THERMODYNAMIC MODELING OF GEOTHERMAL POWER PLANTS

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R. A. Walter
C. M. Bloomster
S. E. Wise

On January 19, 1975, research and development programs of the U.S. Atomic Energy Commission (AEC) became part of the newly formed Energy Research and Development Administration (ERDA). In this report, since it refers to work done in 1974, most references are to AEC programs.

BATTELLE
PACIFIC NORTHWEST LABORATORIES
RICHLAND WASHINGTON 99352

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ABSTRACT

Different types of power plants for generating electricity from geothermal reservoirs have been proposed. A computer program has been constructed at Battelle-Northwest Laboratories to analytically model four different geothermal power plants for these applications. This program is one part of an overall geothermal system model called GEOCOST. GEOCOST, a simulation model which calculates the cost of generating electricity from geothermal energy, is composed of two principal sections: a reservoir model which simulates the exploration, development, and operation of a geothermal reservoir, and a power plant model which simulates the design, construction, and operation of the power plant. Power plant modeling results are given in this report.

The power plant models include the flashed steam plant, binary fluid cycle plant, total flow plant, and the geopressured plant. For each of these plants, fluid mass flow rates, physical and thermodynamic properties of fluids used, thermodynamic state points, component sizes, and plant design performance are determined for a variety of geothermal fluid conditions. This computer model can be used to predict the optimum power plant type and operating conditions for use with a particular geothermal resource. Using the overall model, sensitivity analyses can be performed to determine the relative effect of different economic parameters, assumptions, and uncertainties on the cost of generating electricity.

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INTRODUCTION AND SUMMARY

The objective of this program is to study in detail a limited number of concepts for power generation from geothermal energy and to decide which are most practical for use with different reservoir types. The result of these studies is the power plant model discussed in this report. Four different power generation schemes were modeled; the direct or flashed steam cycle, the binary fluid cycle, the total flow plant, and the geopressed plant. In conjunction with a reservoir simulation model, the total geothermal system and its cost can be investigated. Each of the power generation concepts will be discussed in detail in this report. The computer program used (GEOCOST), is designed to effectively simulate these four power generation concepts in use with geothermal energy. The models provide the information necessary to make both a technical and economic assessment of the potential and best use of the concepts. Major efforts of this program are:

- an analytical model of the power plant
- determination of geothermal fluid flow rates, working fluid flow rates, and cooling water requirements for a variety of geothermal temperatures, pressures and water conditions, plant sizes, and types of working fluids
- determination of component sizes
- calculation of geothermal fluid and working fluid thermodynamic and physical properties
- thermodynamic cycle calculations
- overall design performance, with the ability to perform sensitivity analyses on various operating parameters.

This program provides the necessary data to enable the determination of plant component costs and the optimization of the plant cycle from an economic viewpoint.

Each of the power plant models has been constructed in sufficient detail to form a realistic simulation of the plant but is general enough to retain a high degree of flexibility in operation of the model. The computer program has been modularized so that a particular power plant section or subroutine can be expanded to fit a particular power plant design. These subroutines, or modules, can be deleted, modified, or replaced quite easily. To illustrate this flexibility, a few features of these power plant models are listed, with potentials for further expansion.

- (a) The flashed steam model can simulate single, as well as multiple flashing of the geothermal fluid. Currently the multiple lines of flashed steam are injected into different stages of a multi-inlet steam turbine. The simulation of an additional, low-pressure turbine for the final flashing stage can easily be accomplished.
- (b) In the flashed steam model, steam jet ejectors are used to remove noncondensable gases from the turbine exhaust steam. For high noncondensable gas content in flashed steam, alternate techniques for gas removal may be necessary since the steam flow requirements for ejection may be excessive. These alterations can easily be made in the flash steam model.
- (c) A low level, direct contact condenser for the flashed steam model may be inserted as an alternative to the barometric type currently used in the model.

- (d) For geothermal areas having little or no available cooling water, dry cooling towers will be required. A dry cooling tower module can be easily inserted to replace the forced draft wet cooling tower module now in the program.
- (e) Alternate types of heat exchangers for the binary fluid cycle have been proposed. These heat exchangers could be fluidized bed or direct contact types, among others. These could be analyzed as alternatives to the surface heat exchangers now used.
- (f) A simple Rankine cycle is currently modeled in the binary fluid cycle routine. Reheat and regeneration can easily be introduced into the model.
- (g) There are currently eight different working fluid types available for use in the binary fluid cycle model. Additional fluids are readily insertable.

With the current structure of the model and its adaptability to different types of components and methods, the model provides a powerful tool to assess the feasibility of various power plant types for use with geothermal reservoirs. The power plant models can be operated independently to provide data on a particular plant type, or operated as a part of the GEOCOST program to provide information on a total geothermal system and its associated costs.

BACKGROUND

In light of the current world-wide energy situation, the utilization of the immense reserves of geothermal energy is being actively pursued. The energy contained in a geothermal reservoir consists of heat, mainly stored in rocks but also contained in liquid water and/or steam. Currently, the tapping of these large reserves is limited, and the only operating geothermal electrical generation plants either use dry steam or steam flashed from hot water. If geothermal energy is to make a substantial impact on increased electrical generation, methods must be found to successfully use the lower temperature geothermal reservoirs, high salinity reservoirs, geopressured reservoirs, and high temperature, water-deficient reservoirs. A variety of new and advanced power production techniques must be applied toward these characteristically different types of geothermal reservoirs for successful utilization.

The most common method of geothermal power generation to date has been the extraction of steam directly from the ground and expansion of this steam through a standard reaction turbine similar to those in use at fossil-fueled steam power plants. The same turbines are also used with steam that is flashed from hot water drawn from geothermal wells. Geothermal plants of these types are in operation in the United States, Italy, New Zealand, Mexico, and Japan. A second approach to power generation is the binary fluid cycle in which heat is transferred from hot geothermal water to a suitable working fluid. This working fluid is subsequently used in a Rankine cycle for producing electrical power. There exists only one power plant of this type in the world, at the Paratunka geothermal power station in Russia.⁽¹⁾ These three types of power plants

constitute the majority of techniques currently under consideration for large-scale power generation from geothermal energy.

A fourth novel concept, now under development at the Lawrence Livermore Laboratories,⁽²⁾ has recently been proposed for use with highly saline geothermal waters. The method uses the concept of total flow, where energy is extracted from the entire fluid stream. Thermal energy is converted by expansion of the fluid (possibly a two-phase mixture) through a converging-diverging nozzle. This high velocity fluid is then used to drive a modified hydraulic impulse turbine. Another concept using the total geothermal fluid for power generation is the helical rotary screw expander, or Sprankle engine, which is under development by the Hydrothermal Power Company.^(3,4) The screw expander is driven by the expansion of the fluid which in turn drives a generator.

General Ener-Tech, Inc. is proposing a bladeless turbine-generator which converts energy from the fluid to shaft power through the use of the "boundary layer drag" phenomenon.^(5,6) The last of the proposed geothermal prime movers mentioned here is KROV (Keller Roto-Oscillating Vane).⁽⁷⁾ This prime mover is a positive displacement device of the concentric vane and eccentric rotor type. It has not been tested for geothermal applications.

Seven different concepts have been listed here as examples of the diversity of proposed methods for power generation. When the many different types of geothermal reservoirs and power generation schemes are considered, it is difficult to decide which selections should be made for a particular application. Based on available data, the four schemes previously listed in the abstract were chosen for detailed analysis. The next section discusses the modeling of these schemes. The explanation of these power plant models will include examples of required inputs and obtained output of the program.

DISCUSSION

FLASHED STEAM MODEL

Description of Power Plant

The first and most widely used power plant type is the steam power plant. The model developed for this plant can be used for both direct steam and steam flashed from hot water. Since the only difference between these two is the exclusion of the flashing mechanism for the direct steam type, discussions will be based on the flashed steam model only.

The entire process begins at the site of a geothermal well, where hot geothermal fluid is extracted either as a liquid or a mixture containing a small fraction of steam. The fluid is then transported through an insulated piping system to the power plant where steam is generated by flashing. Provisions have also been made in the model to flash the steam at the wellhead and then transport the steam to the plant, but preliminary results have shown that it is most cost-effective to transport the geothermal fluid to the plant prior to flashing. When the geothermal fluid is transported to the power plant, significant loss in energy and pressure of the fluid is inevitable. This phenomenon, as well as an explanation of the entire transmission system will be discussed in detail in a future Battelle-Northwest paper.

At the power plant, any significant amount of steam present in the geothermal fluid can be separated for expansion through the steam turbine. This separated steam will, of course, be at a higher pressure and enthalpy than the flashed steam. The remaining fluid is then rapidly expanded to a lower

pressure in a flash vessel, and a certain mass fraction of saturated steam is extracted from the flash vessel at the flashing pressure.⁽⁸⁾ The mass fraction of the saturated steam generated is a function of the inlet fluid enthalpy and the flashing pressure. The water remaining after passage through the flash vessel is either returned to the geothermal reservoir or again flashed at a pressure lower than the first. In the model there can be as many as four stages of flashing possible. An option exists in the model to pass the steam through a water scrubber to remove particulates after flashing and prior to expansion through the turbine.

The saturated steam is transported from the flash vessel(s) or steam scrubber to the steam turbine, where it is expanded down to back-pressures ranging from 2 to 8 in. of Hg absolute. If multiple streams of steam are present, they are introduced into different stages of the same turbine. After passing through the turbine, the steam is condensed in a barometric direct-contact condenser which is supplied with cooling water. Noncondensable gases contained in the exhaust steam, principally CO₂, are removed from the condenser with a two-stage steam jet ejector. Cooling water for the flashed steam plant is required for the condenser, the turbine oil cooler, and a cooler for the electrical generator, and is provided by an induced-draft evaporative cooling tower. The cooling tower receives the condensed steam and spent cooling water and requires no make-up water since only a portion of the condensed steam is lost through evaporation and drift losses. Any remaining excess water is reinjected into the geothermal reservoir. Internal power consumption of the power plant includes cooling tower fans, cooling water pumps, and make-up pumps for the geothermal fluid. A flow diagram of the entire flashed steam plant is shown in Figure 1.

FIGURE 1. Flash Steam Plant

Plant Components

The following sections describe the major components of the flashed steam plant. Each of these components is placed in the model in a modular form, to provide a flexible format for modification and replacement with different designs.

Flashing

The geothermal fluid is passed through orifices downstream of a water drum in order to form steam at a pressure lower than the original fluid pressure. The steam is then separated from the water through a variety of techniques and sent to the turbine; the water is transported to a silencer. The model is capable of handling four successive stages of flashing.

The calculations for flashing are handled in subroutine or module FLASH. Inputs to this routine include the inlet water temperature, the fluid pressure, steam fraction in the inlet fluid, number of flashing stages, and a flagging variable to indicate whether steam present in the inlet fluid is to be separated. Based on the input conditions, this subroutine determines:

- (a) saturated steam temperature of each flashing stage
- (b) saturated steam pressure of each flashing stage
- (c) enthalpy of the steam for each flashing stage
- (d) mass fraction of steam formed in each stage
- (e) the number of flashing stages.

Steam Scrubber

This component is optional in the flashed steam model. Realistically, geothermal fluids may exist that do not contain sufficient amounts of solids to warrant scrubbing. The scrubber modeled is a water spray type of system designed to remove the majority of particles in the steam without degrading the steam

quality substantially. Water for use in the spray chamber is extracted from the outlet of the steam condenser and passed through a demineralizer. Steam from flashing or a direct source is passed through the spray chamber and subsequently through a cyclone separator to remove any liquid remaining after scrubbing. Water exiting from the spray chamber and cyclone separator is passed through a heat exchanger before being reinjected; thus, some of the waste heat it contains is used to preheat the scrub water entering the spray chamber.

This system is modeled in a subroutine called SCRUBR and requires the following inputs: steam temperature and pressure at the scrubber inlet; the steam mass flow rate; percent of dissolved solids in the steam; and the temperature and pressure of the scrub water leaving the steam condenser. Outputs from the subroutine include the temperature, pressure, and enthalpy of the steam at the scrubber exit; the quantity of scrub water required; and the power required to operate the scrub water pump.

Turbine-Generator

Turbo-generator sets used for geothermal applications with this type of plant are of two types. The first type, condensing steam turbines operating at atmospheric pressure, are used only at the Larderello steam fields in northern Italy.⁽⁹⁾ The second type of turbine are units that exhaust below atmospheric pressure. All the turbines used at Geysers area in California are of this type.⁽¹⁰⁾ A typical turbine used at the Geysers is a 55 MWe double-flow, six-stage unit with a last stage bucket length of 23 in., manufactured by Toshiba.⁽¹¹⁾ Based on all available evidence from existing geothermal steam power plants, an internal turbine efficiency of 75% was used in the model. The type of generator assumed was a three-phase, hydrogen

cooled, AC synchronous generator. In subroutine TURBO, the brake efficiency and overall turbo-generator efficiency are calculated.

Condenser

The type of condenser used in this model is a barometric direct-contact condenser. Exhaust steam from the turbine is condensed upon mixing with cooling water as it cascades down a series of trays. The mixture of cooling water and condensed steam leaves the condenser by gravity down a barometric leg sealed beneath a water surface.⁽¹²⁾ The condenser operates at a terminal temperature difference (TTD) of 6°F or the difference between the saturated temperature of the exhaust steam and the exit cooling water temperature. A small amount of exit cooling water may be used for scrub water (see section on Steam Scrubber). Subroutine SCNDNSR calculates the operational parameters for the steam condenser and requires as input the thermodynamic condition of the exhaust steam, the amount of noncondensable gases present, the mass flow rate of steam, and the entering cooling water temperature. Output of this subroutine is the mass flow rate of cooling water required for condensing.

Gas Ejector

The method selected for removing noncondensable gases from the steam condenser is a two-stage steam jet ejector with inter- and after-condensers. A single ejector consists of a suction chamber, diffuser, and steam nozzle. Motive steam for the ejector is extracted from the main steam line at pressure and is passed through a steam nozzle. The high velocity jet from the nozzle entrains the noncondensable gases, and kinetic energy in the mixture is converted to pressure energy in the diffuser. The noncondensable gases are brought to atmospheric pressure in two of these stages. The computation of the steam to gas ratio is accomplished using Dalton's law of mixtures,

where the total pressure is the sum of the condensible steam pressure and the noncondensable gas pressure. Since the mixture is saturated with steam, the temperature of the mixture fixes the partial pressure of the vapor. (13,14,15) Inter- and after-condensers are used to condense the ejector steam and are of the barometric direct-contact type (see section on Condensers). The two-stage steam jet ejector is modeled in subroutine GSEJCT and calculates the steam-to-gas ratios required for each stage of ejection, knowing the pressure in the steam condenser. Cooling water flow rates for the inter- and after-condensers are calculated in subroutine SCNDNSR.

System Cooling

In addition to the main steam condenser, cooling water is required for the turbine oil cooler and the generator hydrogen cooler. These are standard units, and the required cooling water flow rates are calculated in subroutine QCC.

Cooling Tower

The cooling tower used in the computer model is an induced-draft evaporative cooling tower. Air is drawn through the tower with large propeller-type fans. The tower model is listed in subroutine TOWER and can handle a variety of cooling water inlet temperatures, approaches, and wet and dry bulb air temperatures. Inputs to this subroutine include the cooling water flow rate and temperature at the inlet, the wet and dry bulb air temperatures, humidity ratio, and approach. Output values include the cooling tower air flow rate, evaporation rate, and overall heat load.

Thermodynamic Process Calculations and Computer Logic

A flow sheet for the flashed steam model is shown in Figure 2. The FLASH subroutine is called first to determine steam characteristics. The subsequent subroutines called are

FIGURE 2. Computer Flow Diagram for Flashed Steam Plant Model

shown in the flow diagram. Subroutine SLOSS calculates the internal power consumption of cooling tower fans, cooling water pumps, and any geothermal fluid pumps. These losses are subtracted from the gross kilowatt output of the plant to obtain the net kilowatt output.

In order to determine the steam mass flow rates and other operating characteristics of a particular power plant, many thermodynamic and physical properties of the fluids are required. The thermodynamic properties of steam and water are determined using a computer code called STEAM LIBRARY.⁽¹⁶⁾ This code is based on the "1967 ASME Steam Tables".⁽¹⁷⁾ Physical properties of water and steam are listed in subroutine FLUID as polynomial functions of temperature and are based on data gathered from various sources.^(17,18,19)

The flashed steam plant model has been used to provide information and plants with many various characteristics. Test cases run to date on the flashed steam model have indicated excellent agreement with available data on existing flashed steam geothermal plants. Various parameters studied range as follows:

- (a) plant sizes from 10 to 220 MWe
- (b) geothermal fluid temperatures from 250 to 570°F
- (c) turbine exhaust pressures from 2 to 8 in. of Hg absolute
- (d) weight fractions of steam in wellhead fluid from 0 to 1.0
- (e) numerous weight percents of noncondensable gases and dissolved solids in the geothermal fluid

- (f) individual well flow rates from 100,000 to 1,020,000 lb/hr
- (g) well lives from 5 to 20 years
- (h) number of flashing stages from 0 to 3 stages.

Input to this model is passed from subroutine NAMELIST through the driving subroutine DSTEAM to operate the model. Sample input for a 110 MWe power plant is shown in Figure 3. Output for this power plant is shown in Figure 4 and includes information on cooling water flow rates, internal losses, and overall results.

BINARY FLUID CYCLE MODEL

As stated in the Background, a single binary fluid cycle power plant using geothermal energy is currently in operation. A number of binary power plants are currently in the conceptual stages, however, and may be constructed in the near future. (20,21,22,23) A wide variety of different types of power plants have been proposed using a whole range of different working fluids. It is the intent of this program to model the most fundamental of these types of plants using the most common working fluids.

Description of Power Plant

The binary power plant is displayed in Figure 5 and is described below. In the binary fluid cycle model, the geothermal fluid is extracted from wells distributed throughout the reservoir and transmitted through insulated pipelines to the binary power plant. Through the use of well feed pumps and/or make-up pumps, the geothermal fluid is constantly maintained at a pressure exceeding the saturation pressure to

SUBROUTINE FLASH

WELLHEAD TEMPERATURE	302.0 °F
WELLHEAD PRESSURE	138.1 psia
WELLHEAD ENTHALPY	272.0 Btu/lb
CONDENSER TEMPERATURE	126.1 °F
CONDENSER ENTHALPY	1116.2 Btu/lb
STEAM FRACTION AT INLET	0.0
NUMBER OF FLASHES	2.0

SUBROUTINE DRIVER

GROSS KILOWATT OUTPUT	15,000.0 KWE	
STEAM TEMPERATURE AT TURBINE INLET	255.0 °F	208.0 °F
ABSOLUTE STEAM PRESSURE AT INLET	32.53 psia	13.6 psia
TURBINE EXHAUST PRESSURE	2.0 psia	
WEIGHT FRACTION OF NON-CONDENSIBLE GASES	0.02 %	
WET BULB AIR TEMPERATURE	65.0 °F	
DRY BULB AIR TEMPERATURE	85.0 °F	
BAROMETRIC PRESSURE	29.92 in.Hg	
HUMIDITY RATIO	0.4	
COOLING WATER INLET TEMPERATURE	75.0 °F	
INTERCONDENSER EXIT COOLING WATER TEMPERATURE	110.0 °F	
AFTERCONDENSER EXIT COOLING WATER TEMPERATURE	120.0 °F	
MAIN CONDENSER EXIT COOLING WATER TEMPERATURE	120.0 °F	
OIL COOLER EXIT COOLING WATER TEMPERATURE	120.0 °F	
HYDROGEN COOLER EXIT COOLING WATER TEMPERATURE	120.0 °F	
PRESSURE OF WATER LEAVING GROUND	138.1 psia	
PERCENT DISSOLVED SOLIDS IN STEAM	3.0 %	
WELLHEAD PRESSURE FACTOR	2.0	

FIGURE 3. Input to Flash Steam Model

FLASH CONDITIONS			
1ST STAGE		2ND STAGE	
STEAM TEMPERATURE	255 °F	208 °F	
STEAM PRESSURE	32.5 psia	13.6 psia	
STEAM ENTHALPY	1165.7 Btu/lb	1149.0 Btu/lb	
FRACTION FLASHED TO STEAM	0.0512	0.0488	

COOLING WATER REQUIREMENTS		
INTERCONDENSER FLOW	324,900	lb/hr
AFTERCONDENSER FLOW	286,480	lb/hr
TURBINE OIL COOLER	5,680	lb/hr
HYDROGEN GENERATOR COOLER	21,590	lb/hr
CONDENSER FLOW	9,408,300	lb/hr
TEMPERATURE OF COOLING WATER INTO TOWER	119.7	°F
COOLING TOWER		
COOLING TOWER AIR FLOW RATE	2,579,300	lb/hr
COOLING TOWER EVAPORATION RATE	377,190	lb/hr
WET BULB AIR TEMPERATURE	65.0	°F
DRY BULB AIR TEMPERATURE	80.0	°F
BAROMETER PRESSURE	29.92	in. Hg
HUMIDITY RATIO	0.4	
HEAT LOAD	4.6 x 10 ⁸	Btu/hr

INTERNAL POWER REQUIREMENTS		
KW REQUIRED FOR FANS	373.6	kW
CONDENSATE PUMPS TO COOLING TOWER	210.0	kW
COOLING WATER PUMPS	410.4	kW
COOLING TOWER REINJECTION PUMP	8.63	kW
FLASH REINJECTION PUMPS	0	kW
VACUUM PUMP ON CONDENSER	0	kW
SCRUBBER PUMP	0.573	kW
WATER PRESSURIZING BOOSTER PUMPS	26.7	kW

OVERALL RESULTS			
ACTUAL GROSS STEAM RATE	26.3	38.8	lb/kW-hr
ACTUAL GROSS HEAT RATE	30,630	44,500	Btu/kW-hr
ACTUAL NET STEAM RATE	29.6	43.6	lb/kW-hr
ACTUAL NET HEAT RATE	34,430	50,040	Btu/kW-hr
AVERAGE NET STEAM RATE		36.6	lb/kW-hr
AVERAGE NET HEAT RATE		42,210	Btu/kW-hr
STEAM FLOW TO TURBINE		469,960	lb/hr
TOTAL STEAM FLOW		492,440	lb/hr
TOTAL WELLHEAD FLOW		5,053,790	lb/hr
TOTAL COOLING WATER REQUIREMENTS		10,047,000	lb/hr
COOLING TOWER AIR FLOW RATE		2,579,300	cfm
COOLING TOWER EVAPORATION RATE		377,190	lb/hr
HEAT LOAD OF COOLING TOWER		4.6 x 10 ⁸	Btu/hr
REINJECTION RATE - BLOWDOWN		115,250	lb/hr
FLASH REINJECTION RATE		4,561,350	lb/hr
KW LOSSES FOR EQUIPMENT		1,029.9	kW
NET KILOWATT OUTPUT		13,970	kW

FIGURE 4. Output of Flash Steam Model

FIGURE 5. Sub-Critical Binary Fluid Cycle Power Plant

prevent flashing. Constant overpressurization prevents the loss of the latent heat of vaporization and reduces the temperature drop of the fluid as it passes from the wellhead to the plant. The precipitation of dissolved solids in the wells, piping and pumps is also substantially reduced if flashing to steam is prevented.

At the power plant, the geothermal fluid is passed through a series of heat exchangers, where thermal energy is transferred to a working fluid. These heat exchangers differ in design depending upon whether a super-critical or a sub-critical cycle is used. In the super-critical cycle, the working fluid receives thermal energy while at a pressure exceeding its critical pressure. The sub-critical cycle involves passing through the vapor dome, i.e., at a pressure below the critical pressure. These heat exchangers will be described in the section entitled Heat Exchangers. The geothermal fluid is reinjected into the reservoir in a liquid state after passing through the tube side of the heat exchangers. The working fluid is brought to a superheated state and then expanded through a turbine which is coupled to a generator. A knock-out drum is installed upstream of the turbine to remove any condensate in the working fluid. After passing through the turbine, the working fluid is condensed in a surface condenser supplied with cooling water. The condensed working fluid is then transferred to a condensate receiving tank which holds the reserve working fluid. The fluid is extracted from this tank, repressurized with booster and feed pumps and returned to the heat exchangers for repetition of the cycle.

Cooling water is supplied to the condenser, turbine oil cooler, and hydrogen cooler from an induced-draft evaporative cooling tower. Make-up water to replace fluid lost due to

blow-down, evaporation, and drift is required for the cooling tower. Internal requirements for power include cooling tower fans, cooling water pumps, geothermal fluid feed pumps, and booster and feed pumps for the working fluid.

Plant Components

The following paragraphs describe each of the major components and the methods used in the model to obtain the operating characteristics of these components.

Heat Exchangers

Two different types of thermodynamic cycles can be modeled in this program - the sub- and the super-critical cycles. The heat exchangers used for the sub-critical cycle will be described first. The following computations are based on a number of sources. (24,25,26)

The basic type of heat exchanger used was a counter-flow, single-pass, shell and tube unit with a fixed tube sheet. In the sub-critical cycle, heat exchangers are required for three applications. In the superheater, the hottest geothermal fluid exchanges heat with the working fluid to bring the working fluid from a saturated vapor to a superheated condition. The vaporizer receives the geothermal fluid from the outlet of the superheater to bring the working fluid from a saturated liquid to a saturated vapor state. The preheater receives the geothermal fluid from the outlet of the vaporizer to heat the working fluid up to a saturated liquid state. This procedure is depicted in Figure 6, with typical water and working fluid temperatures attached.

The characteristics of each of these heat exchangers are determined in subroutine SUBHEX. The number of tubes required for the geothermal fluid is based on the fluid flow rate, flow

FIGURE 6. Typical Heat Exchanger Configuration for Binary Cycle Plant

velocity, and tube diameter, and is calculated in function TUBEN. For the superheater and vaporizer, the inside and outside heat transfer coefficients (for the water and working fluid, respectively) are based on the Dittus-Boelter equation for fully developed turbulent flow.⁽²⁷⁾ These calculations are performed in function HTCDEF. The inside heat transfer coefficient for the vaporizer is also calculated using the same equation and function routine. The outside (boiling) heat transfer coefficient for the working fluid in the vaporizer is determined using Chen's correlation based on two-phase fluid properties.⁽²⁸⁾ The overall heat transfer coefficient for each of the heat exchangers is determined on the basis of these coefficients with the following equation, programmed in function OHTCOE.

$$U = \frac{1}{\frac{A_o}{A_i} \cdot \frac{1}{h_i} + \frac{A_o \ln(r_o/r_i)}{2\pi K \ell} + \frac{1}{h_o} + ff}$$

where U = overall heat transfer coefficient, Btu/hr·ft²·°F

A_o = outside area, ft²

A_i = inside area, ft²

h_i = inside heat transfer coefficient, Btu/hr·ft²·°F

r_i = inside tube radius, ft

r_o = outside tube radius, ft

K = thermal conductivity of tube, Btu/hr·ft·°F

ℓ = tube length, ft

h_o = outside heat transfer coefficient, Btu/hr·ft²·°F

ff = fouling factor, hr·ft²·°F/Btu.

Knowing the amount of overall heat transfer desired, the heat transfer area and heat transfer length can be calculated in functions HTA and HTL respectively.

The heat exchanger used for the super-critical cycle was also a counter-flow, single-pass, shell and tube heat exchanger. This heat exchanger was handled in subroutine SUPHEX. The major difference in operation of this heat exchanger and the one just described is the fact that no boiling section exists. This allows the calculation of the entire heat exchange from the working fluid pump outlet to the turbine inlet to be made as if there were one heat exchanger. However, this assumption is not accurate since the change in thermodynamic and physical properties of the working fluid is quite variable and non-linear in the super-critical region above the vapor dome. For this reason, the super-critical heat exchanger was assumed to be a series of five different heat exchangers. Tube numbers and heat transfer coefficients were calculated for each section, and the heat transfer areas and lengths were summed over all heat exchangers to obtain a single size and length exchanger.

Inputs to the subroutine SUBHEX and SUPHEX are:

- (a) geothermal fluid flow rate
- (b) working fluid flow rate
- (c) geothermal fluid temperatures at the inlet and outlet of each heat exchanger section
- (d) working fluid temperatures and enthalpies at the inlet and outlet of each heat exchanger section
- (e) fouling factors
- (f) tube inside and outside diameters
- (g) water velocities in heat exchanger tubes
- (h) type of working fluid.
- (i) pinch point.

Obviously, not all of the above information is required for computational purposes; different combinations of the above inputs can be used. Output of these subroutines includes:

- (a) number of heat exchanger tubes
- (b) equivalent flow diameter for the shell side
- (c) inside heat transfer coefficient
- (d) outside heat transfer coefficient
- (e) overall heat transfer coefficient
- (f) heat exchanger area
- (g) heat exchanger length.

Contacts with manufacturers of heat exchangers pointed out that the overall heat transfer coefficients initially calculated in this model were somewhat higher (and consequently, the total heat transfer areas lower) than those normally quoted for similar applications. These differences were most likely caused by variability in thermodynamic and physical property data for working fluids and conservative estimates for fouling. The model was adjusted for these factors and now a close agreement on heat transfer coefficients exists.

Turbine Generator

The turbine is an isentropic expansion device with an associated efficiency factor. The internal efficiency for the turbine has been estimated to be 80%. From this value, the brake and overall efficiency of the turbo-generator set can be determined in subroutine WFFRT. With the inlet and exhaust enthalpies of the working fluid known, the isentropic and actual Rankine efficiencies are determined.

Only one turbine has been manufactured in the U.S. for specifically geothermal applications. This turbine has been designed for isobutane as the working fluid and is a 9000 kWe three-stage, radial-flow turbine. It was manufactured by the York Division of Borg-Warner Corporation for Magma Energy, Inc. (29)

Condenser

The working fluid condenser used for this model is similar to the type of heat exchangers described in the above section-- a counter-flow, single-pass, shell and tube heat exchanger. Calculations for the condenser are performed in subroutine CNDNSR. Inputs to this routine include inlet and outlet cooling water temperatures, working fluid enthalpies, temperatures and pressures, type of working fluid and its flow rate, and tube diameters. The output of this subroutine is the condenser cooling water flow rate, condenser area and length, and the individual heat transfer coefficients.

Pumps

The type of pump used in this model for feeding the geothermal fluid from the wells is a turbine pump. This is a pump driven from the surface with turbine bowls submersed in the fluid down in the well. An electric motor is used to drive the pump. The power consumed by the geothermal feed pump is determined in subroutine LOSSES and is a function of flow rate and amount of pressurization required to prevent flashing.

Pumps used for the cooling water and the working fluid are standard units, and power consumptions are determined in a manner similar to that used for the geothermal feed pump. These pumps are assumed to be driven by electric motors even

though a number of proposed power plants use a small vapor turbine to drive the working fluid feed pump. These calculations are made in subroutines PUMP, LOSSES, and BINARY.

System Cooling

System cooling is handled in the same manner as in the flashed steam cycle and is modeled in subroutine ACWFRT. System cooling is discussed on page 12.

Cooling Tower

The same type of cooling tower is used in the binary fluid cycle and the flashed steam cycle. The cooling tower subroutine is described in detail on page 12.

Thermodynamic Calculations

Due to the presence of two separate fluids, one being in a closed loop, the thermodynamic calculations for the binary cycle are more complex than those for the flashed steam cycle. Four separate topics are discussed in this section;

- (a) selection of working fluid
- (b) selection of thermodynamic cycle
- (c) manipulation of thermodynamic equations of state
- (d) determination of thermodynamic cycle state points.

These topics are discussed separately but are intertwined in the actual computer model. They are separated only for purposes of explanation.

Selection of Working Fluid

Many fluids have been suggested for use in binary fluid cycles for geothermal applications. The list is long and includes among others: isobutane, n-butane, R-11*, R-12, R-21,

* The R- prefix stands for a halogenated hydrocarbon or "Freon."

R-22, R-33, propane, ammonia, methane, propylene, R-113, R-114, isopentane, and R-33a. There are numerous criteria to be considered prior to the selection of the proper working fluid.

- (1) cost
- (2) molecular weight
- (3) critical properties
- (4) stability at high temperatures
- (5) corrosiveness
- (6) thermal efficiency
- (7) heat transfer properties
- (8) transport properties
- (9) toxicity
- (10) flammability.

With the available information on these criteria for the listed fluids and some engineering judgment, the following eight fluids were selected for use in the binary cycle model:

- (1) isobutane
- (2) n-butane
- (3) R-11
- (4) R-12
- (5) R-21
- (6) R-22
- (7) R-113
- (8) R-114

Provisions have been made to add additional working fluids as the demand for their use arises, subject to the availability of data on fluid properties.

Since GEOCOST is concerned with the economic as well as technical factors, the most appropriate working fluid for a particular set of conditions is difficult to predict without first operating the GEOCOST model for all fluids.⁽³⁰⁾ Once

this information is at hand, a prediction can be made on the best working fluid for a particular input. Currently isobutane and n-butane appear to be slightly better at the lower geothermal fluid temperatures. At higher temperatures, R-21 has emerged as a prime candidate. These conclusions are not final, however, since 1) noneconomic and nontechnical fluid characteristics such as stability and toxicity have not yet been considered and 2) the comparison was made on only the eight aforementioned fluids. In its final state, the computer model will be capable of predicting the most suitable working fluid for a particular set of input geothermal fluid data.

Selection of Thermodynamic Cycle

In the Description of Power Plant section on binary fluid cycles, two different thermodynamic cycles were mentioned. Although there are many differences in these two types, both sub-critical and super-critical cycles have been proposed for use with relatively low-temperature resources.^(31,32) Figures 7 and 8 illustrate the basic sub-critical and super-critical Rankine cycles, respectively.

FIGURE 7. Sub-Critical Cycle

FIGURE 8. Super-Critical Cycle

In these figures, the ordinate is pressure and the abscissa is enthalpy. The processes that make up these cycles are as follows:

- 3-1: constant pressure addition of heat to the working fluid from the geothermal fluid in the heat exchangers
- 1-2: reversible adiabatic (isentropic) expansion of the working fluid in the turbine (the dotted line depicts the actual path, due to turbine inefficiencies)
- 2-2SATL: constant pressure rejection of heat from the working fluid to cooling water in the condenser
- 2SATL-3: compression of the working fluid with feed pumps.

The thermodynamic cycle selected for a particular set of conditions is determined through an iterative process based on the heat exchanger approach, heat exchanger pinch point, and, of course, the turbine inlet pressure. If the iterative process tends to exceed a turbine inlet pressure of 0.9 times the critical pressure at the maximum pinch point, the super-critical cycle is selected. In cases other than this, the iteration converges on a sub-critical cycle. For flexibility, the super-critical cycle can arbitrarily be selected. This iterative technique is explained in greater detail in the section on Determination of Thermodynamic Cycle State Points.

Manipulation of Thermodynamic Equations of State

In order to accurately predict the thermodynamic properties of various working fluids, complex equations of state were used rather than tabulated data. This was done to

minimize computer memory and to insure consistency and accuracy of the calculated properties. The Martin-Hou equation of state was selected for the halogenated hydrocarbons (Freons) and the Starling equation of state was used for the hydrocarbons (isobutane, n-butane). (33,34,35)

The Martin or Martin-Hou equation of state was first introduced in 1967 and has since evolved from its simplest form to the present expression, containing 21 different virial coefficients.⁽³⁶⁾ This calculation is performed in subroutine EQSTAP.

$$P = \frac{RT}{V-b} + \frac{A_2 + B_2T + C_2\exp(-KT/T_c)}{(V-b)^2} + \frac{A_3 + B_3T + C_3\exp(-KT/T_c)}{(V-b)^3} + \frac{A_4 + B_4T + C_4\exp(-KT/T_c)}{(V-b)^4} + \frac{A_5 + B_5T + C_5\exp(-KT/T_c)}{(V-b)^5} + \frac{A_6 + B_6T + C_6\exp(-KT/T_c)}{\exp(\alpha V)(1 + C\exp(\alpha V))}$$

where P = pressure, psia

V = specific volume, ft³/lb

T = temperature, °R

T_c = critical temperature, °R

The remaining letters correspond to constants for a particular Freon. A reduced form of this equation can be used to determine the saturation pressure at a given temperature;

$$\log_{10} P = A + \frac{B}{T} + C \log_{10} T + DT + E \left(\frac{F-T}{T} \right) \log_{10} (F-T)$$

The saturation pressure calculation is located in subroutine VPRESA. The equations for enthalpy (H) and entropy (S) are also based on the basic equation of state, and are as follows;

$$\begin{aligned}
H = & aT + \frac{bT^2}{2} + \frac{cT^3}{3} + \frac{dT^4}{4} - \frac{f}{T} + JPV + J \left[\frac{A_2}{V-b} + \frac{A_3}{2(V-b)^2} \right. \\
& + \frac{A_4}{3(V-b)^3} + \frac{A_5}{4(V-b)^4} + \frac{A_6}{\alpha} \left(\frac{1}{\exp(\alpha V)} - C \ln \left(1 + \frac{1}{C \exp(\alpha V)} \right) \right) \Bigg] \\
& + J \exp(-KT/T_c) (1 + KT/T_c) \left[\frac{C_2}{(V-b)} + \frac{C_3}{2(V-b)^2} + \frac{C_4}{3(V-b)^3} \right. \\
& + \frac{C_5}{4(V-b)^4} + \frac{C_6}{\alpha \exp(\alpha V)} - \frac{C_6 C'}{\alpha} \ln \left(1 + \frac{1}{C' \exp(\alpha V)} \right) \Bigg] + X
\end{aligned}$$

where $J = 0.1851$

X = reference enthalpy (-40°F for the saturated liquid)

$$\begin{aligned}
S = & a \ln T + bT + \frac{cT^2}{2} + \frac{dT^3}{3} - \frac{f}{2T^2} + JR \ln (V-b) \\
& - J \left[\frac{B_2}{V-b} + \frac{B_3}{2(V-b)^2} + \frac{B_4}{3(V-b)^3} + \frac{B_5}{4(V-b)^4} \right. \\
& + \frac{B_6}{\alpha} \left(\frac{1}{\exp(\alpha V)} - C \ln \left(1 + \frac{1}{C \exp(\alpha V)} \right) \right) \Bigg] \\
& + \frac{JK}{T_c} \exp(-KT/T_c) \left[\frac{C_2}{V-b} + \frac{C_3}{2(V-b)^2} + \frac{C_4}{3(V-b)^3} + \frac{C_5}{4(V-b)^4} \right. \\
& + \frac{C_6}{\alpha \exp(\alpha V)} - \frac{C_6 C'}{\alpha} \ln \left(1 + \frac{1}{C' \exp(\alpha V)} \right) \Bigg] + Y
\end{aligned}$$

where R = universal gas constant/molecular weight

Y = reference entropy (-40°F)

The enthalpy and entropy calculations are performed in subroutines HVAP and SVAP, respectively. Expressions for liquid density, heat of vaporization, and heat capacity are also included in subroutine form, but are not presented here.

(Consult Reference 33 for details.) All working fluids using the Martin-Hou equation of state are based on a datum of the saturated liquid at -40°F. At this point both the enthalpy and entropy are defined to be equal to zero.

These equations are used in the model to describe the state of the following working fluids-- R-11, R-12, R-21, R-22, R-113, and R-114. There are numerous other working fluids, including ammonia, which can be described thermodynamically with the Martin-Hou equation, thus making a relatively easy task of inserting additional working fluids. A detailed comparison has not been made between tabulated data and results of the stated equations, but results have been in good agreement when checked at random.

The Starling equation of state has been recently developed for hydrocarbons and was used in this model because of its accuracy at high reduced pressures and its ability to handle properties in the compressed liquid region. This equation of state is quite similar to the Benedict-Webb-Rubin equation.⁽³⁷⁾ This equation is computed in subroutine PEQST and is shown below;

$$P = \rho RT + \left(B_0 RT - A_0 - \frac{C_0}{T^2} + \frac{D_0}{T^3} - \frac{E_0}{T^4} \right) \rho^2 \\ + \left(bRT - a - \frac{d}{T} \right) \rho^3 + \alpha \left(a + \frac{d}{T} \right) \rho^6 \\ + \frac{c\rho^3}{T^2} (1 + \gamma\rho^2) \exp(-\gamma\rho^2)$$

where P = pressure, psia

ρ = molar density, lb.mole/ft³

R = universal gas constant/molecular weight

T = temperature, °R

The remaining letters correspond to constants for a particular working fluid. The expressions for enthalpy (H) and entropy (S) are located in subroutines HEQST and SEQST respectively and are as follows;

$$H = \left(B_0 RT - 2A_0 - \frac{4C_0}{T^2} + \frac{5D_0}{T^3} - \frac{6E_0}{T^4} \right) \frac{\rho}{\phi} \\ + \frac{1}{2} \left(2bRT - 3a - \frac{4d}{T} \right) \frac{\rho^2}{\phi} + \frac{1}{5} \alpha \left(6a + \frac{7d}{T} \right) \frac{\rho^5}{\phi} \\ + \frac{c}{\gamma \phi T^2} \left[3 - \left(3 + \frac{1}{2} \gamma \rho^2 - \gamma^2 \rho^4 \right) \exp(-\gamma \rho^2) \right] + H_0$$

where ϕ is the conversion from Btu/lb to psia·ft³/lb mole and H_0 is the ideal gas enthalpy calculated in subroutine HIDEAL,

$$H_0 = A_0 + A_1 T + A_2 T^2 + A_3 T^3 + A_4 T^4 + A_5 T^5$$

$$S = - \frac{R}{\phi} \ln(\rho RT) - \left(B_0 R + \frac{2C_0}{T^3} - \frac{3D_0}{T^4} + \frac{4E_0}{T^5} \right) \frac{\rho}{\phi} \\ - \frac{1}{2} \left(bR + \frac{d}{T^2} \right) \frac{\rho^2}{\phi} + \frac{\alpha d \rho^5}{5 T^2 \phi} \\ + \frac{2C}{\gamma T^3 \phi} \left[1 - \left(1 + \frac{1}{2} \gamma \rho^2 \right) \exp(-\gamma \rho^2) \right] + S_0$$

S_0 is the ideal gas entropy calculated in subroutine SIDEAL.

$$S_0 = B_0 + B_1 T + B_2 T^2 + B_3 T^3 + B_4 T^4 + B_5 T^5$$

The calculation of saturation pressure is more complex when using the Starling equation than the Martin-Hou equation. An iteration is required whereby an initial guess is made for

P_{sat} , and then the density roots are found from the equation of state. Caution must be exercised since there is a possibility of three or more density roots to this equation. The largest root is the saturated liquid density, and the smallest root is the saturated vapor density. These two density roots are then used to calculate the liquid and vapor fugacities (f) in sub-routine FUG.

$$\begin{aligned} \ln f = & \ln(\rho RT) + 2 \left(B_0 RT - A_0 - \frac{C_0}{T^2} + \frac{D_0}{T^3} + \frac{E_0}{T^4} \right) \frac{\rho}{RT} \\ & + \frac{3}{2} \left(bRT - a - \frac{d}{T} \right) \frac{\rho^2}{RT} + \frac{6a}{5} \left(a + \frac{d}{T} \right) \frac{\rho^5}{RT} \\ & + \frac{C}{\gamma RT^3} \left[1 - \left(1 - \frac{1}{2} \gamma \rho^2 - \gamma^2 \rho^4 \right) \exp(-\gamma \rho^2) \right] \end{aligned}$$

At saturation, the liquid fugacity is equal to the vapor fugacity. The ratio of these fugacities can be used to generate a better guess for P_{sat} ; this is done until the fugacities converge within a specified tolerance. The final P_{sat} used is then the saturation pressure.

All working fluids using the Starling equation are based on a datum of carbon (graphite) and hydrogen (gas) at 0 psia and 0 °R. At this point, the enthalpy and the expression $S + R \ln P$ are set equal to zero. The enthalpy of the working fluid at this temperature and pressure is equal to the standard heat of formation from carbon (graphite) and hydrogen (gas). For isobutane, this enthalpy is -783.04 Btu/lb; for n-butane it is -733.1 Btu/lb. For this reason the enthalpies of these fluids are negative in the normal operating range.⁽³⁸⁾

The set of equations just described can describe accurately the thermodynamic properties of isobutane and n-butane.

Other hydrocarbons can be handled by the Starling equation, thus making the addition of fluids such as propane, methane, propylene, and ethylene quite easy.

Calculation of Thermodynamic Cycle State Points

Armed with the equations of state discussed in the previous section, a thermodynamic cycle can be described for a particular set of input conditions. Rather than explaining in detail the technique used to calculate each of the state points in the cycle, a more general discussion based on the computer flow logic will be made.

A flow diagram illustrating the determination of the state points in the thermodynamic cycle is shown in Figure 9. Only the sub-critical cycle will be discussed here since the super-critical cycle is computed in a similar but less complex manner. In general, the method used to determine the final state points and thus the working fluid operating pressure is an iterative one, and is based on heat exchanger approach, heat exchanger pinch point, and final geothermal fluid temperature.

The first step in this procedure is the loading of coefficients required for the equations of state. Only those coefficients pertaining to the chosen working fluid are loaded. This procedure is done in subroutine SETCON. If the chosen working fluid is a hydrocarbon, the SETCON subroutine then calls subroutine BUTANE, which is the driver for the state point calculations. Driver subroutine FREON is called if the working fluid is a Freon. For purposes of discussion, it is assumed the chosen working fluid is a hydrocarbon. The calculation of state point values then continues in the following stepwise fashion, finally converging in a final set. It

FIGURE 9. Computer Flow Diagram for Determination of Thermodynamic State Points

will be helpful to refer to Figure 10 to recognize specific points in the cycle.

- (1) The minimum working fluid temperature, $T2SATL$, is fixed on the basis of the cooling water temperature. The saturation pressure, $P2$, is calculated in subroutine PTWO in the manner described in the section on equations of state. In the process of this calculation, the largest density root, or $RHOL_2$, is determined and is the density of the saturated liquid at $P2$. The enthalpy, $H2SATL$, is determined using subroutines HEQST and HIDEAL.

FIGURE 10. Sub-Critical Cycle State Points

- (2) For the iteration to follow, an initial value for P_1 is selected. Other known values are W_{T1} , T_1 and preheater pinch point $PINCHP$.
- (3) The density at state 1 is calculated iteratively using subroutine $PEQST$ and the assumed value of P_1 . The enthalpy and entropy, H_1 and S_1 , are determined using the subroutines $HEQST$, $HIDEAL$, $SEQST$, and $SIDEAL$.
- (4) The thermodynamic properties at state "2EST" are calculated on the basis of an isentropic expansion ($S_{2EST} = S_1$) through the turbine. The temperature and density, T_{2EST} and D_{2EST} , are calculated by simultaneously solving the equation of state and the entropy equation. A program called $RQNWT$ was used for this purpose. This program solves systems of nonlinear, algebraic or transcendental equations. Once temperature and density are known the enthalpy, H_{2EST} , can be determined using subroutines $HEQST$ and $HIDEAL$.
- (5) The working fluid flow rate, Q_{ISO} , can now be calculated using subroutine $WFFRT$, based on plant size, H_1 , H_{2EST} , H_{2SATL} and overall turbine efficiency. The actual exhaust enthalpy, H_2 , is also determined in $WFFRT$.
- (6) The saturation temperature, $T_4 = T_5$, is calculated using a modification of the iteration referred to in Step (1).
- (7) Pressures P_4 and P_5 are calculated based on P_1 . Enthalpies H_4 and H_5 are calculated using T_4 and T_5 and the subroutines $HEQST$ and $HIDEAL$. Entropies S_4

and S5 are calculated using subroutines SEQST and SIDEAL.

- (8) Pressure P3 is calculated on the basis of P1. The temperature and enthalpy, T3 and H3, are determined in subroutine PUMP and are based on the pump work done between state "2SATL" and state "3".
- (9) Water temperatures WT3 and WT4 are determined through a heat balance formulation based on the working fluid states and a specified preheater pinch point PINCHP.
- (10) The original pressure P1 is adjusted and Steps (2) through (9) are reiterated until the proper value of WT4 is obtained for specified conditions.
- (11) After the final value of P1 is found, the actual exhaust temperature and density, T2 and D2, are determined through a procedure similar to Step (4). In this case, however, the enthalpy equation is used in place of the entropy equation.

All pertinent thermodynamic information is known after these steps are completed. The remaining characteristics of the binary cycle plant can now be determined. Before this procedure is explained, the methods used to determine the physical properties of the various working fluids will be discussed.

Determination of Physical Properties

In the section on Thermodynamic Process Calculations for flashed steam plants, a subroutine called FLUID was referenced. This routine is a compilation of as much data as could be acquired in the literature for physical properties of various fluids used in the computer model. Each physical property for a particular fluid is expressed as a polynomial function of temperature in the most elementary form,

$$\text{Property Value} = K_0 + K_1T + K_2T^2 + K_3T^3 \text{ for } T_x < T < T_y$$

These expressions are valid for the temperature range T_x to T_y . This range is a function of available data. These data were acquired through a search of numerous reference texts and publications. (39,40,41,42,43,44,45,46) When a physical property is desired, the following statement is made,

$$\text{Property Value} = \text{FLUID (I,J,T)}$$

where T = temperature at which the property is required, °F

I = 1,13 - Property Index

- = 1 - specific heat (liquid)(Btu/lb/°F)
- = 2 - thermal conductivity (liquid)(Btu/hr/ft/°F)
- = 3 - density (liquid)(lb/ft³)
- = 4 - viscosity (liquid)(lb_m/ft/hr)
- = 5 - specific heat (gas)(Btu/lb/°F)
- = 6 - thermal conductivity (gas)(Btu/hr/ft/°F)
- = 7 - density (gas)(lb/ft³)
- = 8 - viscosity (gas)(lb_m/ft/hr)
- = 9 - heat of vaporization (Btu/lb)
- = 10 - critical pressure (psia)
- = 11 - critical temperature (°F)
- = 12 - surface tension (lb_f/ft)
- = 13 - molecular weight (lb/lb_{mole})

J = 1,9 - Fluid Index

- = 1 - water
- = 2 - isobutane
- = 3 - n-butane
- = 4 - R-11
- = 5 - R-12

- = 6 - R-21
- = 7 - R-22
- = 8 - R-113
- = 9 - R-114

Within the specified temperature range, there is good agreement between the actual and predicted values. If a property value is requested outside the specified temperature range, a warning is printed out to indicate a possible erroneous result.

Calculation of Operating Characteristics

A logic flowsheet for the operating characteristics of the binary cycle plant is shown in Figure 11. Once the thermodynamic cycle state points are determined in subroutines SETCON and BUTANE or FREON, the flow rates, sizes and other pertinent information are determined for the various plant components. Subroutine LOSSES calculates the internal power consumption of cooling tower fans, cooling water pumps, working fluid pumps and geothermal fluid pumps. The net kilowatt output of the binary cycle plant is the difference of the gross power generated and the total internal power consumed.

The binary fluid cycle modeled in this computer program is the most simple of all possible types. In other words, no provisions have been made for reheat or regeneration in the cycle. In most cases, the quality of the low pressure working fluid does not warrant reheat. Also, the relatively small temperature range in which this Rankine cycle operates does not make regeneration desirable. These perturbations can be added to the model at a later date if there is question concerning these conclusions. This simple binary cycle plant does not necessarily operate at maximum thermal efficiency when a turbine inlet pressure is selected through iteration, since the object

- BINARY PLANT CONTROL ROUTINE
- BINARY PLANT MODEL
- THERMODYNAMIC ROUTINE
- GEOTHERMAL FLUID FLOW RATE DETERMINATION
- AUXILIARY COOLING WATER CALCULATION
- SUPERCRITICAL HEAT EXCHANGER MODEL
- SUB-CRITICAL HEAT EXCHANGER MODEL
- SURFACE CONDENSER MODEL
- COOLING TOWER MODEL
- INTERNAL POWER CONSUMPTION MODEL

FIGURE 11. Computer Flow Diagram For Binary Fluid Cycle Plant Model

of the overall GEOCOST program is to optimize the entire geothermal system based on cost. In conclusion, this binary cycle plant model provides a sound basis for investigation of the technical and economic feasibility of applying this type of power generation system to a geothermal resource.

Numerous cases have been run with the binary cycle plant model for varying input parameters. Output from these cases generally agree well with available data on similar power plants. The following are the ranges of parameter variation:

- (a) operation of all available working fluids
- (b) plant sizes from 10 to 220 MWe
- (c) geothermal water temperatures from 200-600°F
- (d) operation of both sub- and super-critical cycles
- (e) varying heat transfer coefficients for the water/working fluid heat exchangers
- (f) varying heat exchanger approaches and pinch points
- (g) cooling water temperatures from 60 to 85°F
- (h) varying atmospheric conditions
- (i) varying geothermal fluid pressures
- (j) individual well flow rates from 100,000 to 1,000,000 lb/hr
- (k) well lives from 5 to 20 years
- (l) varying tube diameters for heat exchangers.

Input to this model is passed from the subroutine NAMELIST through the driving subroutine DBINARY to operate this model. Sample input for a 55 MWe power plant using isobutane as a working fluid is shown in Figure 12. Output for this power plant is shown in Figure 13 and includes information on thermodynamic state points, heat exchanger characteristics, condenser and cooling tower characteristics, internal losses, and overall results.

TYPE OF PLANT	BINARY
SUPERCRITICAL CYCLE	FALSE
FLUID	ISOBUTANE
PRESSURE FACTOR	2.0
FLUID TEMPERATURE	180 ⁰ C
FLOW RATE PER WELL	250,000 LB/HR
PLANT SIZE	64,000 KWE
WELL SPACING	20 ACRES
AVERAGE WELL LIFE	10 YEARS
RESERVOIR SIZE	275,000 KW
WET BULB AIR TEMPERATURE	79.0 ⁰ F
DRY BULB AIR TEMPERATURE	111.0 ⁰ F

FIGURE 12. Input to Binary Cycle Plant Model

THE WORKING FLUID IS ISOBUTANE

THE CYCLE IS SUB-CRITICAL

COOLING WATER

TURBINE OIL COOLER

INLET TEMPERATURE

87.0

⁰F

OUTLET TEMPERATURE

120.0

⁰F

FLOW RATE

33,050

lb/hr

HYDROGEN COOLER

INLET TEMPERATURE

87.0

⁰F

OUTLET TEMPERATURE

120.0

⁰F

FLOW RATE

125,590

lb/hr

TEMPERATURES

T1

325.4

⁰F

T2 EST

212.4

⁰F

T2

228.5

⁰F

T2 SATL

100.0

⁰F

T3

103.9

⁰F

T4

256.6

⁰F

T5

256.6

⁰F

PRESSURES

P1

450.0

psia

P2

72.5

psia

P3

450.0

psia

P4

450.0

psia

P5

450.0

psia

FIGURE 13. Output of Binary Cycle Plant Model

Figure 13 (continued)

SPECIFIC VOLUMES		
V1	0.233	ft ³ /lb
V2 EST	1.595	ft ³ /lb
ENTHALPY		
H1	-558.5	Btu/lb
H2	-588.0	Btu/lb
H2 EST	-596.1	Btu/lb
H3	-783.9	Btu/lb
H4	-664.3	Btu/lb
H5	-611.4	Btu/lb
ENTROPY		
S1	1.254	Btu/lb/°F
S2 EST	1.254	Btu/lb/°F
WATER TEMPERATURES		
WT1	355.5	°F
WT2	313.8	°F
WT3	271.6	°F
WT4	180.0	°F
ESTWT4	176.8	°F
TURBINE EFFICIENCY		
IDEAL RANKINE EFFICIENCY	0.165	
ACTUAL RANKINE EFFICIENCY	0.130	
OVERALL TURBINE EFFICIENCY	0.786	

HEAT EXCHANGERS SUB-CRITICAL CYCLE

	SUPER HEATER	VAPORIZER	PREHEATER	
WORKING FLUID				
INLET TEMPERATURE	256.6	256.6	103.9	°F
INLET PRESSURE	450.0	450.0	450.0	psia
INLET ENTHALPY	-611.4	-664.3	-783.9	Btu/lb
OUTLET TEMPERATURE	325.5	256.6	256.6	°F
OUTLET PRESSURE	450.0	450.0	450.0	psia
OUTLET ENTHALPY	-558.5	-611.4	-664.3	Btu/lb
GEOTHERMAL FLUID				
INLET TEMPERATURE	355.5	313.8	271.5	°F
INLET PRESSURE	224.2	199.2	174.2	psia
OUTLET TEMPERATURE	313.8	271.6	176.8	°F
OUTLET PRESSURE	199.2	174.2	149.2	psia
INSIDE TUBE DIAMETER	0.1	0.1	0.1	ft
OUTSIDE TUBE DIAMETER	0.1104	0.1104	0.1104	ft
NUMBER OF TUBES	706.0	706.0	683.0	
EQUIVALENT OUTSIDE DIAMETER	0.1079	0.1079	0.1079	ft
INSIDE HEAT TRANSFER COEFFICIENT	2,846.7	2,263.5	1,872.6	Btu/hr/ft ² /°F
OUTSIDE HEAT TRANSFER COEFFICIENT	622.7	2,568.1	763.0	Btu/hr/ft ² /°F
OVERALL HEAT TRANSFER COEFFICIENT	239.5	326.9	245.0	Btu/hr/ft ² /°F
HEAT TRANSFER AREA	38,705	37,953	98,550	ft ²
HEAT TRANSFER LENGTH	158.1	155.0	416.0	ft

Figure 13 (continued)

MAIN CONDENSER

WORKING FLUID

INLET TEMPERATURE	228.5	°F
INLET PRESSURE	72.5	psia
INLET ENTHALPY	-588.0	Btu/lb
OUTLET TEMPERATURE	100.0	°F
OUTLET PRESSURE	72.5	psia
OUTLET ENTHALPY	-783.9	Btu/lb

COOLING WATER

INLET TEMPERATURE	87.0	°F
INLET PRESSURE	40.0	psia
OUTLET TEMPERATURE	105.0	°F
OUTLET PRESSURE	35.0	psia
INSIDE TUBE DIAMETER	0.075	ft
OUTSIDE TUBE DIAMETER	0.083	ft
NUMBER OF TUBES	1,914	
EQUIVALENT OUTSIDE DIAMETER	0.082	ft
INSIDE HEAT TRANSFER COEFFICIENT	741.1	Btu/hr/ft ² /°F
OUTSIDE HEAT TRANSFER COEFFICIENT	5,639	Btu/hr/ft ² /°F
OVERALL HEAT TRANSFER COEFFICIENT	261.7	Btu/hr/ft ² /°F
HEAT TRANSFER AREA	114,030	ft ²
HEAT TRANSFER LENGTH	227.7	ft

SUBROUTINE TOWER -- COOLING TOWER

COOLING TOWER AIR FLOW RATE	2,688,800	lb/hr
COOLING TOWER EVAPORATION RATE	1,312,600	lb/hr
WET BULB AIR TEMPERATURE	79.0	°F
DRY BULB AIR TEMPERATURE	111.0	°F
BAROMETRIC PRESSURE	29.92	in. Hg
HUMIDITY RATIO	0.25	
HEAT LOAD	1.41 x 10 ⁹	Btu/hr

COOLING WATER REQUIREMENTS

TURBINE OIL COOLER	33,050	lb/hr
HYDROGEN COOLER	125,590	lb/hr
FLOW RATE IN CONDENSER	81,251,500	lb/hr
MAKEUP FLOW RATE TO COOLING TOWER	1,312,600	lb/hr

KILOWATT LOSSES

FANS	519.3	kW
CONDENSATE PUMPS TO COOLING TOWER	1,802.2	kW
COOLING WATER PUMPS	3,695.3	kW
REINJECTION PUMPS	0	kW
BOOSTER PUMPS TO OVERPRESSURIZE WATER	152.0	kW
DOWNHOLE PUMPS TO PREVENT TWO PHASE FLOW	282.0	kW
BOOSTER PUMPS	241.4	kW
FEED PUMPS	4,314.7	kW

OVERALL RESULTS

GROSS KILOWATT OUTPUT OF THE PLANT	64,000	kW
GEOTHERMAL WATER FLOW RATE	9,221,850	lb/hr
FLOW RATE OF ISOBUTANE	7,393,100	lb/hr
COOLING WATER FLOW RATE	81,410,150	lb/hr
TOTAL KILOWATT LOSSES	13,545.0	kW
NET KILOWATT OUTPUT	50,455	kW
NET HEAT RATE	51,797	Btu/kW-hr

TOTAL FLOW MODEL

Many geothermal reservoirs contain fluids which are high enough in heat content to warrant the construction of a power plant, but are very high in dissolved solids. For example, the Salton Sea geothermal area is enormous, but the water contains up to 30 wt percent of dissolved salts. If a flashed steam or binary cycle plant were fueled with this brine, severe corrosion problems would have to be overcome. One proposed solution is the use of a concept known as "total flow" which is currently under development at the Lawrence Livermore Laboratory. Though still in the laboratory state and needing a significant amount of development, it is a concept of real potential. For this reason, it has been included as one of the power plant models in this program. This model is not as complete or as detailed as the flashed steam or binary cycle models since no real operating data or detailed designs exist.

Description of Power Plant

A flow diagram of the total flow plant is shown in Figure 14. Geothermal fluid that is a normally two-phase mixture of steam and water is extracted from geothermal wells and is transferred through insulated pipes to the power plant. The two-phase mixture is passed directly through a number of converging-diverging nozzles placed around the periphery of a turbine wheel. The thermal energy of the mixture is converted in the nozzles to kinetic energy, which drives the turbine, a modified hydraulic impulse device. The turbine is directly coupled to a generator. The two-phase mixture at the exhaust of the turbine is passed through a centrifugal separator to separate the liquid phase. The liquid is subsequently reinjected into the reservoir. The vapor is passed through a barometric, direct-contact condenser supplied with cooling water,

FIGURE 14. Total Flow Concept

similar to those used in the flashed steam plant. Noncondensable gases are removed from the condenser with a steam jet ejector. System cooling for the turbine and generator is included, as in the other models. The cooling tower used is an induced-draft evaporative type. It can be seen that the only significant difference between the total flow plant and the flashed steam plant is the turbine. For this reason, the turbine is the only component discussed here in detail.

Total Flow Turbine

The type of turbine described in the previous section is only one of a number of types being considered by the Lawrence Livermore Laboratory.⁽⁴⁷⁾ But the use of high velocity jets to drive an impulse turbine is generally thought to be a sound approach when considering the corrosiveness and solids content of the fluid. From the standpoint of modeling, the most important information required is the nozzle inlet velocity, nozzle efficiency and overall turbine efficiency. Based on good practice, the nozzle inlet velocity should not exceed $0.09\sqrt{2gH}$, where H is the effective head in ft of H_2O .⁽⁴⁸⁾ Due to the lack of experimental data, the turbine and nozzle efficiencies were set equal to 90%.⁽⁴⁹⁾ The dependence of the quality of the two phase mixture on nozzle efficiency is not yet clear. With these assumptions and the inlet conditions to the nozzle, the exit conditions and mass flow rates can be determined. These calculations are performed in subroutine TFLOW.

Thermodynamic Calculations and Computer Logic

A flow sheet for the total flow model is shown in Figure 15. Once the inlet and exhaust conditions for the turbine are determined, the same series of subroutines are called as in the flashed steam plant to determine the operating characteristics of the power plant. Thermodynamic and physical

FIGURE 15. Computer Flow Sheet for Total Flow Plant

properties of the two-phase mixture are determined in the same manner as in the flashed steam plant model.

The total flow model has just recently been incorporated into the computer program. Results of the computer model are being compared with Lawrence Livermore data. Parameters which were varied in case runs include plant size, geothermal fluid temperature and steam quality, brine content, and individual well flow rates.

Input to this model is passed from subroutine NAMELIST through the driving subroutine DIFLOW. Sample input for a typical total flow plant is shown in Figure 16, with the corresponding output in Figure 17.

GROSS KILOWATT OUTPUT	55000.00
FLUID TEMPERATURE AT TURBINE INLET	428.58 (F)
ABSOLUTE FLUID PRESSURE AT TURBINE INLET	338.54 (psia)
FLUID ENTHALPY AT TURBINE INLET	561.13 (Btu/lb)
STEAM FRACTION OF FLUID	0.19
FLUID OVERPRESSURIZATION	1.00
TURBINE EXHAUST PRESSURE	2.00 (psia)
WEIGHT FRACTION OF NONCONDENSIBLE GASES IN STEAM	0.0
WET BULB AIR TEMPERATURE	50.00 (F)
DRY BULB AIR TEMPERATURE	70.00 (F)
BAROMETRIC PRESSURE (in. Hg)	29.92
HUMIDITY RATIO	0.20
COOLING WATER INLET TEMPERATURE	80.00
INTERCONDENSER EXIT COOLING WATER TEMPERATURE	110.00 (F)
AFTER CONDENSER EXIT COOLING WATER TEMPERATURE	120.00 (F)
MAIN CONDENSER EXIT COOLING WATER TEMPERATURE	119.08 (F)
OIL COOLER EXIT COOLING WATER TEMPERATURE	120.00 (F)
HYDROGEN COOLER EXIT COOLING WATER TEMPERATURE	120.00 (F)
PER CENT OF DISSOLVED SOLIDS IN STEAM	3.00
PLANT REINJECTION OPTION	1
PLANT REINJECTION PUMPING OPTION	0

FIGURE 16. Input to Total Flow Plant Model

PLANT INLET CONDITIONS		
TEMPERATURE	428.1	°F
PRESSURE	336.8	psia
ENTHALPY	561.2	Btu/lb
QUALITY	0.195	
VELOCITY	14.86	ft/sec
ASSUMED EFFICIENCIES		
WHEEL EFFICIENCY	0.9	
NOZZLE EFFICIENCY	0.9	
NOZZLE EXIT CONDITIONS		
TEMPERATURE	126.1	°F
PRESSURE	2.0	psia
ENTHALPY	446.3	Btu/lb
QUALITY	0.345	
VELOCITY	2275.0	ft/sec
TURBINE EXIT CONDITIONS		
GEOHERMAL FLUID RATE	2,017,550	lb/hr
TEMPERATURE	126.1	°F
PRESSURE	2.0	psia
ENTHALPY	446.3	Btu/lb
QUALITY	0.345	
VELOCITY	965.0	ft/sec
STEAM FLOW RATE	695,420	lb/hr
STEAM ENTHALPY	1,116.15	Btu/lb
NOZZLE DIAMETER	1.77	ft
TOTAL NOZZLE AREA	14.76	ft ²
TURBINE EXIT AREA	5.80	ft ²
COOLING WATER FLOW RATES		
COOLING WATER FLOW RATE	6,752,200	lb/hr
COOLING WATER + STEAM	7,447,640	lb/hr
COOLING WATER TEMPERATURE	119.1	°F
SUBROUTINE TOWER -- COOLING TOWER		
COOLING TOWER AIR FLOW RATE	1,576,340	lb/hr
COOLING TOWER EVAPORATION RATE	235,510	lb/hr
WET BULB AIR TEMPERATURE	50.0	°F
DRY BULB AIR TEMPERATURE	70.0	°F
BAROMETRIC PRESSURE	29.92	in. Hg
HUMIDITY RATIO	0.2	
HEAT LOAD	2.88 x 10 ⁸	Btu/hr
MAKEUP WATER REQUIRED	0	lb/hr
EXCESS COOLING WATER	459,910	lb/hr
REINJECTION RATE	1,782,000	lb/hr
REINJECTION PRESSURE	38.0	psia
SUBROUTINE SLOSS		
KW REQUIRED FOR FANS	228.8	kW
CONDENSATE PUMPS TO COOLING TOWER	148.4	kW
COOLING WATER PUMPS	275.8	kW
COOLING TOWER REINJECTION PUMP	688.8	kW
FLASH REINJECTION PUMPS	0	kW
VACUUM PUMP ON CONDENSER	0	kW
SCRUBBER PUMP	0	kW
WATER PRESSURIZING BOOSTER PUMPS	0	kW
DOWNHOLE PUMPS	0	kW
OVERALL POWERPLANT RESULTS		
NUMBER OF 55 MWe TURBINES	1	
GEOHERMAL FLUID RATE	2,017,550	lb/hr
TOTAL KILOWATT LOSSES	1,341.8	kW
NET KILOWATT OUTPUT	53,658.0	kW
NET FLOW RATE	37.6	lb/kW-hr
NET HEAT RATE	17,828	Btu/kW-hr
PLANT EFFICIENCY	0.192	

FIGURE 17. Output of the Total Flow PLant Model

GEOPRESSURED PLANT MODEL

The most common type of geothermal reservoir proposed for power generation is a hot-water resource at relatively low pressure. A unique type of reservoir has been discovered by oil companies while drilling deep wells for oil and gas. Regions containing water and gas have been observed that are at much higher than normal hydrostatic pressure. These regions have been classified as "geopressured" reservoirs and are located in Louisiana and the Gulf of Mexico. (50,51,52)

The power plant system proposed for the geopressured reservoirs is designed to take advantage of the two unique aspects of this resource--the presence of high pressures and natural gas. The basic power plant used can be one of two types already described--the binary cycle plant or the total flow plant.

Description of Power Plant

The geopressured model operates very much in the same manner as the other models except for the plant selection option. Also, the natural gas is extracted from the geothermal fluid after expansion through the turbine. If the total flow power plant is being used, the natural gas is simply sold. If the binary plant is used, the natural gas can either be sold or used to provide additional heat to the geothermal fluid, thus reducing the required flow rate.

The geopressured plant model is designed to allow sensitivity analyses on various parameters, including plant size, fluid temperature and pressure, and natural gas content. In addition, both the binary cycle plant and total flow concept can be used in conjunction with this reservoir type.

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D. J. Braun
L. L. Clark
M. Clement
M. O. Cloninger
P. D. Cohn
B. W. Cone
J. W. Currie
D. E. Deonigi
J. G. DeSteeze
R. L. Dillon
P. J. Dionne
L. E. Erickson
T. J. Foley
J. C. Fox
J. L. Harris
H. D. Huber
W. S. Kelly

P. N. LaMori
K. W. McGinnis
R. W. McKee
W. McSpadden
T. B. Powers
S. A. Rao
R. G. Rau
W. A. Reardon
D. W. Shannon
J. R. Sheff
D. L. Stewart
L. T. Pederson
R. A. Walter (12)
S. L. Wise
K. E. Yandon
J. R. Young
Economics Library (3)
Technical Information (3)
Technical Publications (1)