PCCND/PP1)**((XK-1.)/XK))
0117      H2TC=H12-NGT2*CPFH*(H12+460.)*(1.-(PCCND/PP1)**((XK-1.)/XK))
0118      C**** SOLVE EQUATION 7 FOR CT1
0119      43 CT1=NEL*(H11-HG1TC)

C
C**** SOLVE EQUATION 8 FOR CPW1
0120      CPW1=0.185*VFCCND*(FF1-PCCND +CLPF1)/NP

C
C**** SOLVE EQUATION 9 FOR COW1
0121      DELHR1=H1TC-CCNDH-CPF3*(CTRG-DLTSC)
0122      IF(DELHR1 .LE. 0.0) GO TO 44
0123      IF( LRG1 .LE. 0.0) GO TO 44
0124      COW1 = 0.263*(HFCND+CPF3*CTRG)/(CPCW*(TCWC-TCWIN))
0125      DELR1=DELHR1
0126      GO TO 45
0127      44 COW1 = 0.263*(DELHR1+HFCND+CPF4*DLTSC)/(CPCW*(TCWC-TCWIN))
0128      CTRG1=DLTSC
0129      DELR1=0.0

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      C
      C**** SOLVE EQUATION 10 FOR CNET1
0130      45 CNET1=CT1-QPW1-QCW1
      C
      C**** SOLVE EQUATION 11 FOR CT2
0131      QT2=NEL*(HI2-HG2TC)
      C
      C**** SOLVE EQUATION 12 FOR CPW2
0132      CPW2=0.185*VFCOND*(PF2-PCCND+DLPF2)/NP
      C
      C**** SOLVE EQUATION 13 FOR CCW2
0133      DELHR2=HG2TC-CCNCH-CPF3*(CTRG-DLTSC)
0134      IF(DELHR2 .LE. 0.0) GO TO 46
0135      IF(URG2 .LE. 0.0) GO TO 46
0136      CCW2 = 0.263*(FFCOND+CPF3*CTRG)/(CPCW*(TCW0-TCWIN))
0137      DELR2=DELHR2
0138      GO TO 47
0139      46 CCW2 = 0.263*(DELHR2+FFCOND+CPF4*DLTSC)/(CPCW*(TCW0-TCWIN))
0140      CTRG2=DLTSC
0141      DELR2=0.0
      C
      C**** SOLVE EQUATION 14 FOR CNET2
0142      47 CNET2=CT2-CPW2-CCW2
0143      IF(NSTOP .NE. 4)GO TO 76
      C
      C**** FIND TEMPERATURES OF THE FLUID IN THE HEATER
0144      CELEN = (HFG1 - DELR1)/10.
0145      ENTH(1) = H11
0146      DO 50 K=2,10
0147      ENTH(K) = ENTH(K-1) - CELEN
0148      50 CONTINUE
0149      TEMP(1) = TF1
0150      DO 59 K=2,10
0151      DO 58 I=1,NUN
0152      IF(I .EQ. NUN)GO TO 54
0153      IF(ENTH(K)-XENTH(I))52,57,58
0154      52 IF(I .EQ. 1)GO TO 55
0155      RASH = (ENTH(K) - XENTH(I-1))/(XENTH(I) - XENTH(I-1))
0156      TEMP(K) = T(I-1) + RASH*(T(I)-T(I-1))
0157      GO TO 59
0158      54 TEMP(K) = TEMP(K-1)-CELEN/CPF
0159      WRITE(6,8000)ENTH(K),TEMP(K)
0160      8000 FORMAT(1X,F10.5,' IS ABOVE CRITICAL,USE COMPUTED TEMP ',F10.5)
0161      GO TO 59
0162      55 WRITE(6,8001)ENTH(K),T(1)
0163      8001 FORMAT(1X,F10.5,' IS BELOW TABLE,USE LOWEST TEMP ',F10.5)
0164      57 TEMP(K) = T(I)
0165      GO TO 59
0166      58 CONTINUE
0167      59 CONTINUE
      C
      C**** COMPUTE BRINE TEMPERATURES WITH FIRST GUESS OF RATIO1 = 1.0
0168      MUC = 0
0169      RATIO1 = 1.0
0170      60 CBRIN = RATIO1*CELEN
0171      TERIN(1) = T(1)

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0172      CC 62 I=2,10
0173      TBRIN(I) = TBRIN(I-1) - CBRIN
0174      62 CONTINUE
C
C**** CHECK FOR TEMPERATURE OVERLAP
0175      CC 70 I=1,10
0176      SCPP(I)=TBRIN(I)-TEMP(I)
0177      IF(SCPP(I)-CLTPP1)64,70,70
0178      64 IF(MLC .GT. 0)GO TO 75
0179      RATIC1 = RATIC1-CRATIC
0180      GO TO 60
0181      70 CONTINUE
0182      T85 = TBRIN(10)
0183      CELRG = CELR1
0184      TRGC = TCOND - CLTSC + DELRG/CPF4
0185      IF((T85 - 10.) .GT. TRGC)GO TO 71
0186      RATIC1 = RATIC1 - CRATIC
0187      GO TO 60
0188      71 RATIC1 = RATIC1 + URATIC
0189      IF(MLC .GT. 10)GO TO 75
0190      MLC = MLC + 1
0191      CC 72 K=1,10
0192      TBRINS(K) = TBRIN(K)
0193      72 CONTINUE
0194      GO TO 60
0195      75 T85 = TBRINS(10)
C
C**** SOLVE EQLATICA 4A FOR QNET/WB=RATIC3
0196      76 RATIC3=RATIC1*(QNET1+RATIO2*QNET2)-1.922496
C**** SOLVE EQLATICA 4B FOR PNET/WB=RATIC4
0197      RATIC4=RATIC3/3.413
0198      WRITE(6,SCC0)RATIO1,I1,CONDH,HFCCND,CPF,DTRG1,CELR1,RATIO2,HI:
1CTRG2, CELR2,CPB
0199      SCC0 FORMAT(1X,12F10.5)
0200      IF(NSTOP .NE. 4)GO TO 78
0201      T81T85 = T81 - T85
0202      GO TO 79
C
C**** SOLVE EQUATION 5 FOR T81-T85=T81T85
0203      78 T81T85=RATIC1*((I1-CCNCH)+FCOND+CPF4*DTRG1-CELR1+RATIO2*((HI
1NDH)+HFCCND+CPF4*CTRG2-CELR2))/CPB
C
C**** SOLVE EQUATION 6 FOR EFF
0204      79 EFF=(RATIC3/(CPB*T81T85))*100.0
C
C**** SOLVE EQUATION 15 T=RL 23
0205      WB=PNET*1.E6/RATIC4
0206      WF1=RATIC1*WB
0207      WF2=RATIC2*WF1
0208      CBF= 1.922496*WB/3.413E6
0209      CCh=(WF1*QCh1+WF2*QCh2)/3.413E6
0210      QFP=(WF1*QPh1+WF2*QPh2)/3.413E6
0211      ACh=WB/7.51E05 + 0.55
0212      ACEWR=ACh/2
0213      PGROSS=(WF1*QT1+WF2*CT2)/3.413E6
0214      PNTPCR=PNET/PGROSS

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0215      PNPWEL=PNET*WBW/WB
0216      IF(INSTOP .NE. 4)GO TO 87
0217      APF1 = 0.0
0218      AB2 = 0.0
0219      APF2 = 0.0
0220      ASF1 = 0.0
0221      ASF2 = 0.0
0222      ASF = 0.0
0223      A2 = 0.0
0224      CHEAT = WB*TB1TB5
0225      AB1 = CHEAT*ALCG((TB1-TF1)/(TB5-TRGC))/UBOIL1/(TB1TB5-TF1+TRGO)
0226      A1 = AB1
0227      ASLM = A1
0228      GC TC 95
0229      87 AB1=WB*CPB*ALCG((TB1P-TF1)/DLTPP1)/UBOIL1
0230      IF((TF1-TF2) .LE. 0.0) GC TO 88
0231      APF1=WB*CPB*ALCG((TB3-TF2)/DLTPP1)/(UPH1*((TF1-TF2)/(TF1+DLTPP1-
1      TB3)-1.0))
0232      GC TC 89
0233      88 APF1=0.0
0234      89 A1=AB1+APF1
C**** SOLVE EQLATIONS 24THRU 32
0235      AB2=WB*CPB*ALOG((TB3P-TF2)/DLTPP2)/UBOIL2
0236      IF(DLTSH1 .LE. 0.0) GC TC 90
0237      ASF1=WF1*CPF1*DLTSH1/LSH1/(TB1P-TB1+DLTSH1)*ALCG((TB1P-TF1)/(TB1-T
111))
0238      GC TC 91
0239      90 ASF1=0.0
0240      91 IF(DLTSH2 .LE. 0.0) GC TO 92
0241      ASF2=WF2*CPF2*DLTSH2/USF2/(TB3P-TB3+DLTSH2)*ALCG((TB3P-TF2)/(TB3-T
112))
0242      GC TC 93
0243      92 ASF2=0.0
0244      93 ASF=ASF1+ASF2
0245      CELRG=(DELR1+RATIC2*(CELR2))/(1.+RATIC2)
0246      TRGC=TCOCD-CLTSC+CELRG/CPF4
0247      TE5=-(TB1TB5-TB1)
0248      IF((TB5-10.) .GT. TRGC) GC TC 94
0249      CTFP1N=DLTPP1+(CTFP1M-CLTPP1)/3.
0250      CTFP2N=DLTPP2+(CTFP2M-CLTPP2)/3.
0251      IF(ICK1 .GE. 1) CLTFP1=CTFP1N
0252      IF(ICK2 .GE. 1) CLTPP2=CTFP2N
0253      IF(ICTM .GT. 10) GC TC 94
0254      ICTM=ICTM+1
0255      GC TC 10
0256      94 APF2=WB *CPB*ALOG((TB5-TRGO)/DLTPP2)/(UPH2*((TF2-TRGO)/(TF2+DLTPP2
1      -TB5)-1))
0257      IF(( ICTM .GE. 10) .AND. (ICK1 .GE. 1)) WRITE(6,2130)
0258      2130 FORMAT(1X,' TEMP BOILER ONE IS TOO SMALL ')
0259      IF((ICTM .GE. 10) .AND. (ICK2 .GE. 1)) WRITE(6,2140)
0260      2140 FORMAT(1X,' TEMP BOILER TWO IS TOO SMALL ')
0261      A2=AB2+APF2
0262      ASLM=A1+A2
0263      WF11=WF1
0264      WF22=WF2
0265      95 IF((LRG1 .LE. 0.0) .AND. (LRG2 .LE. 0.0)) GC TC 96

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0266      QRG=WF1*DELRI+WF2*DELRI
0267      IF(URGI .LE. 0.0) WF1=0.0
0268      IF(URG2 .LE. 0.0) WF2=0.0
0269      96 HMI=(WF1*HG1TC+WF2*HG2TC)/(WF1+WF2)
0270      TMIX=HMI*.7250+ICG7.
0271      CCCNC=WF1*(DELHR1+CPF3*DTRG+HFCND-DELRI)+WF2*(DELHR2+CPF3*DTRG+HFCND-DELRI)
0272      TFCC=TCND-DLTSC
0273      CCNDTC=TFCC-TCWIN
0274      CCNDTI=TMIX-TCWC
0275      AC=QCOND/(UCCND*(CCNDTC-CCNDTI)/(ALOG(CCNDTC/CCNDTI)))
0276      IF((URGI .LE. 0.0) .AND. (URG2 .LE. 0.0)) GO TO 98
0277      IF((TMIX-TRGC) .LE. (.5*CTRG)) TMIX=TRGC+.5*CTRG
0278      ARG=QRG/((URGI+URG2)/2.*(CTRG+DLTSC-TMIX+TRGC)/ALOG((DTRG+DLTSC)/(TMIX-TRGC)))
0279      IF((URGI .LE. 0.0) .OR. (URG2 .LE. 0.0)) ARG=ARG*2.
0280      98 HTA=A1+A2+AC+ARG+ASH
0281      WCH=CCCNC/(CPCW*(TCWC-TCWIN))
0282      GPM=WCH/500.0
0283      ACCH=WCH/7.51E05+.99
0284      NOCHR=NOCH/2

C
C      OVERALL EFF OF ENERGY SUPPLIED
0285      CALN=RATIC3/CPB/(TB1-TCWIN)*100.
0286      GO TO (129,120,121,122,123,124,125,126,127,129) , IFLG
0287      120 ACK = HTA
0288      GC TC 128
0289      121 ACK = ASLM+ASH
0290      GC TC 128
0291      122 ACK = AB1
0292      GC TC 128
0293      123 ACK = AB2
0294      GC TC 128
0295      124 ACK = APH1
0296      GC TC 128
0297      125 ACK = APH2
0298      GC TC 128
0299      126 ACK = AC
0300      GC TC 128
0301      127 ACK = ARG
0302      GC TC 128
0303      128 IF((ABS(ACK-AHTX)/AHTX) .LE. 0.001)GC TO 129
0304      PNETS = PNET
0305      FNET = PNET*(1.+(AHTX-ACK)/AHTX*(1.4-ICT3/10.))
0306      IF(PNET .LE. 0.0) PNET = .8*PNETS
0307      ICT3 = ICT3 + 1
0308      IF(ICT3 .GT. 11)GC TO 129
0309      GC TC 79
0310      129 CONTINUE
0311      IF( IFLG .NE. 10) GC TO 500
0312      AIT=A1+A2
0313      AITI=ATH1+ATH2
0314      IF((ABS(AIT-AITI)/AIT) .LE. .01) GO TO 190
0315      ICT5=ICT5+1
0316      PNETS=PNET
0317      FNET=PNET*(1.+(AIT-AITI)/AIT*(1.4-ICT5/10.))

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0318      IF(PNET .LE. 0.0) PNET=.8*PNETS
0319      WRITE(6,3000) AIT,AIT1,PNET,PNETS,ICT5
0320      IF(ICT5 .GE. 10) GC TC 190
0321      GC TC 79
0322      190 IF(ICK2 .GE. 1) GO TO 200
0323      IF((ABS(ABCL1-AB1)/ABCL1) .LE. .01) GO TO 200
0324      ICT3=ICT3+1
0325      CLTPPN=((AB1-ABCL1)/ABCL1+1.)*CLTPP1
0326      WRITE(6,3000) AB1,ABCL1,CLTPPN,CLTPP1,ICT3
0327      IF(ICT3 .GE. 10) GC TC 200
0328      3000 FORMAT(1X,4(3X,F10.1),15)
0329      CLTPP1=CLTPPN
0330      IF(CLTPP1 .LE. .0) CLTPP1=.001
0331      IF(CLTPP1 .GE. 70.) CLTPP1=70.
0332      GC TC 203
0333      200 LCPC=1
0334      203 IF(ICK1 .GE. 1) GC TC 210
0335      IF((ABS(ABCL2-AB2)/ABCL2) .LE. .01) GO TO 210
0336      ICT4=ICT4+1
0337      CLTP2N=((AB2-ABCL2)/ABCL2+1.)*CLTPP2
0338      WRITE(6,3000) AB2,ABCL2,CLTP2N,CLTPP2,ICT4
0339      IF(ICT4 .GE. 10) GC TC 210
0340      IF(CLTPP2 .LE. .0) CLTPP2=.001
0341      IF(CLTPP2 .GE. 70.) CLTPP2=70.
0342      CLTPP2=CLTP2N
0343      GC TC 10
0344      210 IF(LCPC .LT. 1) GC TC 10
0345      230 AD=A1/A2
0346      ACI=ATH11/ATH21
0347      IF((ABS(AC-ACI)/ACI) .LE. .01) GO TO 240
0348      ICT6=ICT6+1
0349      XRAT=RATIO2
0350      XRATS=XRAT
0351      XRAT=((AC-ACI)/ACI+1.)*XRAT
0352      IF(XRAT .LE. 0.0) XRAT=.01
0353      WRITE(6,3000) AC,ACI,XRAT,XRATS,ICT6
0354      IF(XRAT .GE. 4.) XRAT=4.
0355      IF(ICT6 .GE. 10) GC TC 240
0356      ICT3=0
0357      ICT4=0
0358      GC TC 10
0359      240 CONTINUE
0360      500 WRITE(6,803)
0361      803 FORMAT(1H0,10X,'CASE RESULTS')
0362      WRITE(6,850)RATIO3
0363      850 FORMAT(1H0,'QNET/WB (BTU/LB)',1PE10.4,5X,'EQUATION 4A')
0364      WRITE(6,851)RATIC4
0365      851 FORMAT(1H0,'PNET/WB (WATT-HR/LB)',1PE12.4,5X,'EQUATION 4B')
0366      WRITE(6,852)TB1TB5
0367      852 FORMAT(1H0,'DELTA T BRINE (DEGREES F)',1PE12.4,5X,'EQUATION 5')
0368      WRITE(6,853)EFF
0369      853 FORMAT(1H0,'THERMAL EFFICIENCY NET (%)',1PE12.4,5X,'EQUATION 6')
0370      WRITE(6,854)PNPWEL
0371      854 FORMAT(1H0,'PNET/HELL (LB/HR)',1PE12.4,5X,'EQUATION 20')
0372      WRITE(6,855)RATIC1
0373      855 FORMAT(1H0,'WF1/WB',1PE12.4,5X,'EQUATION 1')

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0374      WRITE(6,856)RATIO2
0375      856 FORMAT(1H0,'WF2/WF1',1PE12.4,5X,'EQUATION 3')
0376      WRITE(6,857)TB3
0377      857 FORMAT(1H0,'TB3 (DEGREES F)',1PE12.4,5X,'EQUATION 2')
0378      WRITE(6,858)QT1
0379      858 FORMAT(1H0,'QT1 (BTU/HR)',1PE12.4,5X,'EQUATION 7')
0380      WRITE(6,859)QPN1
0381      859 FORMAT(1H0,'QPN1 (BTU/LB)',1PE12.4,5X,'EQUATION 8')
0382      WRITE(6,860)CCW1
0383      860 FORMAT(1H0,'CCW1 (BTU/LB)',1PE12.4,5X,'EQUATION 9')
0384      WRITE(6,861)QNET1
0385      861 FORMAT(1H0,'QNET1 (BTU/LB)',1PE12.4,5X,'EQUATION 10')
0386      WRITE(6,862)CT2
0387      862 FORMAT(1H0,'CT2',1PE12.4,5X,'EQUATION 11')
0388      WRITE(6,863)QPN2
0389      863 FORMAT(1H0,'QPN2',1PE12.4,5X,'EQUATION 12')
0390      WRITE(6,864)CCW2
0391      864 FORMAT(1H0,'CCW2',1PE12.4,5X,'EQUATION 13')
0392      WRITE(6,865)QNET2
0393      865 FORMAT(1H0,'QNET2 (BTU/LB)',1PE12.4,5X,'EQUATION 14')
0394      WRITE(6,866)PNET
0395      866 FORMAT(1H0,'PNET (MEGAWATTS)',1PE12.4,5X,'INPUT')
0396      WRITE(6,867)PGROSS
0397      867 FORMAT(1H0,'PGROSS (MEGAWATTS)',1PE12.4,5X,'EQUATION 18')
0398      WRITE(6,868)PATPCR
0399      868 FORMAT(1H0,'PNET/PGROSS',1PE12.4,5X,'EQUATION 19')
0400      WRITE(6,869)WB
0401      869 FORMAT(1H0,'WB (LB/HR)',1PE12.4,5X,'EQUATION 16')
0402      WRITE(6,870)WF2
0403      870 FORMAT(1H0,'WF2 (LB/HR)',1PE12.4,5X,'EQUATION 17')
0404      WRITE(6,871)TF1
0405      871 FORMAT(1H0,'TF1 (DEGREES F)',1PE12.4,5X,'INPUT')
0406      WRITE(6,872)TF2
0407      872 FORMAT(1H0,'TF2 (DEGREES F)',1PE12.4,5X,'INPUT')
0408      WRITE(6,873)AB1
0409      873 FORMAT(1H0,'AB1 (FT**2)',1PE12.4,5X,'EQUATION 21')
0410      WRITE(6,874)APH1
0411      874 FORMAT(1H0,'APH1 (FT**2)',1PE12.4,5X,'EQUATION 22')
0412      WRITE(6,875)A1
0413      875 FORMAT(1H0,'A1 (FT**2)',1PE12.4,5X,'EQUATION 23')
0414      WRITE(6,876)AB2
0415      876 FORMAT(1H0,'AB2 (FT**2)',1PE12.4,5X,'EQUATION 24')
0416      WRITE(6,877)APH2
0417      877 FORMAT(1H0,'APH2 (FT**2)',1PE12.4,5X,'EQUATION 25')
0418      WRITE(6,878)A2
0419      878 FORMAT(1H0,'A2 (FT**2)',1PE12.4,5X,'EQUATION 26')
0420      WRITE(6,879)ASUM
0421      879 FORMAT(1H0,'ASUM (FT**2)',1PE12.4,5X,'EQUATION 27')
0422      WRITE(6,880)AC
0423      880 FORMAT(1H0,'ACCNC (FT**2)',1PE12.4,5X,'EQUATION 29')
0424      WRITE(6,881)HTA
0425      881 FORMAT(1H0,'HTA (FT**2)',1PE12.4,5X,'EQUATION 30')
0426      WRITE(6,882)WCH
0427      882 FORMAT(1H0,'WCH (LB/HR)',1PE12.4,5X,'EQUATION 31')
0428      WRITE(6,883)GPM
0429      883 FORMAT(1H0,'GPM (GAL/MIN)',1PE12.4,5X,'EQUATION 32')

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0430      WRITE(6,884)QCOND
0431      884 FORMAT(1H0,'QCOND (BTU/HR)',1PE12.4,5X,'EQUATION 28')
0432      WRITE(6,885) WF1
0433      885 FORMAT(1H0,' WF1 (LB/HR) ',1PE12.4,5X,' EQUATION 16 ')
0434      WRITE(6,886) TRGC
0435      886 FORMAT(1H0,' TEMP REG CUT ',1PE12.4,5X,' DEG F ')
0436      WRITE(6,887) TB5
0437      887 FORMAT(1H0,' TEMP BR CT T ',1PE12.4,5X,' DEG F ')
0438      WRITE(6,888) DELRG
0439      888 FORMAT(1H0,' DEL ENTH RGT ',1PE12.4,5X,' BTU/LB ')
0440      WRITE(6,889) ARG
0441      889 FORMAT(1H0,' REG AREA ',1PE12.4,5X,' FT SQ ')
0442      WRITE(6,890) CLTPP1
0443      890 FORMAT(1H0,' PIN PT 1 ',1PE12.4,5X,' DEG F ')
0444      WRITE(6,891) CLTPP2
0445      891 FORMAT(1H0,' PIN PT 2 ',1PE12.4,5X,' DEG F ')
0446      WRITE(6,892) TB1P
0447      892 FORMAT(1H0,'BRINE SUPERHEAT 1 EXIT',1PE12.4,5X,' DEG F ')
0448      WRITE(6,893) TB3P
0449      893 FORMAT(1H0,'BRINE SUPERHEAT 2 EXIT',1PE12.4,5X,' DEG F ')
0450      WRITE(6,894) HI1
0451      894 FORMAT(1H0,' TUR ONE ENTH IN ',1PE12.4,5X,' BTU/LB ')
0452      WRITE(6,895) HI2
0453      895 FORMAT(1H0,' TUR TWO ENTH IN ',1PE12.4,5X,' BTU/LB ')
0454      WRITE(6,896) ASH1
0455      896 FORMAT(1H0,' AREA SUPHEAT ONE',1PE12.4,5X,' FTSQ ')
0456      WRITE(6,897) ASH2
0457      897 FORMAT(1H0,' AREA SUPHEAT TWO',1PE12.4,5X,' FTSQ ')
0458      WRITE(6,898) CALN
0459      898 FORMAT(1H0,' OVERALL EFFECTIVE USE CF ENERGY SUPPLIED % ',1PE12.4)
0460      WRITE(6,899)QBP
0461      899 FORMAT(1H0,' BRINE PUMPING POWER ',1PE12.4,5X,' MW ')
0462      WRITE(6,900)CCW
0463      900 FORMAT(1H0,' CONDENSING WATER PUMPING POWER ',1PE12.4,5X,' MW ')
0464      WRITE(6,901)CFP
0465      901 FORMAT(1H0,' WORKING FLUID PUMPING POWER ',1PE12.4,5X,' MW ')
0466      WRITE(6,903)TMIX
0467      903 FORMAT(1H0,' CONDENSER INLET TEMPERATURE ',1PE12.4,5X,' DEGREES F'
1)
0468      WRITE(6,902)NCW
0469      902 FORMAT(1H0,' NUMBER CF BRINE SUPPLY WELLS ',I5)
0470      WRITE(6,904)NCBW
0471      904 FORMAT(1H0,' NUMBER CF BRINE REINJECTION WELLS',I5)
0472      WRITE(6,905)NCCW
0473      905 FORMAT(1H0,' NUMBER CF COOLING WATER SUPPLY WELLS',I5)
0474      WRITE(6,906)NCCWR
0475      906 FORMAT(1H0,' NUMBER CF COOLING WATER REINJECTION WELLS',I5)
0476      WRITE(6,998)
0477      998 FORMAT(6X,' TEMP BR INLET      TP BR SPH 1 OT      TP BR PRH 1 OT
1 TP BR SPH 2 OT      TP BR CUTLET      BRINE FLOW RATE ')
0478      WRITE(6,1000) TB1,TB1P,TB3,TB3P,TB5,WB
0479      1000 FORMAT(1X,6(2X,E15.4,3X))
0480      WRITE(6,1200)
0481      1200 FORMAT(6X,' TP TUR1 INLET      EN TUR1 INLET      TP BOILER ONE
1 FLOW RT TR ONE      EN EST BOIL 1      EN ACT EX TUR1 ')
0482      WRITE(6,1000) TI1,HI1,TF1,WF1,HG1,HG1TC

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0483      WRITE(6,1300)
0484      1300 FORMAT(6X,' TP TUR2 INLET      EN TUR2 INLET      TP BOILER TWO
1      FLW RT TR TWO      EN EXT BCIL 2      EN ACT EX TUR2 ')
0485      WRITE(6,1000) TI2,HI2,TF2,WF2,HG2,HG2TC
0486      WRITE(6,1400)
0487      1400 FORMAT(6X,' AREA BCILER 1      AREA BCILER 2      AREA PREHET 1
1      AREA PREHET 2      AREA CONDENSER      AREA REGENERATOR ')
0488      WRITE(6,1000) AB1,AB2,APH1,APH2,AC,ARG
0489      WRITE(6,1500)
0490      1500 FORMAT(6X,' AREA SUPHET 1      AREA SUPHET 2      CELT PIN PT 1
1      CELT PIN PT 2      TEMP REG OUT      FLOW RATE CCCL ')
0491      WRITE(6,1000) ASH1,ASH2,CLTPP1,CLTPP2,TRGO,WCV
0492      DO 105 I=1,50
0493      A(I)=C(I)
0494      105 CONTINUE
0495      IF(1-ASTCP)5,106,106
0496      106 CONTINUE
0497      END
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