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ANCR-1295

DATE PUBLISHED — JANUARY 1976

UC-66

IDAHO NATIONAL ENGINEERING LABORATORY

GEOTHERMAL STEAM PLANT
MODELING AND POWER
TRADEOFF STUDIES

MASTER

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Printed in the United States of America
Available from
National Technical Information Service
U. S. Department of Commerce
5285 Port Royal Road
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Price: Printed Copy \$5.50; Microfiche \$2.25

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GEOTHERMAL STEAM PLANT
MODELING AND POWER
TRADEOFF STUDIES

by

C. J. Shaffer

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SUBMITTED TO THE IDAHO OPERATIONS OFFICE AND THE DIVISION OF GEOTHERMAL RESEARCH
U. S. ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION

AEROJET NUCLEAR COMPANY

Date Published - January 1976

PREPARED FOR THE U. S. ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
IDAHO OPERATIONS OFFICE
UNDER CONTRACT NO. E(10-1)-1375

Contract Technical Coordinator - J. L. Griffith

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ABSTRACT

A computer model of an open loop flashed steam plant for producing electrical power from a hot geothermal water supply is documented. The model has the capability of tracing the dissolved gases, N_2 and CO_2 , throughout the plant. Sensitivity studies were performed on several of the geothermal source, design, and atmospheric parameters using the source conditions found in the Raft River Valley as a base case.

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NOMENCLATURE

Variables

a	drift factor
A	cross sectional area
C	sonic velocity
C _p	constant pressure specific heat
C _v	constant volume specific heat
D	diameter
f	Moody single-phase friction factor
g	gravitational acceleration, 32.2 ft/sec ²
g _c	32.2 lbm-ft/lbf-sec ²
h	enthalpy
Δh	enthalpy difference
k	head loss coefficient
k	ratio of specific heats
K	solubility constant
L	length
M	mach number
M	molecular weight
P	pressure
ΔP	pressure difference
Pow	power
R	gas constant
s	entropy
Δs	entropy difference
T	temperature
ΔT	temperature difference
V	velocity
W	mass flow rate
x	quality of gas
Y	mass fraction of constituent in a particular phase

NOMENCLATURE (continued)

Variables (continued)

η	efficiency
ρ	density
σ	ratio of flow rates in air ejector, 1 to 7
ϕ	ratio of velocities in air ejector, 4 to 3
ϕ	relative humidity
ω	specific humidity

Subscripts

a	air
atm	atmospheric
B	brake (mechanical)
c	carbon dioxide
f	saturated liquid
g	gas
G	generator or motor
i	inlet
I	nonreversibilities
m	mixture
n	nitrogen
o	outlet
s	source
sat	saturation
st	stagnation
T	total
v	water vapor
w	liquid water
1-7	state point numbers
*	critical or sonic

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I. INTRODUCTION

One method of producing electrical power from a hot geothermal brine is reducing the system pressure below the saturation pressure and driving a turbine with the resulting steam. The geothermal brine discovered in the Raft River Valley⁽¹⁾ with a temperature of around 300°F is one such brine.

A model of such a flashed steam electrical generating plant has been completed and sensitivity studies of the significant parameters were performed using the source conditions found in the Raft River Valley as a base case. The parameters studied include geothermal source, design, and atmospheric parameters. One significant capability of this model is the ability to track the dissolved gases, nitrogen, and carbon dioxide found in geothermal brines through the plant and include the effects of these gases upon power output. All studies were performed for a net electrical output of 10 Mw.

II. MODEL

1. General System Description

An open-loop steam-flash geothermal plant generates electrical power from a hot geothermal brine of water containing amounts of dissolved minerals and gases. The calculational model describes: (1) the primary system consisting of the flashers for steam generation from hot water, the turbine to drive the electrical generator, and a direct-contact condenser; (2) a system for removal of noncondensable gases from the condenser; and (3) a system to supply cooling water to the condensers and turbine-generator set. A system schematic of the model is shown in Figure 1.

Hot water is flashed to steam in the primary system by reducing the system pressure below the saturation pressure. This can be done in stages producing more than one steam flow at successively lower pressures. This model allows up to four flash stages. These steam flows are then fed into a multistage turbine model which is directly connected to a direct-contact condenser. The amounts of noncondensable gases are tracked through the flashers, turbine, and condenser in both the dissolved and free states. This model allows for nitrogen and/or carbon dioxide gases.

The noncondensable gases flowing through the turbine into the condenser must be removed to preserve the subatmospheric condenser pressure needed for plant operation. An air compressor is one method of removing the gases and is an option supplied in the model. A generally more practical method for high gas flow rates, however, is a two-stage air ejector system driven by steam from a preflasher. A preflasher is a flasher inserted into the system just prior to the flashers which drive the turbine. The advantage of a preflasher (as will be shown later) is that the noncondensable gases evolve from the brine rapidly once the system pressure is dropped below saturation pressure. Therefore, more of the noncondensable gases are used to drive the air ejectors and do not enter the turbine flow. The model strikes a balance between the noncondensable gases flowing through the turbine and the steam-noncondensable gas mixture driving the air ejectors. Since steam is used

C: Compressor

E1-2: Air Ejector

F1-4: Flasher/Separators

P1-10: Pumps

PF: Preflasher

R1-9: Hydraulic Resistances

Fig. 1 System schematic

to drive the air ejectors and a certain amount of vapor is carried along with the noncondensibles from the condenser, condensers are required after each air ejector stage as shown in Figure 1 to condense the steam. These condensers are of the direct-contact type and are called the inter and after condensers.

The cooling water system consists of a wet type cooling tower and the associated pumping and piping to supply cooling water to the condensers. A shallow aquifer well is included to supply makeup water to the cooling tower or if desired the entire coolant flow can be pumped up from the well instead of using the cooling tower.

2. Component Description

2.1 Flasher/Separator Model

The flasher is an orifice which throttles the fluid and allows the pressure to decrease causing water to flash to steam. The separator then separates the steam to drive a turbine or an air ejector. The flasher/separator is hereafter referred to as the flasher.

The flasher model applies to a geothermal brine flow of water and steam containing CO_2 and/or N_2 gases either in the free or dissolved states. Given the temperature and dissolved gas concentrations of a water flow, the conditions at any reduced pressure can be determined by assuming the following:

- (1) Thermal equilibrium
- (2) Solubility equilibrium
- (3) Adiabatic
- (4) Two-phase slip ratio of one
- (5) Enthalpy of dissolved gas equal to the enthalpy of free gas at the same temperature.

The flasher model involves system mass and energy balances, Henry's law of gas solubilities, Dalton's law of partial pressures, and the perfect gas law.

Henry's law which states that the quantity of gas dissolved is proportional to the partial pressure of that gas governs the gas solubility in water under equilibrium conditions. The proportionality constants used in Henry's law were determined experimentally and are temperature dependent^(2,3,4). Equilibrium gas evolution is then the difference in gas solubilities at different conditions. Equilibrium gas conditions may not actually exist in practical situations.

Gas evolution and steam flashing occur simultaneously as the pressure is decreased below the bubble formation pressure. There is some maximum pressure for which a gas bubble can exist depending upon the temperature and the noncondensable gas concentrations. Above this pressure, the water is subcooled and all gases are dissolved. Dalton's law which states that the sum of the partial pressures of each gas is equal to the total system pressure and the perfect gas law which relates the partial pressure to gas densities (or mole fractions) determines the relative amounts of CO₂, N₂, and vapor in the gas bubbles. The partial pressure of the vapor is the saturation pressure of water at the fluid temperature. The flash model involves the simultaneous solution of the following system of equations:

Energy balance:

$$x = \frac{(Y_{ws} C_{p_w} + Y_{ns} C_{p_n} + Y_{cs} C_{p_c})(T_s - T)}{Y_v (h_g - h_f)} \quad (1)$$

Mass balances:

$$x Y_{ng} + (1-x) Y_{nw} = Y_{ns} \quad (2)$$

$$x Y_{cg} + (1-x) Y_{cw} = Y_{cs} \quad (3)$$

$$Y_{ng} + Y_{cg} + Y_v = 1 \quad (4)$$

$$Y_{ws} + Y_{ns} + Y_{cs} = 1 \quad (5)$$

$$Y_w + Y_{nw} + Y_{cw} = 1 \quad (6)$$

Henry's Law:

$$P_n = K_n Y_{ng} \frac{M_w}{M_n} \quad (7)$$

$$P_c = K_c Y_{cg} \frac{M_w}{M_c} \quad (8)$$

Dalton's Law:

$$P_v + P_n + P_c = P_T \quad (9)$$

Perfect Gas Law:

$$\frac{P_v}{P_n} = \frac{Y_v}{Y_{ng}} \frac{M_n}{M_v} \quad (10)$$

$$\frac{P_v}{P_c} = \frac{Y_v}{Y_{cg}} \frac{M_c}{M_v} \quad (11)$$

The solution of this system of equations requires a double iterative numerical solution.

2.2 Turbine Model

The turbine model models a single unit of a variable number of stages (up to four) depending upon the number of flashes specified. The model is then broken down into two parts: (1) expansion, and (2) mixing of the flows from the last expansion stage and next inlet flow if two or more flashes are used. Figure 2 shows a two-stage turbine temperature-entropy diagram. The appropriate equations were modified to include the effects of the noncondensable gases throughout the turbine. The effects of the gases are negligible at low gas concentrations. The quality at state point 2 after an isentropic expansion is given by:

$$x_2 = \frac{x_1(Y_{c1}\Delta s_c + Y_{n1}\Delta s_n) + (1-x_1)s_{f1} + x_1Y_{v1}s_{g1} + x_1(1-Y_{v1})s_{g2} - s_{f2}}{s_{g2} - s_{f2}} \quad (12)$$

Fig. 2 Two-stage turbine temperature-entropy diagram.

State point 3 is determined from the enthalpy difference from points 1 to 3 as computed by:

$$\Delta h_{1-3} = \eta_1 \left[x_1 (Y_{c1} C_{pC} + Y_{n1} C_{pN}) \Delta T_{1-2} + (1-x_1) h_{f1} + x_1 Y_{v1} h_{g1} - (1-x_2) h_{f2} - x_2 Y_{v2} h_{g2} \right] \quad (13)$$

The quality at point 3 is given by:

$$x_3 = \frac{x_1 (Y_{c1} C_{pC} + Y_{n1} C_{pN}) \Delta T_{1-3} + (1-x_1) h_{f1} + x_1 Y_{v1} h_{g1} + x_1 (1-Y_{v1}) h_{g3} - \Delta h_{1-3} - h_{f3}}{h_{g3} - h_{f3}} \quad (14)$$

The quality of the mixed flows at point 5 is given by:

$$x_5 = \frac{U_2 h_3 + U_3 h_4 - U_2 x_3 (Y_{c3} C_{pC} + Y_{n3} C_{pN}) \Delta T_{5-3} - U_3 x_4 (Y_{c4} C_{pC} + Y_{n4} C_{pN}) \Delta T_{5-4} + U_1 h_{g5} - h_{f5}}{h_{g5} - h_{f5}} \quad (15)$$

$$\text{where } U_1 = U_2 x_3 (Y_{c3} + Y_{n3}) + U_3 x_4 (Y_{c4} + Y_{n4})$$

$$U_2 = \frac{W_3}{W_3 + W_4}$$

$$U_3 = \frac{W_4}{W_3 + W_4}$$

$$h_3 = (1-x_3) h_{f3} + x_3 Y_{v3} h_{g3}$$

$$h_4 = (1-x_4) h_{f4} + x_4 Y_{v4} h_{g4}$$

State points 6 and 7 and succeeding stages are determined using equations (12) through (15).

Figure 2 shows temperatures T_2 , T_3 , T_4 , and T_5 equal in magnitude, however, at high noncondensable gas concentrations, these temperatures are not necessarily equal. The temperatures are a function of the vapor partial pressure which in turn depends upon the noncondensable gas concentrations at each point. The sum of the partial pressures at each point must be equal to the system pressure. At low gas concentrations, these equations can be considerably simplified.

The turbine gross electrical power output is determined by:

$$P_{ow} = \eta_G \eta_B [W_1 \Delta h_{1-3} + (W_1 + W_4) \Delta h_{5-7} + \dots] \quad (16)$$

2.3 Condenser Model

The condenser model for the turbine, inter, and after condensers is of the direct-contact type. The condenser is a vessel in which cooling water and water vapor mix causing the vapor to condense. Noncondensable gases carried into the condenser by the vapor are drawn off at the vessel top by a vacuum. The mass flow rate of cooling water necessary to condense a given vapor flow rate is computed by:

$$W_{wi} = \frac{E_{fg} + E_{sc} + E_g}{Cp_w (T_{wo} - T_{wi})} \quad (17)$$

where

$$E_{fg} = (Y_{vi} x_{gi} W_{gi} - Y_{vo} W_{go}) (h_g - h_f)$$

$$E_{sc} = (W_{gi} - W_{go}) Cp_w (T_{gi} - T_{wo})$$

$$E_g = (Y_{vo} Cp_v + Y_{no} Cp_n + Y_{co} Cp_c) W_{go} (T_{gi} - T_{go})$$

In Equation (17), E_{fg} accounts for the latent heat energy necessary to condense the steam; E_{sc} accounts for the subcool energy necessary to cool the condensate to the outlet cooling water temperature, and E_g accounts for energy given up by the noncondensable gases and vapor carried out by the gases as the gases are cooled by the cooling water. The solution of Equation (17) requires the use of Dalton's Law and the perfect gas law.

2.4 Air Ejector Model

An air ejector consists of a nozzle, mixing region, and a diffuser as shown schematically in Figure 3. Steam is accelerated through the nozzle reducing the static pressure to the level of vacuum required to evacuate the condenser. After the high velocity steam mixes with the condenser flow, a diffuser decelerates the flow to a higher pressure.

Fig. 3 Air ejector single stage model

This model assumes single phase isentropic flow through the nozzle and diffuser but the nonreversibilities are accounted for in the mixing region using experimental data ⁽⁵⁾ reduced using these assumptions as shown in Figure 4. All flows through the ejector can be any mixture of steam, nitrogen, and carbon dioxide. The inlet and outlet flows are assumed to have negligible velocities. The model does not account for condensation in the air ejector.

The following system of equations are solved in succeeding order to finally calculate the ratio of the air ejector driving steam flow to the flow evacuated from the condenser which then is used to compute mass flow rates and cross sectional areas. Equations (18) to (20) and (22) to (24) are isentropic flow equations; Equation (21) is a simple energy balance to determine the static temperature after mixing and Equation (25) relates the mixing flow ratio to the mixing velocity ratio either by assuming ideal reversible flow or by using the experimental data ⁽⁵⁾ in Figure 4 for non-reversible flow. Equations (21) to (25) require an iterative solution iterating on the mixing flow ratio. Simply speaking, the nozzle exit velocity and the diffuser entrance velocity are computed using isentropic flow theory but the thermoproperties require the temperatures and gas compositions which necessitates an iterative solution.

$$V_3 = \left[\left(\frac{2}{k_1 - 1} \right) \left(\frac{144 P_1 g_c}{\rho_1} \right) \left[1 - \left(\frac{P_3}{P_1} \right)^{\left(\frac{k_1 - 1}{k_1} \right)} \right] \right]^{\frac{1}{2}} \quad (18)$$

$$M_3 = \left[\left(\frac{2}{k_1 - 1} \right) \left[\left(\frac{P_1}{P_3} \right)^{\left(\frac{k_1 - 1}{k_1} \right)} - 1 \right] \right]^{\frac{1}{2}} \quad (19)$$

$$T_3 = \frac{T_1 + 460}{\left[1 + \left(\frac{k_1 - 1}{2} \right) M_3^2 \right]} - 460 \quad (20)$$

$$T_4 = \frac{\left(\frac{\sigma C p_3}{C p_7} \right) T_3 + T_7}{1 + \frac{\sigma C p_3}{C p_7}} \quad (21)$$

Fig. 4 Experimental air ejector non-reversibilities data

$$M_4 = \left[\left(\frac{2}{k_4 - 1} \right) \left[\left(\frac{P_6}{P_4} \right)^{\left(\frac{k_4 - 1}{k_4} \right)} - 1 \right] \right]^{\frac{1}{2}} \quad (22)$$

$$T_6 = (T_4 + 460) \left[1 + \left(\frac{k_4 - 1}{2} \right) M_4^2 \right] - 460 \quad (23)$$

$$V_4 = \left[\left(\frac{2}{k_4 - 1} \right) \left(\frac{P_6}{P_4} \right) \left[1 - \left(\frac{P_4}{P_6} \right)^{\left(\frac{k_4 - 1}{k_4} \right)} \right] \right]^{\frac{1}{2}} \quad (24)$$

$$\sigma = \begin{cases} \frac{\phi}{1 - \phi} ; \text{ ideal flow} \\ \text{data; actual flow} \end{cases} \quad (25)$$

Critical or sonic data including pressures, temperatures, densities, and sonic velocities are computed using the following equations:

$$P_* = P_{st} \left(\frac{2}{k+1} \right)^{\left(\frac{k}{k-1} \right)} \quad (26)$$

$$T_* = \left(\frac{2}{k+1} \right) (T_{st} + 460) - 460 \quad (27)$$

$$\rho_* = \rho_{st} \left(\frac{2}{k+1} \right)^{\left(\frac{1}{k-1} \right)} \quad (28)$$

$$C_* = \left[k g_c R (T_* + 460) \right]^{\frac{1}{2}} \quad (29)$$

Ratios of cross sectional areas to the critical or sonic areas are computed using the following equation:

$$\frac{A}{A_*} = \frac{1}{M} \left[\frac{2 + (k-1) M^2}{2 + (k-1)} \right]^{\left(\frac{k+1}{2(k-1)} \right)} \quad (30)$$

2.5 Cooling Tower Model

This model assumes the wet type of cooling tower in which heat is dissipated by evaporative cooling. Cooling water is sprayed into an air flow circulated through the tower by electrically driven fans. The mass flow rate of air required to cool a given flow rate of cooling water is computed using energy and mass balances, Dalton's law, and the perfect gas law, as given by the following equations:

$$W_a = \frac{W_{wi} C_{pw} (T_{wi} - T_{wo})}{\left(\frac{C_{pa} + \omega_{ai} C_{pv}}{\omega_{ai} + 1} \right) (T_{ao} - T_{ai}) + \left(\frac{Y_{ai} - Y_{ao}}{Y_{ao}} \right) \left[(h_{go} - h_{wi}) + C_{pw} (T_{wi} - T_{wo}) \right] + a C_{pw} (T_{ao} - T_{wo})} \quad (31)$$

where

$$\omega_{ai} = \frac{0.622 \phi P_{sat}(T_{ai})}{P_{atm} - \phi P_{sat}(T_{ai})} \quad (32)$$

$$Y_{ai} = \frac{1}{\omega_{ai} + 1}$$

$$Y_{ao} = \frac{1}{\frac{P_{sat}(T_{ao}) M_v}{[P_{atm} - P_{sat}(T_{ao})] M_a} + 1}$$

The cooling water exit temperature, T_{wo} , can approach but not be less than wet bulb temperature of the atmosphere. The air exit temperature, T_{ao} , can approach but not exceed the inlet cooling water temperature. These approach temperatures must be considered in the model input. A term has been added to Equation (31) to account for liquid water entrained in the air (drift) and removed from the tower. The amount of drift is assumed linearly proportional to the mass flow rate of air. The drift rate is generally negligible.

The fan electrical power requirement is computed from the following compressible flow equation:

$$P_{ow} = \frac{W_a}{\eta} \left(\frac{k}{k-1} \right) RT_i \left[\left(\frac{P_o}{P_i} \right)^{\frac{k-1}{k}} - 1 \right] \quad (33)$$

Equation (33) is also used if an air compressor is specified instead of the standard two-stage air ejector system for removing the noncondensable gas from the direct-contact condenser.

2.6 Pump Power and Pressure Loss Model

This model is used in computing the power requirements to pump brine and cooling water flows throughout the plant. Given the inside diameter, length, static head, surface roughness, head loss coefficients, temperature, pump efficiency, and the inlet and outlet pressures, the model computes the frictional pressure loss and the required pumping power. The friction pressure is computed using single phase Moody friction factors and the Darcy-Weisbach equation as given by:

$$\Delta P = \rho \left(f \frac{L}{D} + \sum k \right) \frac{V^2}{2g} \quad (34)$$

The pumping power is given by:

$$P_{ow} = \frac{W \Delta P}{\eta_p} \quad (35)$$

3. Computer Program Description

A computer program of this geothermal plant model was written and named GOSP standing for Geothermal Open-Loop Steam-Flash Plant. The program was written in FORTRAN IV and is operational on the IBM 360/75. The program computes the required geothermal brine flow rate to produce a specified net electrical power output.

The program first solves for the gross electrical power per pound of brine per hour, then computes the plant power requirements by assuming a brine mass flow rate and iterates on the brine flow rate until the net electrical equals the specified value. Although the gross electrical can be computed on a per pound basis, it is necessary to specify the brine flow rate in order to compute the frictional pressure losses and thereby the pumping power requirements. Figure 5 shows a flow chart for the program.

Selection of flash pressure can be done either by the code user as input or by the code itself. When the program is allowed to select the flash pressures, the flash temperatures are divided into equal increments between the preflasher temperature and the turbine condenser temperature. This type of flash pressure selection has been found to be approximately optimum and was verified by sensitivity studies.

The program iterates on the preflasher flash pressure until just the right amount of steam is generated to drive the air ejectors. The lower the preflasher pressure, the more noncondensable gases evolve and are used to drive the air ejectors instead of flowing through the turbine. The smaller the amount of noncondensable gases flowing through the turbine, the less steam needed to drive the air ejectors. The program iterates until it strikes a balance. The plant inlet pressure is required input as it is needed in computing pumping power in the brine supply line. If the specified plant inlet pressure is lower than the bubble formation pressure, gases will evolve in the brine supply line and it is possible that sufficient flashing will occur before the plant inlet to drive the air ejectors. Should this be the case, the brine supply line itself acts as the preflasher and the program merely draws off the necessary flow for the air ejectors and passes the remainder through to the first flash stage. If an air compressor is specified by the input to draw the gases out of the condenser or if the initial dissolved gases is specified as zero, the preflasher pressure is set at the plant inlet pressure and any gases which might have already evolved are passed through to the first flash stage.

* Indicates subroutine using the properties library containing subroutines LIB, SATT, PROP, TBL.

Fig. 5 GOSP Flow chart

Pressure losses are taken into account throughout the plant either by input or by a single phase flow calculation. The pressure drops between each flash stage and the turbine, between the preflasher and the air ejectors, between the turbine exit and the turbine condenser inlet, and between the turbine and inter condenser exits to the next air ejectors inlet are all required program input. Nine pressure drops in brine supply and return lines, and cooling water circulation system as shown in Figure 1 are computed using the Darcy-Weisbach equation and Moody friction factors. Down-hole pumping power for the supply wells can be included in either of two ways. Down-hole pumping power can be input on a per pound per hour basis or it can be included into the number nine pump power calculation by adjusting the input such as the gravity head and well head pressure. Enough pumps were included into the cooling water circulating system so that the cooling tower elevation can be adjusted as desired relative to the condenser elevations.

An important aspect of the program is that no design considerations have been implied by the program but are input by the user. This means that all approach temperatures in the condensers and cooling tower, all efficiencies of the turbine, pumps and compressors, and all piping parameters, are input by the user. Cooling water temperatures must be specified to agree with the direct-contact condenser saturation temperature, the wet bulb temperature, and the condenser and cooling tower approach temperatures. The brine supply and return pipes can be divided up into any number of equal length and equal diameter pipes to accommodate the pattern of the well field.

Even as the design parameters are user input for better code utilization, all thermodynamic properties are self-contained in an internal library to reduce the quantity of program input. These thermoproperties include: (1) both constant pressure and constant volume specific heats; (2) gas constants; (3) molecular weights; (4) saturation pressures and temperatures; (5) water viscosity; (6) water density; (7) saturated vapor and liquid enthalpies; (8) saturated vapor and liquid entropies; and (9) slightly superheated enthalpies and entropies. The program is capable of handling geothermal brine temperatures up to 600°F.

The appendix contains a detailed program input description and a sample program output.

III. RESULTS

A sensitivity study was performed on the more significant model input parameters by varying each parameter separately from a base case.

1. Base Case

The Raft River Project's geothermal source and atmospheric conditions were used for the base case. Reasonable values for input design parameters such as the number of flash stages or the condenser pressure were specified. Actual specification of optimum design parameters will require the inclusion of economics. Table 1 lists the model input data which specifies the base case. The base case allows the program to select the flash pressures and uses the air ejectors and preflasher to remove the noncondensable gases from the condenser.

The most significant results of the base case input to the model are listed in Table 2. More detailed results are found in the sample output of the base case located in the appendix. The results show that significant gas evolution and steam flashing occurs prior to the plant inlet to drive the air ejectors so that the brine supply line becomes the preflasher. The steam flow driving the air ejectors is relatively small due to the small concentrations of dissolved noncondensable gases in the brine and to the significant fractions of the gases which evolved during preflashing and are used to drive the air ejectors. Essentially all of the dissolved gases evolved in either the preflasher or the first flasher stage.

2. Sensitivity Study

A sensitivity study was performed by varying several of the geothermal source, design, and atmospheric parameters. These sensitivities are correlated according to the geothermal brine mass flow rate required to produce 10 Mw net electrical.

TABLE 1

BASE CASE CONDITIONSGeothermal Source Parameters:

Temperature -	300°F
Dissolved gases:	
Nitrogen -	.00004 gm/gmH ₂ O
Carbon-dioxide -	.000012 gm/gmH ₂ O

Design Parameters:

Net Power -	10 MW
Number of Flash Stages -	2
Pressure:	
Plant inlet -	70 psia
Well head -	70 psia
Reinjection -	312 psia
Turbine condenser -	1.5 psia
Inter condenser -	7 psia
After condenser -	12.5 psia
Differential Pressures:	
Flash to turbine or air ejector -	0.5 psi
Turbine to condenser -	0.25 psi
Condenser to air ejector -	0.25 psi
Cooling tower fan -	0.05 psi
Temperatures:	
Cooling water at condenser inlet -	75°F
Cooling water at condenser outlet -	110°F
Differential Temperatures:	
Condenser approach -	15°F
Cooling tower approach -	10°F
Efficiencies:	
Turbine/Generator:	
Internal (turbine steam path) -	85%
Mechanical -	96%
Generator (losses + power factor) -	90%
Cooling tower fan:	
Internal -	83%
Motor -	96%
Pumps -	82%
Down-hole Pumping -	287 ft
Miscellaneous electrical -	500 KW
Diameters:	
Brine supply lines (7 lines) -	10 in.
Brine return lines (4 lines) -	12 in.
Cooling water main lines -	30 in.
Lengths:	
Brine supply lines -	2000 ft
Brine return lines	4300 ft
Total condenser-cooling tower pump head -	70 ft

Atmospheric Parameters:

Pressure -	12.3 psia
Dry bulb temperature -	96°F
Relative humidity -	0.23

TABLE 2

BASE CASE RESULTS

Pressures (psia)

Bubble formation -	105.2
Preflasher -	70.0
Flasher 1 -	24.2
Flasher 2 -	7.0

Flow Rates (lbM/hr)

Brine supply -	3.89×10^6
Brine return -	3.41×10^6
Preflasher steam -	596
Flasher 1 steam -	0.257×10^6
Flasher 2 steam -	0.226×10^6
Cooling water supply -	13.6×10^6
Cooling water supply to brine supply ratio -	3.50

Dissolved Gas Distribution (%)

	<u>N₂</u>	<u>CO₂</u>
Preflasher -	26.1	11.8
Flasher 1 -	73.9	87.8
Flasher 2 -	0.01	0.4
Brine Return -	Trace	Trace

Powers (KW)

Net -	10,000
Gross -	13,530
Net-to-gross ratio -	0.74
Losses -	3530
Brine supply pumping -	569
Brine return pumping -	1191
Coolant pumping -	627
Cooling tower fan -	643
Miscellaneous electrical -	500

2.1 Geothermal Source Parameters

The geothermal source temperature and dissolved gas concentrations were varied separately from the base case to determine the sensitivities. These sensitivities are shown in Figure 6, 7(a), and 7(b) for the temperature, nitrogen, and carbon-dioxide gases, respectively. The gross and net brine effectivenesses which are the gross and net powers per pound mass per hour of brine are also plotted in Figures 6 and 7. These results assume solubility equilibrium.

The brine mass flow rate required for 10 MW net electrical increases rapidly for temperatures below about 300°F or for dissolved gas concentrations above about 1% by weight. As the temperature decreases or the concentrations increase, the net to gross power ratio decreases. The dissolved gas concentrations specified in the base case input have an almost negligible effect upon required brine mass flow rate.

2.2 Design Parameters

The effectiveness of using ejectors instead of air compressors for the removal of noncondensable gases from the turbine condenser is demonstrated in Figures 8(a) and 8(b) for nitrogen and carbon dioxide gases, respectively. At low dissolved gas concentrations, neither the air ejectors or the air compressor show a significant influence on the brine mass flow rate but at high dissolved gas concentrations the air ejectors are considerably more effective in removing the gases than the air compressor. The main reason for the increased effectiveness of the air ejectors is that a significant portion of the dissolved gases are removed from the brine during preflashing and used to drive the air ejectors and therefore do not have to be removed from the condenser. The accuracy of the air ejector calculations may be affected by condensation under certain conditions, however, the order of magnitude of the effect has been illustrated. Dissolved gas evolution during a pressure reduction is demonstrated in Figures 9 and 10 for nitrogen and carbon dioxide gases, respectively. These figures show the percentage of a dissolved gas remaining in solution as the brine is flashed to a lower pressure assuming the initial pressure was high enough to completely dissolve the specified gas concentration. The one-hundred percent line represents the bubble formation pressure. As the pressure reduction gets close to the saturation pressure of water (at the brine temperature of 300°F in this case),

Fig. 6 Source temperature sensitivity

a) Nitrogen

b) Carbon-dioxide

Fig. 7 Gas concentration sensitivity

a) Nitrogen

b) Carbon-dioxide

Fig. 8 Air ejector-compressor trade-off

Fig. 9 Remaining dissolved nitrogen after flash (300°F initial temperature)

Fig. 10 Remaining dissolved carbon-dioxide after flash (300°F initial temperature)

the dissolved gases evolve much more rapidly. The saturation pressure line is asymptotic with the one-hundred percent line at zero gas concentrations. Steam flashing rapidly increases as the saturation pressure is approached which increases the gas volume, decreases the partial pressures of the non-condensable gases, and allows more gases to evolve.

The optimum flash pressures seem to be those pressures at which the saturation temperatures are spaced at approximately equal divisions between the preflasher temperature and the turbine condensation temperature. A parametric study of the flash pressures of a double flash system is shown in Figure 11. The parametric study was performed by fixing the first flash pressure at a given value and then varying the second flash pressure. The first and second flash pressures for the base case as listed in Table 2 are 24.2 and 7 psia. The brine flow rate goes through a minimum for each fixed first flash pressure as the second flash pressure is varied and smallest minimum is approximately at the base case flash pressures. Figure 11 supports the program's option for selecting the flash pressures.

The sensitivity of the number of flash stages is shown in Figure 12 for four flash stages. The effective use of the brine increases with an increasing number of flash stages, however, the rate of increase decreases. An actual plant design will have to seek an economic optimum between the capital cost of increasing the number of flashes to the cost savings of decreasing the brine flow rate.

The sensitivity of the turbine condenser pressure is shown in Figure 13. The brine flow rate increases rapidly with increasing condenser pressure due to low (subatmospheric) second flash pressure of 7 psia. The condensing temperature decreases as the pressure decreases and thereby decreases the cooling water exit temperature. The cooling water flow rate increases as the difference between the condenser temperature and atmospheric wet bulb decreases causing the curve in Figure 13 to tend to flatten out and eventually turn upward. It appears that the optimum condenser pressure considering all other parameters fixed is slightly under 1 psia.

Fig. 11 Flash pressure sensitivity for a two-flash system

Fig. 12 The value of increasing the number of flashes

Fig. 13 Condenser pressure sensitivity

2.3 Atmospheric Parameters

The sensitivities of the relative humidity and dry bulb temperature are shown in Figure 14. These sensitivities are shown both as a function of the brine flow rate and the cooling tower effectiveness. The cooling tower effectiveness as used here is the cooling tower electrical fan power divided by the cooling water flow rate supplied to the plant. As the cooling tower efficiency decreases, the plant must expend more power to operate and therefore requires larger brine flow rates to produce a given net electrical. A plant design should consider a range of values for the atmospheric conditions which correspond to plant location.

Fig. 14 Sensitivity of atmospheric temperature (dry bulb) and humidity on cooling tower and overall plant performance

IV. CONCLUSIONS

A model of a geothermal flashed steam electrical power plant has been developed and programmed for the IBM 360/75. The program structure was oriented toward assisting plant designers.

Sensitivities of several geothermal source, design, and atmospheric parameters were completed, using the source conditions found in the Raft River Valley as a base case. The most sensitive parameters studied included the initial brine temperature, the dissolved gas concentrations when high, the number of flash stages, and the condenser pressure. A preflasher-air ejector system was found the most effective method of removing the non-condensable gases from the condenser. These studies indicate that 10 MW(e) net can be produced from the Raft River Valley geothermal brine (300°F) using less than 4 million pounds per hour of brine.

V. REFERENCES

1. JFKunze, LGMiller, DTNeill, and CRNichols, "A Low Temperature Demonstration Geothermal Power Plant in the Raft River Valley", ANCR-1138 (April 1974).
2. AJEllis, "The Solubility of Carbon Dioxide in Water at High Temperatures", American Journal of Science, Vol. 257, Pp 217-234 (March 1959).
3. AJEllis and RMGolding, "The Solubility of Carbon Dioxide Above 100°C in Water and in Sodium Chloride Solutions", American Journal of Science, Vol. 261, Pp 47-60 (January 1963).
4. HAPray, CESchweickert, and BHMinnich, "Solubility of Hydrogen, Oxygen, Nitrogen, and Helium in Water at Elevated Temperatures", Industrial and Engineering Chemistry, Vol. 44, Pp 1146-1151 (May 1952).
5. GARrowsmith, "The Production of Vacuum for Industrial Chemical Processes", Transactions of the Institution of Chemical Engineers, Vol. 27, Pp 101-112 (1949).

APPENDIX A

CODE INPUT

GOSP INPUT

Card 1: Control Card, Format (7I5)

Cols 1-5	NTF	Number of flash stages to turbine (1-4)
Cols 6-10	IDEAL	Type of air ejector flow in mixing region = 0 isentropic = 1 nonisentropic
Cols 11-15	IEJECT	= 0 Preflasher and air ejectors = 1 Condenser compressor
Cols 16-20	NPRES	= 0 Code computes flash pressures = 1 Flash pressures must be input
Cols 21-25	ITOW	= 0 Code uses wells instead of cooling tower = 1 Cooling tower fan at inlet = 2 Cooling tower fan at outlet
Cols 26-30	NSPOKE	Number of brine supply lines
Cols 31-35	MSPOKE	Number of brine return lines

Card 2: Source Card, Format (F10.3, 2E10.0)

Cols 1-10	TS	Source temperature ($^{\circ}\text{F}$)
Cols 11-20	YNS	Dissolved N_2 fraction at source (gm/gm)
Cols 21-30	YCS	Dissolved CO_2 fraction at source (gm/gm)

Card 3: Pressure Card, Format (7 F10.3)

Cols 1-10	PIN	Brine supply plant inlet pressure (psia)
Cols 11-20	PCOND(1)	Barometric condenser pressure (psia)
Cols 21-30	PCOND(2)	Intercondenser pressure (psia)
Cols 31-40	PCOND(3)	Aftercondenser pressure (psia)
Cols 41-50	PWH	Supply line well head pressure (psia)
Cols 51-60	PIJECT	Reinjector well head pressure (psia)
Cols 61-70	PA	Atmospheric pressure (psia)

Card 4: Flash Pressures Card, Format (4F10.3)

(Enter only if NPRES on Card 1 = 1)

Cols 1-10	PF(1)	Flash pressure of Stage 1
Cols 11-20	PF(2)	Flash pressure of Stage 2
Cols 21-30	PF(3)	Flash pressure of Stage 3
Cols 31-40	PF(4)	Flash pressure of Stage 4

Card 5: Pressure Loss Card, Format (6F10.3)

Cols 1-10	DPT(1)	Flasher 1 to turbine pressure loss (psi)
Cols 11-20	DPT(2)	Flasher 2 to turbine pressure loss (psi)
Cols 21-30	DPT(3)	Flasher 3 to turbine pressure loss (psi)
Cols 31-40	DPT(4)	Flasher 4 to turbine pressure loss (psi)
Cols 41-50	DPE(1)	Preflasher to air ejector 1 pressure loss (psi)
Cols 51-60	DPE(2)	Preflasher to air ejector 2 pressure loss (psi)

Card 6: Pressure Loss Card, Format (4F10.3)

Cols 1-10	DPCOND(1)	Turbine exit to barometric condenser inlet (psi)
Cols 11-20	DPCOND(2)	Barometric condenser to ejector Stage 1 inlet (psi)
Cols 21-30	DPCOND(3)	Intercondenser to ejector Stage 2 inlet (psi)
Cols 31-40	DPAIR	Pressure increase by cooling tower fan (psi)

Card 7: Temperature Card, Format (4F10.3)

Cols 1-10	TCE	Condenser inlet cooling water ($^{\circ}\text{F}$)
Cols 11-20	TCO(1)	Barometric condenser cooling water outlet ($^{\circ}\text{F}$)
Cols 21-30	TCO(2)	Intercondenser cooling water outlet ($^{\circ}\text{F}$)
Cols 31-40	TCO(3)	Aftercondenser cooling water outlet ($^{\circ}\text{F}$)

Card 8: Cooling Tower Card, Format (5F10.3)

Cols 1-10	TAI	Atmospheric dry-bulb temperature ($^{\circ}\text{F}$)
Cols 11-20	TAQ	Cold aquifer water temperature ($^{\circ}\text{F}$)
Cols 21-30	DTOW	Cooling tower air to inlet water approach temperature ($^{\circ}\text{F}$)
Cols 31-40	PHI	Relative humidity
Cols 41-50	A	Drift factor (lbm drift/lbm air)

Card 9: Power Card, Format (3F10.3)

Cols 1-10	PNET	Net power (kW)
Cols 11-20	DOWN	Down-hole pump power factor (kW/lbm brine/hr)
Cols 21-30	UTIL	Miscellaneous electrical (kW)

Card 10: Noncondensable Gas to Inlet Cooling Water Approach Temperature Card, Format (3F10.2)

Cols 1-10	DT(1)	Barometric condenser ($^{\circ}\text{F}$)
Cols 11-20	DT(2)	Intercondenser ($^{\circ}\text{F}$)
Cols 21-30	DT(3)	Aftercondenser ($^{\circ}\text{F}$)

Card 11: Efficiency Card, Format (3F10.3)

Cols 1-10	EFF	Turbine Nonreversibilities
Cols 11-20	BEFF	Turbine mechanical
Cols 21-30	GEFF	Overall generator

Card 12: Efficiency Card, Format (7F10.3)

Cols 1-10	EC(1)	Pump 1
Cols 11-20	EC(2)	Pump 2
Cols 21-30	EC(3)	Pump 3
Cols 31-40	EC(4)	Pump 4
Cols 41-50	EC(5)	Pump 5
Cols 51-60	EC(6)	Pump 6
Cols 61-70	EC(7)	Pump 7

Card 13: Efficiency Card, Format (7F10.3)

Cols 1-10	EC(8)	Pump 8
Cols 11-20	EC(9)	Pump 9
Cols 21-30	EC(10)	Pump 10

Card 13: Efficiency Card, Format (7F10.3) (continued)

Cols 31-40	EC(11)	Cooling tower fan nonreversibilities
Cols 41-50	EC(12)	Cooling tower fan motor
Cols 51-60	EC(13)	Condenser compressor nonreversibilities (If IEJECT = 1)
Cols 61-70	EC(14)	Condenser compressor motor (If IEJECT = 1)

Cards 14-22: Piping Cards, Format (9(5F10.3))

Cols 1-10	DC(I)	Diameter of pipe I (in.)
Cols 11-20	LC(I)	Length of pipe I (ft)
Cols 21-30	HC(I)	Gravity head of pipe I (ft) (+ for upward flow)
Cols 31-40	RC(I)	Surface roughness of Pipe I (ft)
Cols 41-50	SKC(I)	Sum of head loss coefficients of pipe I

APPENDIX B

SAMPLE CODE OUTPUT

-----PROGRAM INPUT-----

CONTROL INPUT

NTE= 2
IDEAL= 1
IEJECT= 0
NPRES= 0
ITOW= 1
NSPOKE= 7
MSPOKE= 4

SOURCE TEMPERATURE AND GAS CONCENTRATIONS

TS,YNS,YCS= 3.0000E 02 4.0000E-05 1.2000E-05

PRESSURES

PIN,PCOND(BARO,INTER,AFTER)= 7.0000E 01 1.5000E 00 7.0000E 00 1.2500E 01
PWH,PIJECT,PA= 7.0000E 01 3.1200E 02 1.2300E 01

DIFFERENTIAL PRESSURES

FLASHERS TO TURBINE(1-4)= 5.0000E-01 5.0000E-01 5.0000E-01 5.0000E-01
FLASHERS TO AIR-EJECTORS(1-2)= 5.0000E-01 5.0000E-01
TURBINE EXIT TO INLET BARO. COND.= 2.5000E-01
BARO. COND. TO STAGE 1 INLET= 2.5000E-01
INTER. COND. TO STAGE 2 INLET= 2.5000E-01
TOWER FAN= 5.0000E-02

TEMPERATURES

COND. INLET= 7.5000E 01
COND. OUTLET(BARO,INT,AFT)= 1.1000E 02 1.1000E 02 1.1000E 02
DT(1-3) FOR COND.= 1.5000E 01 1.5000E 01 1.5000E 01
OUTSIDE AIR TEMP= 9.6000E 01
COLD AQUIFER TEMP= 9.0000E 01
TOWER AIR-WATER APPROACH= 1.0000E 01

MISCELLANEOUS

NET POWER (KW)= 1.0000E 04
 DOWN HOLE PUMP POW FACTOR= 1.316E-04
 UTILITIES (KW)= 5.0000E 02

RELATIVE HUMIDITY= 2.3000E-01
 TOWER DRIFT FACTOR= 1.0000E-04

EFFICIENCIES

TURBINE (INTERNAL)= 8.5000E-01
 TURBINE (MECHANICAL)= 9.6000E-01
 GENERATOR= 9.0000E-01

PUMPS AND COMPRESSORS

1 8.2000E-01
 2 8.2000E-01
 3 8.2000E-01
 4 8.2000E-01
 5 8.2000E-01
 6 8.2000E-01
 7 8.2000E-01
 8 8.2000E-01
 9 8.2000E-01
 10 8.2000E-01
 11 8.3000E-01
 12 9.6000E-01
 13 8.3000E-01
 14 9.6000E-01

PRESSURE DROP PARAMETERS

DC(I),LC(I),HC(I),RC(I),SKC(I)

1	1.0000E 00	2.0000E 01	-2.0000E 00	6.0000E-04	5.0000E 00
2	1.0000E 00	1.0000E 01	-2.0000E 00	6.0000E-04	5.0000E 00
3	3.0000E 01	2.0000E 00	-2.0000E 00	6.0000E-04	5.0000E 00
4	3.0000E 01	1.0000E 02	2.0000E 01	6.0000E-04	2.0000E 00
5	3.0000E 01	1.0000E 02	5.0000E 01	6.0000E-04	2.0000E 00
6	3.0000E 00	1.0000E 02	-2.0000E 01	6.0000E-04	5.0000E 01
7	1.0000E 01	2.0000E 03	0.0	6.0000E-04	4.0000E 01
8	1.2000E 01	4.3000E 03	0.0	6.0000E-04	2.0000E 01
9	4.0000E 00	5.0000E 02	4.0000E 02	6.0000E-04	5.0000E 00

-----PROGRAM OUTPUT-----

MAXIMUM PRESSURE FOR BUBBLE FORMATION,PSIA

TOTAL= 1.0523D 02

VAPOR PARTIAL= 6.7004D 01

NITROGEN PARTIAL= 3.7762D 01

CARBON-DIOXIDE PARTIAL= 4.6731D-01

FIRST FLASH PRESSURE GREATER THAN PLANT INLET PRESSURE

EJECTOR STAGE NO.= 1

EJECTOR PRESSURES, PSIA

NOZZLE INLET PRESSURE= 6.9500E 01
NOZZLE THROAT PRESSURE= 3.4721E 01
NOZZLE EXIT PRESSURE= 1.2500E 00
DIFFUSER THROAT PRESSURE= 3.8029E 00
EJECTOR EXIT PRESSURE= 7.0000E 00

EJECTOR TEMPERATURES, F

NOZZLE INLET TEMPERATURE= 2.9956E 02
NOZZLE THROAT TEMPERATURE= 1.2858E 02
NOZZLE EXIT TEMPERATURE= -2.5970E 02
DIFFUSER ENTRANCE TEMPERATURE= -1.6059E 02
DIFFUSER THROAT TEMPERATURE= -9.5713E 01
EJECTOR EXIT TEMPERATURE= -3.8576E 01

EJECTOR DENSITIES, LBM/FT3

NOZZLE INLET DENSITY= 1.5835E-01
NOZZLE THROAT DENSITY= 1.0209E-01
DIFFUSER THROAT DENSITY= 1.9833E-02
EJECTOR EXIT DENSITY= 3.1558E-02

EJECTOR MASS FRACTIONS

CONDENSER EXIT STEAM FRACTION= 3.3609E-01
CONDENSER EXIT NITROGEN FRACTION= 4.8889E-01
CONDENSER EXIT CARBON DIOXIDE FRACTION= 1.7503E-01

NOZZLE INLET STEAM FRACTION= 9.2261E-01
NOZZLE INLET NITROGEN FRACTION= 6.8131E-02
NOZZLE INLET CARBON-DIOXIDE FRACTION= 9.2564E-03

EJECTOR EXIT STEAM FRACTION= 7.1828E-01
EJECTOR EXIT NITROGEN FRACTION= 2.1471E-01
EJECTOR EXIT CARBON-DIOXIDE FRACTION= 6.7006E-02

EJECTOR THERMAL PROPERTIES

RATIO OF SPECIFIC HEATS AT NOZZLE INLET= 1.5810E 00
RATIO OF SPECIFIC HEATS AT DIFFUSER INLET= 1.3137E 00

CONST. PRES. SPECIFIC HEAT OF CONDENSER EXIT= 3.0709E-01
CONST. PRES. SPECIFIC HEAT AT NOZZLE EXIT= 4.1509E-01

EJECTOR VELOCITIES

NOZZLE EXIT VELOCITY, FPS= 2.9233E 03
DIFFUSER ENTRANCE VELOCITY, FPS= 1.5793E 03
RATIO OF DIFFUSER ENTRANCE TO NOZZLE EXIT VEL.= 5.4023E-01

NOZZLE THROAT SONIC VELOCITY, FPS= 1.5790E 03
DIFFUSER THROAT SONIC VELOCITY, FPS= 1.0807E 03

NOZZLE EXIT MACH NO.= 3.4101E 00
DIFFUSER ENTRANCE MACH NO.= 1.6119E 00

EJECTOR AREA RATIOS

NOZZLE EXIT TO THROAT AREAS= 4.4223E 00

DIFFUSER ENTRANCE TO THROAT AREAS= 1.2787E 00

ACTUAL EJECTOR FLOW RATIO= 1.8696E 00

EJECTOR STAGE NO.= 2

EJECTOR PRESSURES, PSIA

NOZZLE INLET PRESSURE= 6.9500E 01
NOZZLE THROAT PRESSURE= 3.4721E 01
NOZZLE EXIT PRESSURE= 6.7500E 00
DIFFUSER THROAT PRESSURE= 6.7538E 00
EJECTOR EXIT PRESSURE= 1.2500E 01

EJECTOR TEMPERATURES, F

NOZZLE INLET TEMPERATURE= 2.9956E 02
NOZZLE THROAT TEMPERATURE= 1.2858E 02
NOZZLE EXIT TEMPERATURE= -1.3758E 02
DIFFUSER ENTRANCE TEMPERATURE= -3.7892E 01
DIFFUSER THROAT TEMPERATURE= -4.7951E 01
EJECTOR EXIT TEMPERATURE= 2.0109E 01

EJECTOR DENSITIES, LB_M/FT³

NOZZLE INLET DENSITY= 1.5835E-01
NOZZLE THROAT DENSITY= 1.0209E-01
DIFFUSER THROAT DENSITY= 3.5806E-02
EJECTOR EXIT DENSITY= 5.6876E-02

B-7

EJECTOR MASS FRACTIONS

CONDENSER EXIT STEAM FRACTION= 6.1103E-02
CONDENSER EXIT NITROGEN FRACTION= 7.1558E-01
CONDENSER EXIT CARBON DIOXIDE FRACTION= 2.2332E-01

NOZZLE INLET STEAM FRACTION= 9.2261E-01
NOZZLE INLET NITROGEN FRACTION= 6.8131E-02
NOZZLE INLET CARBON-DIOXIDE FRACTION= 9.2564E-03

EJECTOR EXIT STEAM FRACTION= 4.3688E-01
EJECTOR EXIT NITROGEN FRACTION= 4.3317E-01
EJECTOR EXIT CARBON-DIOXIDE FRACTION= 1.2995E-01

EJECTOR THERMAL PROPERTIES

RATIO OF SPECIFIC HEATS AT NOZZLE INLET= 1.5810E 00
RATIO OF SPECIFIC HEATS AT DIFFUSER INLET= 1.3303E 00

CONST. PRES. SPECIFIC HEAT OF CONDENSER EXIT= 2.5038E-01
CONST. PRES. SPECIFIC HEAT AT NOZZLE EXIT= 4.1520E-01

EJECTOR VELOCITIES

NOZZLE EXIT VELOCITY, FPS= 2.5247E 03
DIFFUSER ENTRANCE VELOCITY, FPS= 9.9577E 02
RATIO OF DIFFUSER ENTRANCE TO NOZZLE EXIT VEL.= 3.9441E-01

NOZZLE THROAT SONIC VELOCITY, FPS= 1.5790E 03
DIFFUSER THROAT SONIC VELOCITY, FPS= 1.0787E 03

NOZZLE EXIT MACH NO.= 2.1604E 00
DIFFUSER ENTRANCE MACH NO.= 9.1209E-01

EJECTOR STAGE NO.= 2

EJECTOR PRESSURES,PSIA

NOZZLE INLET PRESSURE= 6.9500E 01
NOZZLE THROAT PRESSURE= 3.4721E 01
NOZZLE EXIT PRESSURE= 6.7500E 00
DIFFUSER THROAT PRESSURE= 6.7538E 00
EJECTOR EXIT PRESSURE= 1.2500E 01

EJECTOR TEMPERATURES,F

NOZZLE INLET TEMPERATURE= 2.9956E 02
NOZZLE THROAT TEMPERATURE= 1.2858E 02
NOZZLE EXIT TEMPERATURE= -1.3758E 02
DIFFUSER ENTRANCE TEMPERATURE= -3.7892E 01
DIFFUSER THROAT TEMPERATURE= -4.7951E 01
EJECTOR EXIT TEMPERATURE= 2.0179E 01

EJECTOR DENSITIES,LB/FT3

NOZZLE INLET DENSITY= 1.5835E-01
NOZZLE THROAT DENSITY= 1.0209E-01
DIFFUSER THROAT DENSITY= 3.5806E-02
EJECTOR EXIT DENSITY= 5.6876E-02

EJECTOR MASS FRACTIONS

CONDENSER EXIT STEAM FRACTION= 6.1103E-02
CONDENSER EXIT NITROGEN FRACTION= 7.1558E-01
CONDENSER EXIT CARBON DIOXIDE FRACTION= 2.2332E-01

NOZZLE INLET STEAM FRACTION= 9.2261E-01
NOZZLE INLET NITROGEN FRACTION= 6.8131E-02
NOZZLE INLET CARBON-DIOXIDE FRACTION= 9.2564E-03

EJECTOR EXIT STEAM FRACTION= 4.3688E-01
EJECTOR EXIT NITROGEN FRACTION= 4.3317E-01
EJECTOR EXIT CARBON-DIOXIDE FRACTION= 1.2995E-01

EJECTOR THERMAL PROPERTIES

RATIO OF SPECIFIC HEATS AT NOZZLE INLET= 1.5810E 00
RATIO OF SPECIFIC HEATS AT DIFFUSER INLET= 1.3303E 00

CONST. PRES. SPECIFIC HEAT OF CONDENSER EXIT= 2.5038E-01
CONST. PRES. SPECIFIC HEAT AT NOZZLE EXIT= 4.1520E-01

EJECTOR VELOCITIES

NOZZLE EXIT VELOCITY,FPS= 2.5247E 03
DIFFUSER ENTRANCE VELOCITY,FPS= 9.9577E 02
RATIO OF DIFFUSER ENTRANCE TO NOZZLE EXIT VEL.= 3.9441E-01

NOZZLE THROAT SONIC VELOCITY,FPS= 1.5790E 03
DIFFUSER THROAT SONIC VELOCITY,FPS= 1.0787E 03

NOZZLE EXIT MACH NO.= 2.1604E 00
DIFFUSER ENTRANCE MACH NO.= 9.1209E-01

EJECTOR AREA RATIOS
NOZZLE EXIT TO THROAT AREAS= 1.7622E 03
DIFFUSER ENTRANCE TO THROAT AREAS= 1.0070E 00

ACTUAL EJECTOR FLOW RATIO= 7.7345E-01

*****FLASH DATA*****NUMBER OF FLASHES= 3

	1	2	3	4	5	6
FLASH PRESSURES	7.0000E 01	2.4199E 01	7.0229E 00			
FLASH TEMPERATURES	2.9956E 02	2.3826E 02	1.7699E 02			
FLASHED GAS WELL FRACTION	1.5328E-04	6.6052E-02	5.7993E-02			
FLASHED GAS FRACTION	1.5328E-04	6.6062E-02	6.2105E-02			

FLASHED GAS N2 FRACTION	6.8131E-02	4.4735E-04	1.0017E-07			
FLASHED GAS CO2 FRACTION	9.2564E-03	1.5954E-04	7.3678E-07			
N2 FRACTION IN FLASH WATER	3.3430E-06	6.2122E-09	3.8234E-13			
CO2 FRACTION IN FLASH WATER	7.0233E-06	4.5766E-08	7.9759E-11			
BRINE RETURN WELL FRACTION	8.7580E-01					

*****FLOW MASS FRACTIONS*****

		WELL	N2	CO2
<hr/>				
TURBINE ENTRANCE FLOWS (2 FLOWS)				
	1	6.6052E-02	4.4735E-04	1.5954E-04
	2	5.7993E-02	1.0017E-07	7.3678E-07
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CONDENSER EXIT GAS FLOWS				
	1	6.0452E-05	4.8889E-01	1.7503E-01
	2	5.2051E-05	7.1558E-01	2.2332E-01
	3	5.3793E-05	7.4333E-01	2.2299E-01
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AIR-EJECTOR ENTRANCE AND EXIT STEAM FLOWS				
ENTRANCE	1	1.1302E-04	6.8131E-02	9.2564E-03
ENTRANCE	2	4.0259E-05	6.8131E-02	9.2564E-03
TOTAL ENTRANCE		1.5328E-04		
EXIT	1	1.7348E-04	2.1471E-01	6.7006E-02
EXIT	2	9.2310E-05	4.3317E-01	1.2995E-01

*****TURBINE DATA*****

INLET PRESSURES AND TEMPERATURES (PSIA,F)

1	2.3699E 01	2.3826E 02
2	6.5229E 00	1.7699E 02

EXIT PRESSURE (PSIA)= 1.7500E 02

TURBINE STAGE EXIT CONDITIONS-XW,XG,YN,YC,T

1	6.6052E-02	9.4774E-01	4.7200E-04	1.6833E-04	1.7370E 02
2	1.2405E-01	9.2637E-01	2.5721E-04	9.2072E-05	1.2120E 02

TURBINE FLOW MIXING CONDITIONS-XG,YN,YC,T

3	9.7290E-01	2.4492E-04	8.7671E-05	1.7370E 02
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TURBINE POWER OUTPUT BY STAGE(BTU/LBM) AND GROSS(KW(E)/LB-DRINE/HR)

1	7.6315E 01
2	7.0103E 01
	3.4784E-03

*****CONDENSER DATA*****

INLET
BARO. COND. WATER FLOW (LBM/HR PER LB-BRINE/HR)= 3.4437E 00
INTER. COND. WATER FLOW= 3.1830E-03
AFTER. COND. WATER FLOW= 1.0467E-03
TURBINE-GEN COOLING= 5.3561E-02

TOTAL COOLING LOAD(LBM/HR PER LB-BRINE/HR)= 3.5015E 00
TOTAL COOLING LOAD(GPM/LB-BRINE/HR)= 7.0167E-03

OUTLET
BARO. COND. WATER FLOW (LBM/HR PER LB-BRINE/HR)= 3.5677E 00
INTER. COND. WATER FLOW= 3.3044E-03
AFTER. COND. WATER FLOW= 1.0852E-03

TOTAL COOLING LOAD(LBM/HR PER LB-BRINE/HR)= 3.6256E 00
AVERAGE COND. OUTLET TEMP (F)= 1.1000E 02

*****BRINE FLOW RATE*****

BRINE FLOW RATE (LBM/HR)= 3.8896E 06

SPECIFIC POWER LOSS(KW/LB-BRINE/HR)= 9.0743E-04

*****PLANT POWER USEAGES*****

BRINE SUPPLY PIPING PUMP POWER-----

REYNOLD NUMBER=	1.9247E 06
FRICTION FACTOR=	1.8728E-02
HYDRAULIC RESISTANCE(SEC2/IN2-FT3)=	3.0793E-02
FRICTION PRESSURE DROP (PSI)=	1.2802E 01
HYDROSTATIC PRESSURE DROP (PSI)=	0.0
PUMPING PRESSURE BOOST (PSI)=	1.2802E 01
PUMPING POWER (KW)=	5.7472E 01

BRINE RETURN PIPING PUMP POWER-----

REYNOLD NUMBER=	1.2710E 06
FRICTION FACTOR=	1.8111E-02
HYDRAULIC RESISTANCE(SEC2/IN2-FT3)=	1.7110E-02
FRICTION PRESSURE DROP (PSI)=	1.5782E 01
HYDROSTATIC PRESSURE DROP (PSI)=	0.0
PUMPING PRESSURE BOOST (PSI)=	3.2076E 02
PUMPING POWER (KW)=	1.1911E 03

DOWN HOLE PUMPING POWER-----

PUMPING POWER (KW)=	5.1187E 02
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COOLANT POWER USAGE----

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TOWER AIR FLOW CALCULATION--
SPECIFIC HUMIDITY AT INLET=      9.9345E-03
INLET AIR MASS FRACTION=        9.9916E-01
OUTLET AIR MASS FRACTION=       9.5053E-01
AIR FLOW RATE (LB1/HR)=         1.0962E 07
MAKE-UP WATER FLOW RATE (L3H/HR)= -2.8894E 04

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TOWER FAN CALCULATION--
FAN OUTLET TEMPERATURE (F)=     9.6774E 01
FAN POWER (KW)=                  6.4267E 02

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CONDENSER INLET HEADER PRESSURE CALCULATION--
REYNOLDS NUMBER=                 3.0846E 06
FRICTION FACTOR=                 1.4840E-02
HYDRAULIC RESISTANCE (SEC2/IN2-FT3)= 2.2429E-05
FRICTION PRESSURE DROP (PSI)=    4.9899E 00
HYDROSTATIC PRESSURE DROP (PSI)= -8.6425E-01
INLET HEADER PRESSURE (PSI)=    5.6256E 00

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TOWER WELL TO COND. HEADER PUMP POWER--
REYNOLD NUMBER=                  3.1364E 06
FRICTION FACTOR=                 1.4835E-02
HYDRAULIC RESISTANCE (SEC2/IN2-FT3)= 1.1606E-05
FRICTION PRESSURE DROP (PSI)=    2.6694E 00
HYDROSTATIC PRESSURE DROP (PSI)= 8.6425E 00
PUMPING PRESSURE BOOST (PSI)=    4.6376E 00

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PUMPING POWER (KW)=              6.7131E 01

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INTERCONDENSER PUMPING POWER--
 REYNOLD NUMBER= 8.5532E 04
 FRICTION FACTOR= 3.5573E-02
 HYDRAULIC RESISTANCE(SEC2/IN2-FT3)= 3.3599E 01
 FRICTION PPESSURE DROP (PSI)= 6.3858E 00
 HYDROSTATIC PRESSURE DROP (PSI)= -8.6425E-01
 PUMPING PRESSURE BOOST (PSI)= 6.8959E 00
 PUMPING POWER (KW)= 9.0741E-02

AFTERCONDENSER PUMPING POWER--
 REYNOLD NUMBER= 2.8126E 04
 FRICTION FACTOR= 3.7201E-02
 HYDRAULIC RESISTANCE(SEC2/IN2-FT3)= 5.0489E 01
 FRICTION PPESSURE DROP (PSI)= 1.0377E 00
 HYDROSTATIC PRESSURE DROP (PSI)= -8.6425E-01
 PUMPING PRESSURE BOOST (PSI)= 7.0478E 00
 PUMPING POWER (KW)= 3.0496E-02

CONDENSER OUTLET HEADER PRESSURE CALCULATION--
 REYNOLDS NUMBER= 4.8324E 06
 FRICTION FACTOR= 1.4734E-02
 HYDRAULIC RESISTANCE(SEC2/IN2-FT3)= 1.1588E-05
 FRICTION PRESSURE DROP (PSI)= 2.8748E 00
 HYDROSTATIC PRESSURE DROP (PSI)= 2.1477E 01
 OUTLET HEADER PRESSURE (PSI)= 3.6652E 01

 BACINETRIC CONDENSER OUTLET PUMP CALCULATION--
 REYNOLD NUMBER= 4.7552E 06
 FRICTION FACTOR= 1.4738E-02
 HYDRAULIC RESISTANCE(SEC2/IN2-FT3)= 0.0
 FRICTION PRESSURE DROP (PSI)= 0.0
 HYDROSTATIC PRESSURE DROP (PSI)= 0.0
 PUMPING PRESSURE BOOST (PSI)= 3.5152E 01
 PUMPING POWER (KW)= 5.2158E 02

 INTERCONDENSER OUTLET PUMP CALCULATION--
 REYNOLD NUMBER= 1.3213E 05
 FRICTION FACTOR= 3.5272E-02
 HYDRAULIC RESISTANCE(SEC2/IN2-FT3)= 0.0
 FRICTION PRESSURE DROP (PSI)= 0.0
 HYDROSTATIC PRESSURE DROP (PSI)= 0.0
 PUMPING PRESSURE BOOST (PSI)= 2.9652E 01
 PUMPING POWER (KW)= 4.9750E-01

 AFTERCONDENSER OUTLET PUMP CALCULATION--
 REYNOLD NUMBER= 4.3392E 04
 FRICTION FACTOR= 3.6370E-02
 HYDRAULIC RESISTANCE(SEC2/IN2-FT3)= 0.0
 FRICTION PRESSURE DROP (PSI)= 0.0
 HYDROSTATIC PRESSURE DROP (PSI)= 0.0
 PUMPING PRESSURE BOOST (PSI)= 2.4152E 01
 PUMPING POWER (KW)= 1.0907E-01

TURBINE COOLING PUMP CALCULATION--

REYNOLD NUMBER= 4.7975E 05
 FRICTION FACTOR= 2.5433E-02
 HYDRAULIC RESISTANCE(SEG2/IN2-FT3)= 2.6929E 00
 FRICTION PRESSURE DROP (PSI)= 1.4492E 02
 HYDROSTATIC PRESSURE DROP (PSI)= -8.6425E 00
 PUMPING PRESSURE BOOST (PSI)= 1.6731E 02

PUMPING POWER (KW)= 3.7045E 01

TOTAL COOLANT POWER (KW)= 1.2691E 03

 PLANT UTILITIES (INPUT) (KW)= 5.0000E 02

 TOTAL PLANT POWER USAGES (KW)= 3.5295E 03

PLANT FLOW RATES (LBM/HR)

BRINE SUPPLY= 3.8896E 06
 BRINE RETURN= 3.4065E 06

INLET COOLING WATER

TOTAL= 1.3619E 07
 HARD. COND. COOLING WATER= 1.3395E 07
 INTER. COND. COOLING WATER= 1.2380E 04
 AFTER. COND. COOLING WATER= 4.0712E 03
 TURBINE COOLING WATER= 2.0833E 05

OUTLET COOLING WATER

TOTAL= 1.4102E 07
 HARD. COND. COOLING WATER= 1.3877E 07
 INTER. COND. COOLING WATER= 1.2853E 04
 AFTER. COND. COOLING WATER= 4.2210E 03

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POWER

GROSS POWER (KW)=	1.3530E 04
NET EFFECTIVENESS (KW/LB-BRINE)=	2.5710E-03
GROSS EFFECTIVENESS (KW/LB-BRINE)=	3.4784E-03

FLOW RATES (LBM/HR)

PREFLASHER STEAM=	5.9619E 02
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FLASHER STEAM=	
1	2.5691E 05
2	2.2557E 05

TURBINE EXIT=	4.8248E 05
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CONDENSER GAS OUTLET	
BAROMETRIC=	2.3513E 02
INTER=	2.0246E 02
AFTER=	2.0923E 02

EJECTOR SUPPLY=	
1	4.3961E 02
2	1.5659E 02

EJECTOR EXIT=	
1	6.7475E 02
2	3.5905E 02

EJECTOR DIAMETERS (IN)

EJECTOR 1	
NOZZLE THROAT=	3.7269E-01
NOZZLE EXIT=	7.8375E-01
DIFFUSER THROAT=	1.2662E 00
DIFFUSER ENTRANCE=	1.4319E 00

EJECTOR 2	
NOZZLE THROAT=	2.2243E-01
NOZZLE EXIT=	2.9528E-01
DIFFUSER THROAT=	6.8810E-01
DIFFUSER ENTRANCE=	6.9050E-01

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