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Modeling and Optimization of Geothermal Power Plants Using the Binary Fluid Cycle

by
R. A. Walter

September 1976

**Prepared for the Energy Research
and Development Administration
under Contract E(45-1)-1830**

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GEOTHERMAL POWER PLANTS USING
THE BINARY FLUID CYCLE

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MODELING AND OPTIMIZATION OF GEOTHERMAL
POWER PLANTS USING THE BINARY FLUID CYCLE

I. INTRODUCTION

Geothermal reservoirs are a potentially large energy source currently under development in the United States and other countries. Efforts to utilize vast reserves of existing geothermal energy have resulted in the design of numerous power production facilities. The research described in this report was sponsored by the Energy Research and Development Administration whose objective is to develop new energy sources that meet the increasing demands for electrical power in the United States. This research was also developed in part to fulfill the requirements for an M.S. thesis.

One of the most promising of these power production techniques is the binary fluid cycle power plant. In order to realize a rapid development in this area, a complete understanding of the design of binary fluid cycles for use in generating electricity from geothermal energy sources is necessary. In addition, techniques for determining the type and conditions for a binary fluid cycle which result in the optimum utilization of a geothermal energy source require development, in both an economic and a thermodynamic sense. This report describes a computer simulation of a binary fluid cycle power plant for use with geothermal

energy sources, and the subsequent optimization of this power plant type over a range of geothermal source conditions. The optimization technique employed for this analysis was based upon the principle of maximum use of available geothermal energy.

The use of the binary cycle for power generation is practical in liquid-dominated geothermal reservoirs or in those geothermal zones where a high concentration of non-condensable gases are present. In the binary fluid cycle power plant, geothermal fluid is supplied through insulated pipelines which has been extracted from wells distributed throughout the reservoir. The geothermal fluid is constantly maintained at a pressure exceeding the saturation pressure with deep-well or surface pumps. The maintenance of the fluid in a sub-cooled state prevents the loss of latent heat through vaporization, reduces the pressure drop of the geothermal fluid during transmission to the power plant, and minimizes the precipitation of dissolved solids out of the geothermal fluid. 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 sub-critical or super-critical binary 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 the

working fluid passing through the vapor dome, i.e., at a pressure below the critical pressure. The geothermal fluid is reinjected into the existing reservoir in a liquid state after passing through the heat exchangers. The working fluid is brought to a superheated state and then expanded through a turbine which is coupled to a generator. 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.

Cooling water is supplied to a surface condenser, a turbine oil cooler, and a generator cooler from an induced-draft evaporative cooling tower. Make-up water to replace cooling water lost due to blow-down, evaporation, and drift is supplied 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.

The binary fluid cycle power plant described in the preceding paragraphs is simulated in a computer model called GEOCOST.⁽¹⁾ The description of the power plant model is explained in this report, including both a physical and thermodynamic design of the power plant. Major power plant components are determined, and the plant operating conditions are identified. A computer user manual for the

binary cycle version of GEOCOST was published in March 1976.⁽⁸⁵⁾ The source code was been made available on magnetic tape through the Argonne Code Center.

Optimal utilization of the energy from geothermal reservoirs is necessary because of the relatively low energy-content of the geothermal fluid and the subsequent reduced efficiencies of power conversion schemes compared with those of fossil-fueled or nuclear power plants. With the incentive of efficient energy utilization and the availability of a binary fluid cycle power plant model, a technique of optimization has been developed. This optimization technique is used to develop a binary cycle design which maximizes geothermal energy utilization based upon the limits of geothermal resource temperature and the prevailing ambient temperature for heat rejection. Illustrations of the use of this technique are reported for a range of geothermal resource temperatures.

II. LITERATURE REVIEW

II-1. The Geothermal Resource

The earth contains vast quantities of thermal energy. Unfortunately, this energy is normally well dispersed and at depths which make it impractical for extraction. In many areas of the earth, however, there exist abnormally concentrated heat flows near the surface. These areas are called hyperthermal zones, and are often associated with recent volcanic activity and surface manifestations such as hot springs, geysers, and fumaroles. Numerous potential sites for geothermal exploitation have been identified in recent years.

There exist five basic types of geothermal fields, or reservoirs, that are exploitable at the present time.

These types are as follows

- (1) water-deficient hot rocks
- (2) liquid dominated reservoirs with low salinity
- (3) liquid dominated reservoirs with high salinity
- (4) reservoirs that produce a mixture of water and steam
- (5) dry steam reservoirs.

The classification of these geothermal reservoir types is based primarily upon the difference in methods of exploitation. Since this report is based upon energy conversion methods, these classifications were made instead of differentiating by reservoir temperature, rock type, hydrologic systems, or geological formations.

The first type of reservoir listed is hot dry rocks. These regions may well be more numerous than any other type, but exploration for these reservoirs has been limited due to difficulties in exploitation. The lack of sufficient fluid to transfer heat from the reservoir to the surface makes power production difficult. Three regions where hot dry rocks may exist have been identified in the United States. The Valles Caldera is a typical volcanic caldera in New Mexico which has been under study by the U.S. Geological Survey and the Los Alamos Scientific Laboratory.⁽²⁾ A second potential hot dry rock site has been identified in Montana at Marysville by Blackwell.⁽³⁾ Subsequent investigation by the National Science Foundation and Battelle Northwest Laboratories indicated the area is actually a hydrothermal anomaly in Genozoic intrusive.⁽⁴⁾ A third hot dry rock area has been identified in California at Coso Hot Springs by the U.S. Geological Survey and the U.S. Naval Weapons Center.⁽⁵⁾ Exploration in other countries is currently being promoted through the NATO committee on Challenges of Modern Society.⁽⁶⁾

The second geothermal reservoir type identified is the liquid dominated reservoir with low salinity. The presence of low salt concentrations make this resource type desirable because of fewer existing energy conversion problems. Numerous reservoirs of this type exist around the world, and have been utilized for electrical

production, space heating, agriculture, health spas, minerals recovery, and industrial processing. The well known reservoirs of this type include Klamath Falls, Oregon;⁽⁷⁾ Raft River, Idaho;⁽⁸⁾ and Beowawe, Nevada⁽⁹⁾ in the United States, Paratunka⁽¹⁰⁾ and Pauzhetka⁽¹¹⁾ in the Soviet Union, Reykjavik, Iceland,⁽¹²⁾ and Kizildere, Turkey.⁽¹³⁾ This resource is the most applicable of the five types listed for utilization by the binary cycle.

Unlike the second resource type, liquid dominated reservoirs with high salinity are less common. Known sites exist in southern North America and Central America. Due to the presence of high salt concentrations and associated problems of corrosion, scaling, and waste disposal, these areas for the most part resist development. One area has overcome these difficulties and now contains an operating power plant; Cerro Prieto in Mexico.⁽¹⁴⁾ Two other areas, Ahuachapan, El Salvador⁽¹⁵⁾ and the Imperial Valley, California⁽¹⁶⁾ have received more attention but are not as yet being utilized for electrical power generation.

The known geothermal reservoirs producing a two phase mixture of steam and water are located in New Zealand at Wairakei, Kawerau, and Rotorura⁽¹⁷⁾ and in Otake, Japan.⁽¹⁸⁾ The characteristics of a field producing two phase flow are similar to those fields which contain pure steam. The reservoir temperatures are normally high

(above 100°C) and the wells yield more water than steam. The ratio of liquid to vapor varies widely from reservoir to reservoir and from well to well in a single reservoir.

The final reservoir type is the one producing dry steam, and is the most heavily utilized but least prevalent. Dry steam reservoirs are easily adaptable to power production, and normally the steam can be directly expanded through a low pressure turbine. Electrical power has been generated at the Lardarello field in Italy since 1913,⁽¹⁹⁾ and at the Geysers field in California since 1960.⁽²⁰⁾ Italy has developed two additional fields at Monte Amiata⁽²¹⁾ and Travale.⁽²²⁾ Japan has operated a power plant on a dry steam reservoir at Matsukawa since 1966.⁽²³⁾ Dry steam reservoirs have also been discovered at Namafjall, Iceland⁽²⁴⁾ and Mount Eburru, Kenya.⁽²⁵⁾

Geothermal development is underway in many countries around the world. By 1980, the estimated generating capacity to all sites will be 2500 MWe. Even by present standards, this is still less than 1% of the total world capacity. As stated in the previous paragraphs, there exist numerous geothermal reservoirs of different types around the world, but many of these sites have not been developed due to the difficulties of power generation from a low energy heat source which often contains corrosive contaminants. The binary fluid cycle may be acceptable for use at many of these undeveloped sites, and may be

the key to increasing the generating capacity of geothermal resources at a faster pace. Before discussing the binary fluid cycle, a summary of existing geothermal power plants is appropriate.

II-2. Power Production from Geothermal Reservoirs

Power production from geothermal reservoirs has been a reality since 1913 when a 250-KW generator was installed at Lardarello, Italy. The majority of the world's generating capacity has been installed in the past sixteen years. All of the operating power plants except one use steam for direct expansion through a standard, low-pressure, reaction turbine similar to those in use at fossil-fueled steam power plants. The geothermal steam for use at these plants is supplied in one of three ways, (a) through dry steam wells, (b) through separation of a two phase mixture of steam and water, or (c) through flashing of geothermal fluid at a lower pressure with subsequent steam separation. Through one of these techniques, steam has been successfully supplied to power plants in Italy, New Zealand, United States, Japan, Mexico, U.S.S.R., and Iceland. A summary of these power plants is shown in Table 1. By 1980, electrical power production is expected to be underway at Ahuachpan, El Savador; Hengill, Iceland; Hachimantai, Japan; Onikobe, Japan; Kunashin, U.S.S.R., and Imperial Valley, United States.⁽²⁶⁾ The power plant

TABLE 1. Operating Geothermal Power Plants

<u>Country</u>	<u>Location</u>	<u>Generating Capacity (MWe)</u>	<u>Reference</u>
Italy	Lardarello	365	19
	Monte Amiata	25	21
	Travale	15	22
New Zealand	Wairakei	192	17
	Kawerau	10	17
United States	Geysers	396	20
Japan	Matsukawa	20	23
	Onuma	10	22
	Otake	13	18
Mexico	Cerro Prieto	15	14
U.S.S.R.	Paratunka	1	10
	Pauzhetka	5	11
Iceland	Namafjall	<u>3.4</u>	24

TOTAL 1130.4 MWe

at Paratunka, U.S.S.R. is the only plant listed in Table 1 that is not based on a steam expansion cycle. The Paratunka plant is of a binary fluid cycle design with Freon-12 as the working fluid.

The installation of geothermal power plants has preceded at a slow pace even though recent design refinements in turbines, condensers, heat rejection systems, and well drilling techniques have increased available plant efficiencies and the ability to develop geothermal reservoirs economically. However, two major difficulties

still exist. Corrosion and scaling in plant components are caused by the transmission and collection of geothermal fluids containing large amounts of impurities. In addition, economic methods of utilizing low-enthalpy geothermal fluids in the liquid state have not been developed sufficiently for wide scale installation. One of the most promising power production techniques is the binary fluid cycle design. The binary fluid cycle power plant reduces the contact of the geothermal fluid with major plant components, and if designed properly, can economically produce electrical power. The use of the binary fluid cycle with geothermal resources is discussed in the following section.

II-3. The Binary Fluid Cycle Geothermal Power Plant

For relatively low-temperature geothermal reservoirs, especially those with high non-condensable gas content or high solids concentration, the binary fluid cycle has been proposed. The binary fluid cycle is not a new concept. In 1885, Frank Ofeldt patented and produced a binary fluid cycle process using light naptha. This power cycle design was considered for use in automobiles, but by 1905 was replaced by the internal combustion engine. Since that time, the binary fluid cycle has been proposed for use in numerous applications such as commercial power plants,⁽²⁷⁾ pressurization stations for natural gas pipelines,⁽²⁸⁾

refrigeration plants,⁽²⁹⁾ power production at industrial plants from hot exhaust gases,⁽³⁰⁾ road vehicles,⁽³¹⁾ and geothermal power plants.⁽³²⁾ For the most part, the use of these binary fluid cycles has not been applied on a wide scale; most efforts have resulted in only small experimental units or merely published technical papers.

The binary fluid cycle can be applied in general to four of the five reservoir types discussed in Section II-1. Dry steam reservoirs are the exception. The reservoirs producing dry steam are most easily and economically utilized by direct expansion through a turbine for power generation. It is unlikely that the addition of a second working fluid loop with heat exchangers can compete economically. The use of the binary fluid cycle with dry steam reservoirs should be discouraged unless the non-condensable gas content or solids concentration is extremely high. Liquid dominated reservoirs with low salinity are the most applicable for use with binary fluid cycles of all the reservoir types. If the precipitation problems associated with highly saline brines can be minimized, liquid dominated reservoirs with high salt content will also be conducive to binary fluid cycle plant operation. Two-phase reservoirs can become single-phase reservoirs if deep well pumps are used at sufficient depths to prevent flashing of the geothermal fluid in the well. If

these pumps are developed with large enough capacities, a two-phase reservoir will have much the same potential for binary plant operation as a liquid dominated reservoir. The hot dry rock reservoirs are somewhat in the future as far as utilization is concerned, but here also the binary fluid cycle may apply if water can be recirculated through the reservoir formation, picking up heat in the formation and exchanging heat with a binary working fluid. In summary, the binary fluid cycle is potentially applicable to the majority of geothermal resource types. To date, the binary fluid cycle has been proposed for use with only liquid dominated geothermal reservoirs with low salinity. The most significant application of the binary fluid cycle is at the Paratunka geothermal electric plant in the U.S.S.R. In September, 1967 the Paratunka plant was completed with a 340-KW Freon turbine designed at The Thermal Physics Institute of the Siberian Branch of the U.S.S.R. Academy of Sciences. This plant is successfully operating using geothermal water at a temperature of 178°F. Freon-12 is heated to a superheated state at 150°F and 202 psia where it is then expanded through a Shatura HEPP-5 turbine. Geothermal water flow is 637,000 lb/hr while the cooling water flow is 3,350,000 lb/hr. The Paratunka power plant has recently been expanded to 1000 KWe.

Although no geothermal power plants using the binary fluid cycle have been constructed in the United States, numerous designs have been proposed. In 1973, the Magma Power Company received a patent on the Magmamax Process which is an improved binary fluid cycle design using isobutane in a supercritical cycle.⁽³³⁾ A later design called the Hutchinson-Holt Process has been developed using isobutane, but the details are not known for proprietary reasons.⁽³⁴⁾ Typical operating conditions, however, are shown for a binary fluid cycle operating with 400°F geothermal brine and isobutane in a supercritical state. Preliminary power plant designs have been published by TRW, Inc.,⁽³⁵⁾ Lawrence Berkeley Laboratories,⁽³⁶⁾ the University of Hawaii,⁽³⁷⁾ Aerojet Nuclear Company,⁽³⁸⁾ San Diego Gas and Electric,⁽³⁹⁾ and Roger's Engineering.⁽⁴⁰⁾ Some important features of these designs are shown in Table 2. All preliminary power plant designs have been made using isobutane in a 10 MWe size. There are five sub-critical cycle designs and one super-critical design. All designs except that of San Diego Gas and Electric use the geothermal fluid in a compressed liquid state.

In addition to the publication of binary cycle power plant designs, there has been limited work done on comparing the binary cycle with other geothermal plant designs and with variations on the binary cycle design

TABLE 2. Proposed Binary Cycle Geothermal Plants

<u>Plant</u>	<u>Size (MWe)</u>	<u>Working Fluid</u>	<u>Geothermal Fluid Temp (°F)</u>	<u>Max Working Fluid Temp (°F)</u>	<u>Max Working Fluid Pressure(psia)</u>	<u>Comments</u>
TRW, Inc.	10.0	Isobutane	350°F	310°F	450	subcritical cycle with regeneration; heat rejection with air cooled condenser.
Lawrence Berkeley Labs	10.0	Isobutane	392°F	370°F	600	simple supercritical Rankine cycle.
U. of Hawaii	10.0	Isobutane	350°F	320°F	400	subcritical cycle with regeneration.
Aerojet Nuclear	10.0	Isobutane	300°F	250°F 190°F	350 160	dual boiling system with high pressure and low pressure turbine, series geothermal flow through high, then low temperature side, regeneration used.
SDG&E	10.0	Isobutane	370°F	300°F	500	subcritical cycle; geothermal fluid flashed in four successive stages; steam/isobutane heat exchangers are used; for the present, the turbine is simulated by an expansion valve.
Rogers Engr.	10.0	Isobutane	325°F	290°F	500	simple subcritical Rankine cycle.

itself. (41)(42)(43) These studies, and the power plant designs, are in general reporting a single design at a particular geothermal source temperature or investigating a particular component of a power plant. A comprehensive and systematic method of selecting the best cycle for a particular geothermal source condition over a wide range of those conditions has not been reported in the literature. This report is meant to address this need by providing information on a computer code that has been developed to simulate the geothermal binary cycle power plant, and an optimization technique to be used for cycle selection.

III. DESCRIPTION OF THE COMPUTER MODEL

III-1. Discussion

Research on the binary fluid cycle for geothermal power plants has led to the construction of a computer model which simulates the major components and the overall design of the power plant. This binary cycle plant model is part of an overall geothermal system model called GEOCOST, which has been developed at Battelle Northwest Laboratories.⁽⁴⁴⁾ GEOCOST is basically a simulation model which calculates the cost of generating electricity from geothermal energy, and is composed of two principal sections: a reservoir model which simulates the exploration, development, and operation of geothermal reservoir, and a power plant model which simulates the design, construction, and operation of the power plant. In this report, the power plant model of the binary fluid cycle is described in detail. A general description of four different types of power plants simulated in GEOCOST; (a) flashed steam plant, (b) total flow plant, (c) binary fluid cycle plant, and (d) a geopressured reservoir plant is provided in an additional publication.⁽⁴⁵⁾

Computer programs have been used to simulate binary fluid cycle power plants for geothermal applications by other researchers. M. A. Green and H. S. Pines⁽⁴⁶⁾ are currently developing a geothermal power plant model based upon the original work of S. L. Milora.⁽⁴⁷⁾ This work

has been focused mainly on specific potential geothermal sites under consideration by the Lawrence Berkeley Laboratories, University of California. J. P. Abbin and W. R. Leuenberger have published a report describing a computer program called CYCLE which models various Rankine cycle systems.⁽⁴⁸⁾ This work resulted from research efforts at Sandia Laboratories on solar energy. A number of different working fluids are available for analysis of binary fluid cycles but unfortunately the working fluid properties are calculated by the Hirshfelder-Buehler-McGee-Sutton (HBMS) equation of state.⁽⁴⁹⁾ This equation does not accurately predict thermodynamic properties for working fluids at high reduced pressures (ratio of actual pressure to critical pressure near or greater than one). A third publication by T. Juul-Dann and H. F. Dunlop simulates the binary fluid cycle in general terms but mainly emphasizes the economic evaluation of a geothermal exploration and production venture.⁽⁵⁰⁾ One final report still in publication is by S. L. Milora and J. W. Tester. This publication describes a binary fluid cycle model which selects optimum working fluids for different geothermal resource conditions.⁽⁵¹⁾ Important factors in geothermal energy development are also discussed in this publication.

The present work in developing a binary fluid cycle plant model is in part based upon S. L. Milora's original

work, but is mainly constructed on the basis of providing an accurate and flexible simulation of this type of geothermal power plant for cost prediction purposes. The basic criteria for this power plant model are as follows:

- capable of selecting binary cycle plant designs for a wide range of geothermal resource conditions
- ability to be operated by a user with a minimum of training in thermodynamics
- availability of numerous working fluids
- capable of simulating both sub-critical and super-critical cycles
- ability to accurately compute thermodynamic and physical properties of selected working fluids
- simulation of all major power plant components and their basic size
- applicable to a wide range of power plant sizes
- capable of performing thermodynamic cycle calculations
- ability to compute geothermal flow rates, working fluid flow rates, and cooling water flow rates
- capable of predicting overall design performance, with the ability to perform sensitivity analyses on selected operating parameters
- capable of predicting an optimum plant design based on maximum geothermal fluid utilization.

The current binary fluid cycle plant model meets these design criteria, and can be operated independently or as a part of the overall program, GEOCOST. The basic structure of the model and major plant components are discussed in the following paragraphs.

III-2. Structure of the Computer Model and Plant Description

A typical binary fluid cycle power plant is displayed in Figure 1. This plant has been described briefly in the Introduction. The computer program is started at the point where the geothermal fluid enters the heat exchangers and ends where the geothermal fluid leaves the heat exchangers. All elements in the closed working fluid cycle and the cooling water loop are included in the program. On the basis of the power plant size, the thermodynamic condition of the geothermal fluid as it enters the heat exchangers, the selected working fluid type, and the meteorological conditions, the computer model selects key operating points of the working fluid cycle. Once these selections have been made, the properties of all fluids present (geothermal fluid, working fluid, and cooling water) are computed at the inlet and outlet of each major plant component. Knowing these state conditions, the flow rates of all fluids and the size of major components are determined. Thus, the computer model can be divided into five major sections,

FIGURE 1. Sub-Critical Binary Fluid Cycle Power Plant

(a) model input, (b) cycle selection, (c) property calculations, (d) component size and flow rate calculations, and (e) model output. To provide a better understanding of the simulated binary fluid cycle power plant, the major components are briefly described below.

Geothermal Brine/Working Fluid Heat Exchangers -

Heat exchangers are required in a binary fluid cycle power plant to transfer heat from the geothermal brine to the working fluid, as indicated in Figure 1. The power plant shown in the figure is composed of a sub-critical cycle, in which the working fluid is passed through the heat exchangers at a pressure below the critical pressure. A supercritical binary plant is similar to the one shown in the figure, except for the maximum working fluid pressure being greater than the critical pressure.

The basic type of heat exchanger simulated for all binary fluid cycle plants was a counter-flow, single-pass, shell and tube unit with a fixed tube sheet. Standard material of construction was assumed to be carbon steel. In the sub-critical cycle, heat exchangers are required for three different applications -- preheating, vaporization, and superheating. In the superheater, the hottest geothermal fluid exchanges heat with the working fluid to bring the working fluid from a saturated vapor state to a superheated state. The temperature difference between the entering geothermal fluid and the leaving working fluid is

defined as the "approach to the heat exchanger." The vaporizer receives the geothermal fluid from the outlet of the superheater and provides the heat of vaporization for the working fluid, bringing it 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. The temperature difference between the entering geothermal fluid and the leaving working fluid is defined as the pinch point of the heat exchanger. This entire procedure is depicted in Figure 2, with typical water and working fluid temperatures attached.

The heat exchangers for the supercritical cycle are similar in geometry to those in the sub-critical cycle, except for the fact that no boiling section exists. The working fluid is always maintained above the critical pressure while in the heat exchangers; therefore, the working fluid does not pass through the vapor dome.

Turbine - Generator - The turbine for the binary fluid cycle plant is simulated as an isentropic expansion device with an associated internal efficiency. By combining this internal efficiency with the mechanical efficiency and generator efficiency, the overall efficiency of the turbo-generator set can be determined. Based upon the inlet, exhaust, and saturated liquid enthalpies of the working fluid, the isentropic and actual Rankine cycle

FIGURE 2. Typical Heat Exchanger Configuration for Binary Cycle Plant

efficiencies can be determined, according to the following relations.

$$\eta_{\text{Rankine, isentropic}} = \frac{H1-H2_{\text{EST}}}{H1-H2_{\text{SATL}}} \quad (1)$$

$$\eta_{\text{Rankine, actual}} = \frac{H1-H2}{H1-H2_{\text{SATL}}} \quad (2)$$

Only one turbine for use with working fluids in a geothermal application has been manufactured in the United States. This turbine was designed for use with isobutane and is a 9000 kWe three-stage radial inflow turbine. It was manufactured by the York Division of Borg-Warner Corporation for Magma Energy, Inc. (52)

Condenser - The working fluid condenser used in 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. The condenser is actually two separate units. One unit is the desuperheater which uses water as a cooling medium to bring the working fluid from a superheated state at the turbine exhaust to a saturated vapor state. The second unit is a condensing heat exchanger which also uses cooling water to bring the working fluid from a saturated vapor to a saturated liquid state.

Pumps - The type of pump simulated in this model for transmitting the geothermal fluid from the reservoir to the power plant is a turbine pump. This is a pump driven

from the surface with turbine bowls submersed down in the geothermal well. The pump should be carefully placed deep enough in the well to prevent flashing of the geothermal fluid. An electric motor is used to drive the turbine pump.

Pumps for the cooling water and working fluid loops are standard centrifugal pumps, and are assumed to be driven by electric motors. Power consumption for all pumps is determined according to the following equation where $(P_a - P_b)$ is the pressure rise across the pump.

$$kW_{\text{pump}} = \frac{\dot{m}(P_a - P_b)(144)}{\rho(1.98 \times 10^6)(.745)\eta_{\text{pump}}} \quad (3)$$

System Cooling - In addition to the requirements for cooling water to the working fluid condenser, some water is required for the turbine oil cooler and the generator cooler. These units are based on standard manufacturer's designs and require a small percentage of the total cooling water flow.

Cooling Tower - The cooling tower simulated in this model is an induced draft evaporative heat rejection device. Air is induced with large propeller-type fans to pass upward through fill material which contains heated water. Latent heat is passed to the air stream by partial evaporation of the water stream. The design of this cooling tower is dependent upon the relative humidity, the wet bulb air temperature and the temperature

difference between the entering water temperature and the wet bulb temperature.

With the basic structure and components of the model described, the details of the model are described in the following paragraphs.

III-3. Computer Analysis of the Binary Fluid Cycle for a Geothermal Power Plant

The computer model which simulates the binary fluid cycle plant consists of numerous sections. The working fluids, their selection, and prediction of their thermodynamic properties will be discussed initially. The particular thermodynamic cycles chosen for use in this computer program will be described after working fluids. An explanation of the program control logic will follow. The main body of the model description is in the performance of thermodynamic cycle calculations using the logic format established in the computer program. Final topics discussed are the determination of physical properties for existing materials and the determination of the operating characteristics of the power plant itself.

Working Fluids - Numerous different working fluids have been suggested for use in binary fluid cycles. The halogenated hydrocarbons (Freons) have received a great deal of attention due to their low boiling points and high molecular weights. The following Freons have been suggested for use in geothermal applications: R-11

(or Freon-11), R-12, R-13, R-21, R-22, R-32, R-31/114 (azeotrope of Chlorofluoromethane and Dichlorotetrafluoroethane), R-33a, R-112, R-113, R-114, R-115, and R-116. (53, 54, 55, 56, 57) Several light hydrocarbons have also been promoted, including isobutane, n-butane, propane, methane, propylene, and isopentane. (54, 56, 57) Other miscellaneous selections have included ammonia, carbon dioxide, methyl chloride, mercury, and aluminum bromide. (53, 54, 57, 58) The large number of possible working fluids and their variations in properties makes the selection of a particular working fluid quite difficult. Ultimately, the following criteria must be considered;

- (1) low cost
- (2) high molecular weight
- (3) reasonable critical properties
- (4) stability at high temperatures
- (5) non-corrosive nature
- (6) high thermal efficiency
- (7) good heat transfer properties
- (8) low toxicity
- (9) low flammability
- (10) availability of technical data.

In order to provide a range of critical temperatures and pressures as well as molecular weights, nine working fluids were selected for use in the binary fluid cycle model. One additional consideration for selection was the

availability of thermodynamic property data. The working fluids are listed in Table 3, along with their critical properties.

This list of working fluids does not imply that these are the best candidates. Most likely, different applications and different resource conditions will cause the power plant designer to select different fluids. Due to the lack of geothermal power plant experience and analytical analysis, and the general area of acceptance, almost all proposed binary cycle plants for geothermal applications have used isobutane, as is seen in Table 2. For this reason, the sample output and optimization analysis in this report will use isobutane as the working fluid.

In order to accurately predict the thermodynamic properties of these various working fluids, complex equations of state were used rather than tabulated data. This was done to minimize required computer memory and to insure consistency and accuracy of the calculated properties. The Martin-Hou equation of state was selected for use with the halogenated hydrocarbons (Freons) and ammonia. (59, 60, 61) The Starling equation of state was chosen for use with the light hydrocarbons, isobutane and n-butane. (62)

The Martin or Martin-Hou equation of state was first introduced in 1967, and was the result of an extensive

TABLE 3. Selected Working Fluids

<u>Compound</u>	<u>Critical Temperature (°F)</u>	<u>Critical Pressure (psia)</u>	<u>Critical Density (lb/ft³)</u>
Isobutane	274.96	529.06	13.80
n-butane	305.62	550.66	14.24
R-11; trichlorofluoromethane	388.4	640.0	34.6
R-12; dichlorodifluoromethane	233.6	597.0	34.84
R-21; dichlorofluoromethane	353.3	750.0	36.0
R-22; chlorodifluoromethane	204.8	721.9	32.8
R-113; trichlorotrifluoroethane	417.4	498.9	36.0
R-114; dichlorotetrafluoroethane	294.3	473.0	36.3
Ammonia	271.4	1657.0	14.6

study of the properties of real gases. The present form of the equation, with 21 virial coefficients, predicts properties quite accurately and has a superior fit to experimental data in the super-critical region. Fortunately, these 21 virial coefficients for the six Freons and ammonia were made available through the efforts of R. C. Downing and S. L. Milora. The Martin-Hou equation of state involves the general form which can be solved explicitly for pressure, as

$$P = \sum_{i=1}^5 f_i (V-b)^{-i} \quad (4)$$

where f_i is a temperature dependent function

$$f_i = A_i + B_i T + C_i \exp(-KT/T_c). \quad (5)$$

These equations reflect the original form of the Martin-Hou equation; the present equation can be expanded as follows;

$$\begin{aligned} P = & \frac{RT}{(V-b)} + \frac{A_2 + B_2 T + C_2 \exp(-KT/T_c)}{(V-b)^2} \quad (6) \\ & + \frac{A_3 + B_3 T + C_3 \exp(-KT/T_c)}{(V-b)^3} + \frac{A_4 + B_4 T + C_4 \exp(-KT/T_c)}{(V-b)^4} \\ & + \frac{A_5 + B_5 T + C_5 \exp(-KT/T_c)}{(V-b)^5} + \frac{A_6 + B_6 T + C_6 \exp(-KT/T_c)}{\exp(\alpha V)(1 + C' \exp(\alpha V))} \end{aligned}$$

The letters A_i , B_i , C_i , B , α , etc. represent the virial coefficients. These coefficients are listed in Appendix I for each Freon. A reduced form of this equation of state

can be used to determine the saturation pressure explicitly as a function of temperature.

$$\log_{10} P(\text{saturation}) = A + \frac{B}{T} + C \log_{10}(T) + DT + E\left(\frac{F-T}{T}\right) \log_{10}(F-T) \quad (7)$$

The equations for enthalpy and entropy are also derived from the original equation of state

$$\begin{aligned} H = & aT + \frac{bT^2}{2} + \frac{cT^3}{3} + \frac{dT^4}{4} - \frac{f}{T} + JPV + J \left[\frac{A_2}{(V-b)} + \frac{A_3}{2(V-b)^2} \right. \\ & + \frac{A_4}{3(V-b)^3} + \frac{A_5}{4(V-b)^4} + \frac{A_6}{\alpha} \left(\frac{1}{\exp(\alpha V)} - C \ln \left(1 + \frac{1}{C \exp(\alpha V)} \right) \right) \Big] \\ & + J \exp(-KT/T_c) (1 + KT/T_c) \left[\frac{C_2}{(V-b)} + \frac{C_3}{2(V-b)^2} + \frac{C_4}{3(V-b)^3} \right. \\ & + \frac{C_5}{4(V-b)^4} + \frac{C_6}{\alpha \exp(\alpha V)} - \frac{C_6 C}{\alpha} \ln \left(1 + \frac{1}{C \exp(\alpha V)} \right) \Big] + X \end{aligned} \quad (8)$$

where $J = .1851$

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

$$\begin{aligned}
S = & a \ln T + bT + \frac{cT^2}{2} + \frac{dT^3}{3} - \frac{f}{2T^2} + JR \ln(V-b) \\
& - J \left[\frac{B_2}{(V-b)} + \frac{B_3}{2(V-b)^2} + \frac{B_4}{3(V-b)^3} + \frac{B_5}{4(V-b)^4} \right. \\
& \left. + \frac{B_6}{\alpha} \left(\frac{1}{\exp(\alpha V)} - C' \ln \left(1 + \frac{1}{C' \exp(\alpha V)} \right) \right) \right] \\
& + \frac{JK}{T_c} \exp(-KT/T_c) \left[\frac{C_2}{(V-b)} + \frac{C_3}{2(V-b)^2} + \frac{C_4}{3(V-b)^3} + \frac{C_5}{4(V-b)^4} \right. \\
& \left. + \frac{C_6}{\alpha \exp(\alpha V)} - \frac{C_6 C'}{\alpha} \ln \left(1 + \frac{1}{C' \exp(\alpha V)} \right) \right] + Y
\end{aligned} \tag{9}$$

where Y = reference entropy (based on -40°F for the saturated entropy)

Expressions for liquid density, heat of vaporization, and heat capacity are also available but not listed here. All working fluids using the Martin-Hou equation of state are based on a datum of the saturated liquid at -40°F , where both the enthalpy and entropy are defined to be equal to zero. In conclusion, this equation of state is superior to any generalized equations of state for the halogenated hydrocarbons, but is obviously applicable to computer use only.

The Starling equation of state has recently been developed for light hydrocarbons, and is an improvement upon the original work of M. Benedict, G. B. Webb, and L. C. Rubin.⁽⁶³⁾ The Starling equation is superior for predicting thermodynamic properties at reduced temperatures as low as $T_r = 0.3$ and reduced densities as high as $\rho_r = 3.0$. The tabulation of data by Starling

complements the earlier work by Canjar and Manning on the compilation of gas phase properties of light hydrocarbons. (64)

The Starling equation of state can be expressed as pressure equal to a function of temperature and molar density as follows:

$$P = \rho RT + \left(B_0 RT - A_0 - \frac{C_0}{T^2} + \frac{D_0}{T^3} - \frac{E_0}{T^4} \right) \rho^2 + \left(bRT - a - \frac{d}{T} \right) \rho^3 + \alpha \left(a + \frac{d}{T} \right) \rho^6 + \frac{C_0 \rho^3}{T^2} (1 + \gamma \rho^2) \exp(-\gamma \rho^2) \quad (10)$$

The letters B_0 , A_0 , C_0 , D_0 , E_0 , a , b , c , d , α , and γ represent constant coefficients for a particular hydrocarbon. These values are listed in Appendix I for isobutane and n-butane. The expressions for enthalpy and entropy are based upon this equation of state.

$$H = \left(B_0 RT - 2A_0 - \frac{4C_0}{T^2} + \frac{5D_0}{T^3} - \frac{6E_0}{T^4} \right) \frac{\rho}{\phi} + \frac{1}{2} \left(2bRT - 3A - \frac{4d}{T} \right) \frac{\rho^2}{\phi} + \frac{1}{5} \alpha \left(6a + \frac{7d}{T} \right) \frac{\rho^5}{\phi} + \frac{C}{\gamma \phi T^2} \left[3 - \left(3 + \frac{1}{2} \gamma \rho^2 - \gamma^2 \rho^4 \right) \exp(-\gamma \rho^2) \right] + H_0 \quad (11)$$

where ϕ is the conversion from Btu/lb_m to psia ft³/lb mole and H_0 is the ideal gas enthalpy at the temperature in question;

$$H_0 = A_{00} + A_{10}T + A_{20}T^2 + A_{30}T^3 + A_{40}T^4 \quad (12)$$

$$S = - \frac{R}{\phi} \ln(\rho RT) - \left(B_0 R + \frac{2C_0}{T^3} - \frac{3D_0}{T^4} + \frac{4E_0}{T^5} \right) \frac{\rho}{\phi} - \frac{1}{2} \left(bR + \frac{d}{T^2} \right) \frac{\rho^2}{\phi} \\ + \frac{\alpha d \rho^5}{5T^2 \phi} + \frac{2C}{\gamma T^3 \phi} \left[1 - \left(1 + \frac{1}{2} \gamma \rho^2 \right) \exp(-\gamma \rho^2) \right] + S_0 \quad (13)$$

where S_0 is the ideal gas entropy at the temperature in question.

$$S_0 = B_{00} + B_{10}T + B_{20}T^2 + B_{30}T^3 + B_{40}T^4 \quad (14)$$

The calculation of the saturation pressure as a function of temperature is more complex when using the Starling equation than the Martin-Hou equation. An iteration is required whereby an initial guess is made for $P_{\text{saturation}}$, and then the density roots are found from equation (10). Caution must be exercised since there is a possibility of three or more real density roots to this equation. The largest root is the saturated liquid density and the smallest root is the saturated vapor density. Once these two density roots are found at the $P_{\text{saturation}}$ estimate, they are used to calculate the liquid and vapor fugacities using the following equation,

$$\ln f = \ln(\rho RT) + 2 \left(B_0 RT - A_0 - \frac{C_0}{T^2} + \frac{D_0}{T^3} + \frac{E_0}{T^4} \right) \frac{\rho}{RT} \\ + \frac{3}{2} \left(bRT - a - \frac{d}{T} \right) \frac{\rho^2}{RT} + \frac{6\alpha}{5} \left(a + \frac{d}{T} \right) \frac{\rho^5}{RT} \\ + \frac{C}{\gamma RT^3} \left[1 - \left(1 - \frac{1}{2} \gamma \rho^2 - \gamma^2 \rho^4 \right) \exp(-\gamma \rho^2) \right] \quad (15)$$

At saturation, the liquid and vapor fugacities are equal. the ratio of these computed fugacities can be used to generate a better guess for $P_{\text{saturation}}$. The iteration is continued until the fugacities converge within a specified tolerance. The final $P_{\text{saturation}}$ guess is the saturation pressure at the known temperature.

All working fluids using the Starling equation of state 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 defined to be equal to zero.. The enthalpy of the working fluid at this temperature is equal to the standard heat of formation from carbon (graphite) and hydrogen (gas). For isobutane, this enthalpy is -783.04 Btu/lbm; for n-butane it is -733.1 Btu/lbm. For this reason the enthalpies of these fluids are always negative in the normal operating range. (64)

The set of Starling equations just described can be used for other hydrocarbons like propane, methane, propylene, isopentane, and ethylene. The only requirement for the computer model are the set of constant coefficients for these fluids.

A detailed description of the Martin-Hou and Starling equations of state has been given here since they are the backbone of the thermodynamic cycle analysis, which will be explained following the discussion of selected thermodynamic cycles and program control logic.

Selection of Thermodynamic Cycles - The binary fluid cycles selected for use in this computer model are based on simple Rankine cycles. In other words, no provisions have been made for reheat, regeneration, or combinations of cycles. 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 practical. Both of these phenomena exist because of the low-energy system involved with a geothermal resource.

In many previous sections of this report, two different types of simple Rankine cycles have been described. These are the sub-critical and super-critical cycles. Although there exist many differences in these two cycle types, both have been proposed for use with geothermal resources because of the wide variety of geothermal reservoirs and working fluid types.^(65,66) Figures 3 and 4 illustrate the basic sub-critical and super-critical Rankine cycles, respectively.

FIGURE 3. Sub-Critical Cycle

FIGURE 4. Super-Critical Cycle

The working fluid state points are indicated; this nomenclature will be used throughout this report. In these figures, the ordinate is pressure and the abscissa is enthalpy. The working fluid vapor dome and critical point are shown. The processes that make up these cycles are as follows:

- 1-2EST: reversible adiabatic (isentropic)
 expansion of the working fluid
 in the turbine.
- 1-2: actual expansion path of the working
 fluid in the turbine, reflecting
 turbine inefficiencies.
- 2-2SATV: constant pressure heat extraction
 from the working fluid between the
 turbine exhaust and the saturated
 vapor state.

- 2SATV-
2SATL: constant pressure removal of the heat of vaporization from the working fluid.
- 2SATL-3: compression of the liquid working fluid.
- 3-1: constant pressure addition of heat to the working fluid from the geothermal fluid.
- 3-4: (sub-critical only) heat addition to the working fluid between pump exhaust and the saturated liquid state at constant pressure.
- 4-5: (sub-critical only) additions of the heat of vaporization to the working fluid at constant pressure.
- 5-1: (sub-critical only) superheating of the working fluid at constant pressure.

The thermodynamic cycle selected for a particular set of conditions is dependent upon the geothermal fluid temperature, the available sink temperature, and the working fluid. This selection is based upon the optimization technique to be discussed after description of the model. For flexibility, an arbitrary cycle can be chosen by specifying the maximum working fluid pressure, the heat exchanger "approach," and the heat exchanger pinch point.

Control Logic - The control of the binary fluid cycle plant model is accomplished through the use of three subroutines: DBINARY, BINARY, and OPBIN. The following input data are supplied at the beginning of the program:

- type of plant - binary
- super-critical cycle - false
- working fluid type
- pressure factor
- geothermal fluid temperature
- geothermal flow rate per well
- plant size
- well spacing
- average well life
- reservoir size
- wet and dry bulb air temperatures
- geothermal fluid pressure at plant inlet
- cooling water pressures
- geothermal fluid velocities through heat
exchanger tubes
- heat exchanger tube diameters
- fouling factors in heat exchangers.

Once this input data is loaded, subroutine DBINARY is called. Subroutine DBINARY serves a threefold purpose; it insures an adequate flow of geothermal fluid to the power plant through the transmission piping from the geothermal

wells, maintains the pressure of the geothermal fluid high enough to prevent two phase flow anywhere in the system, and controls the design of the reinjection system which returns the spent geothermal fluid to the reservoir. Subroutine DBINARY also called subroutine BINARY, which controls the design of the binary fluid cycle plant. Subroutine BINARY determines 1) the thermodynamic cycle state points, and 2) the operating characteristics of the major components of the power plant. The plant design made in BINARY is based upon the optimum selection made by subroutine OPBIN, which is discussed in the fourth major section of this report.

Collectively, these three subroutines control the design of the binary cycle power plant and insure a proper interface with the other sections of the total geothermal power system.

Thermodynamic Cycle Calculations - The determination of the thermodynamic state points for the binary cycle power plant is outlined in Figure 5. Once subroutine BINARY has received the optimum cycle characteristics from subroutine OPBIN, subroutine SETCON is called. SETCON loads for later use the values for the coefficients used in the equation of state and other equations discussed in the Working Fluid sub-section. Only those coefficients required for the particular working fluid being used are loaded. The available working

FIGURE 5. Computer Flow Diagram for Determination of Thermodynamic State Points

fluids are isobutane, n-butane, R-11, R-12, R-21, R-22, R-113, R-114, and ammonia. If the selected working fluid is a light hydrocarbon, subroutine SETCON calls subroutine BUTANE once the coefficients are loaded. If the selected working fluid is a halogenated hydrocarbon (Freon) or ammonia, SETCON calls subroutine FREON. Since BUTANE and FREON are quite different in nature due to the dissimilarity of the equation of states, these subroutines will be discussed separately. Subroutine BUTANE is described first.

Subroutine BUTANE computes the thermodynamic state points for a hydrocarbon working fluid, with either a sub-critical or super-critical cycle. From subroutine OPBIN and input data, the entering geothermal fluid temperature and pressure, heat exchanger approach and pinch point, maximum working fluid pressure, and minimum cooling water temperature are known. Each of the state points in the cycle are computed in the following stepwise fashion. It will be helpful to refer to Figure 6 to recognize specific points in the cycle.

FIGURE 6. Sub-Critical Cycle State Points

A) The minimum working fluid temperature $T2SATL$, at the outlet of the surface condenser is set based upon the lowest cooling water temperature. This temperature difference is defined as the TTD of the condenser. Based upon this temperature, $T2SATL$, the saturation pressure $P2$ is determined in subroutine PTW0. PTW0 determines through an iterative process the pressure at which the largest and smallest density roots of equation (10) have equal fugacities (eq. 15). The density roots are determined in subroutine RHOITR and the fugacities are calculated in subroutine FUG.

This iterative technique is described in the discussion of the Starling equation of state. The largest density root at convergence is the

density of the working fluid at state 2SATL. Once the saturation pressure P_2 is known at temperature T_{2SATL} , the enthalpy at this state point, H_{2SATL} , is determined using subroutines HEQST and HIDEAL. Subroutines HEQST and HIDEAL are composed of equations (11) and (12), respectively.

- B) After the required thermodynamic properties at state 2SATL are determined in step A, the properties at state 1 are determined in step B. State 1 corresponds to the working fluid at the turbine inlet. The pressure, P_1 , and the heat exchanger "approach," APP, are known from subroutine OPBIN. The working fluid temperature, T_1 , is computed by subtracting the heat exchanger "approach" from the inlet geothermal fluid temperature, WT_1 . The density (D_1) of the working fluid at State 1 is computed next from equation (10), which is listed in subroutine PEQST. Since this equation is expressed as pressure equaling a function of temperature and density, the density is determined iteratively in the following manner: An initial value of D_1 is calculated based upon an ideal gas at T_1 and P_1 . The pressure corresponding to the temperature T_1 and the guess for density D_1 is computed in

subroutine PEQST (equation 10). This value of pressure is compared with the known value of P_1 . If the difference between P_1 and the computed pressure is less than 0.5 psia, the density D_1 has been found. If this difference is greater, a new estimate for D_1 is found based on the following equation,

$$D_{1_{\text{new}}} = D_{1_{\text{old}}} * (1.0 + (P_1 - P_{\text{guess}})/P_1). \quad (16)$$

This new value of D_1 is then used to compute a new calculated value for pressure.

The iteration is continued until the convergence criteria is satisfied. The enthalpy H_1 at state 1 is computed with subroutines HEQST and HIDEAL (equations (11) and (12)). The entropy, S_1 , at state 1 is calculated using equations (13) and (14) in subroutines SEQST and SIDEAL, respectively.

- C) This step is involved with the state 2EST, which is an isentropic expansion from State 1 to pressure P_2 . The entropy at state 2EST, S_{2EST} , is therefore equal to S_1 computed in step B. The temperature and density, T_{2EST} and D_{2EST} , are calculated by simultaneously solving the equations (10) and (13) with (14). These equations are of the form

$$P = f(T, D) \quad (\text{equation 10})$$

$$S = f(T, D) + f(T) \quad (\text{equation 13 \& 14})$$

With the pressure (P2) and entropy (S2EST) known, T2EST and D2EST are found using a subroutine called RQNW. Equations (10), (13) and (14) are supplied to RQNW through subroutine BFT2D2.

Subroutine RQNW solves systems of nonlinear, algebraic, or transcendental equations with random initial estimates using a quasi-Newton algorithm. Once the temperature and density of state 2EST are known, the enthalpy H2EST can be determined using subroutines HEQST and HIDEAL.

- D) In step D, the working fluid flow rate required for the binary fluid cycle plant is computed upon the basis of plant size GRKW, turbine inlet enthalpy H1, ideal turbine exhaust enthalpy H2EST, and the overall turbine efficiency. The equation for this computation is as follows:

$$Q_{ISO} = GRKW * 3414.43 / (H1 - H2EST) / OEFF \quad (17)$$

This calculation is performed in subroutine WFFRT. The actual enthalpy at state 2, H2, is also computed as follows:

$$H2 = H1 - (H1 - H2EST)(OEFF) \quad (18)$$

In addition, the actual Rankine cycle efficiency REFF is determined according to the equation

$$\text{REFF} = (H1 - H2) / (H1 - H2\text{SATL}) \quad (19)$$

The enthalpy $H2$ and pressure $P2$ are now known at state 2, which is the actual working fluid condition at turbine exhaust. The temperature and density, $T2$, and $D2$, are determined using subroutines RQNW, PEQST, HEQST, and HIDEAL in a manner similar to that in step C, except for the substitution of the enthalpy equations (11) and (12) for the entropy equations (13) and (14). Equations (10), (11), and (12) are supplied to RQNW through subroutine BFT2V2.

- E) Step E is performed only if a subcritical cycle is under consideration, since this step is involved with states 4 and 5. The saturation temperature at pressure $P1$ is determined using a subroutine called TFOUR. This temperature is the same at states 4 and 5 (i.e. $T4 = T5$). The calculation of $T4 = T5$ is accomplished in a manner similar to that described in step A, except the pressure at saturation is now known instead of the temperature at which the liquid and vapor densities result in equal fugacities using the known pressure $P1$. In the process, the actual saturated liquid density $D4$ and saturated vapor density $D5$ are found. Using these computed temperatures and densities, the enthalpy at

states 4 and 5 are computed using subroutines HEQST and HIDEAL. Likewise, the entropy at states 4 and 5 are determined using subroutines SEQST and SIDEAL. The pressures P4 and P5 are determined based on P1.

- F) After steps A through E the thermodynamic properties are known at states 2SATL, 1, 2EST, 2, 4, and 5. The conditions at state 3 are now determined. The pressure at state 3 is based on P1. The enthalpy and temperature at state 3 are found in subroutine PUMP, which simulates the compression of the liquid working fluid from state 2SATL to state 3. The change in enthalpy between states 2SATL and 3 is based on an expression for pump work as follows:

$$\Delta H = H_3 - H_{2SATL} = \frac{(P_3 - P_2)(144)(.00128509 \text{ Btu/ft}^3 \text{ lb})}{D_{2SATL} (\text{pump efficiency})} \quad (20)$$

$$\text{then } H_3 = \Delta H + H_{2SATL} \quad (21)$$

The temperature at state 3 is determined from ΔH and the temperature at 2SATL as follows:

$$T_3 = T_{2SATL} + \frac{(\Delta H)(\text{Mol Wt})(K-1)}{(K)(R)} \quad (22)$$

where K is the ratio of specific heats, and R is the universal gas constant.

G) Step G is applicable to the subcritical cycle only. The geothermal fluid temperature at the inlet to the preheater, WT3, is computed based on the sum T4 (from step E) and the heat exchanger pinch point. The required geothermal flow rate, QWAT, is also calculated in step G according to the equation

$$QWAT = \frac{QISO(HI-H4)}{Cp(WT1-WT3)} \quad (23)$$

In steps A through G, all required thermodynamic properties of the cycle state points are computed for light hydrocarbons. A similar stepwise technique is used to determine state points for the Freons and ammonia. Subroutine FREON is used for this purpose, and a description of the logic for this subroutine follows.

Subroutine FREON computes the thermodynamic state points for the halogenated hydrocarbons and ammonia, with either a sub-critical or super-critical cycle. As in BUTANE, this subroutine is supplied with values for inlet geothermal fluid temperature and pressure, heat exchanger "approach" and pinch point, maximum working fluid pressure, and minimum cooling water temperature. The generation of thermodynamic properties for each state point is done in a step-wise fashion. The notation for each step is in double letters to distinguish these steps from the BUTANE subroutine.

AA) As previously noted for the hydrocarbons, the minimum working fluid temperature, T2SATL, is set based upon the lowest cooling water temperature available. The saturation pressure P2 at this temperature is computed based upon equation (7), which is listed in subroutine VPRESA. The density of the working fluid (DSAT) as a saturated liquid is calculated based upon one of the following three equations.

TABLE 4. Saturated Density Equations

<u>Working Fluid of Interest</u>	<u>Subroutine Called</u>	<u>Density Equation [lb_m/ft³]</u>
R-11, R-22, NH ₃	DNLIQA	$\rho = AL + BL(1 - T/T_c)^{1/3} + CL(1 - T/T_c)^{2/3}$ $+ DL(1 - T/T_c) + EL(1 - T/T_c)^{4/3}$
R-12, R-114	DNLIQB	$\rho = AL + BL(T_c - T) + CL(T_c - T)^{1/2}$ $+ DL(T_c - T)^{1/3} + EL(T_c - T)^2$
R-21, R-113	DNLIQC	$\rho = AL + BL(T) + CL(T)^2$

The coefficients A1, BL, CL, DL, and EL are provided as constants for the particular working fluid of interest from subroutine SETCON. The enthalpy at state 2SATL is computed based upon the enthalpy at P2 as a saturated vapor, since the saturated liquid enthalpy at low temperatures is not computed accurately by the Martin-Hou

equation of state. The equation used to compute the enthalpy at state 2SATL, H2SATL, is as follows:

$$H2SATL = H2SATV - SDHVAP \quad (24)$$

where H2SATV = saturated vapor enthalpy at P2

and SDHVAP = heat of vaporization at P2.

The first quantity to be determined in equation (23) is H2SATV. In order to compute H2SATV, the specific volume at this state, SVSATV, must first be determined. An initial guess based on ideal gas laws is made for SVSATV. This estimated specific volume and the known temperature T2SATL are used in equation (6) to calculate a pressure. Equation (6) is positioned in a subroutine called EQSTAP. The pressure computed in subroutine EQSTAP is compared with the known pressure P2. If the absolute value of the difference between these two pressures is within 0.5 psia, a suitable value for SVSATV has been found. If the convergence criterion is not satisfied, a new estimate for SVSATV is found based on the following equation

$$SVSATV_{new} = SVSATV_{old} \times (1 + (P_{calc} - P2)/P2) \quad (25)$$

This new value of SVSATV is then used to determine a new calculated value of pressure from subroutine EQSTAP, which is then compared to P2. Iteration continues until convergence is accomplished.

The enthalpy at the saturated vapor state, H2SATV is then computed using subroutine HVAP which contains equation (8), based upon T2SATL, P2 and SVSATV.

The heat of vaporization is the second quantity to be determined in equation (24). This quantity is defined by the following equation:

$$\text{SDHVAP} = Z * \text{SDPDT} * (\text{T2SATL} + 459.67) \left(\text{SVSATV} - \frac{1}{\text{DSAT}} \right) \quad (26)$$

In equation (26) for the heat of vaporization, Z is a constant with the value of

$$0.185053 \frac{\text{in}^2}{\text{ft}^2} \cdot \frac{\text{Btu}}{\text{ft-lb}_f}, \text{ SDPDT is the derivative}$$

of pressure with respect to temperature, and DSAT is the liquid density determined earlier in step AA. SDPDT is listed as subroutine VPRESB, and is defined as

$$\text{SDPDT} = P \left[\frac{-\ln 10 \times \{BFR + EF \times \log_{10}(F - T)\}}{T^2} + \frac{C - E}{T} + D \ln 10 \right] \quad (27)$$

The constants B, C, D, E and F are provided in subroutine SETCON. The enthalpy of saturated liquid, H2SATL can now be computed by equation (24).

- BB) After the thermodynamic properties at state 2SATL are determined in step AA, the properties at state 1 are determined in step BB. The pressure P1 is known from subroutine OPBIN. The temperature T1 is the difference between the entering geothermal fluid temperature WT1 and the heat exchanger "approach" APP. The specific volume at state 1, V1, is computed through an identical technique used in step AA for SVSATV. The enthalpy H1 is computed from subroutine HVAP using equation (8). The entropy S1 is calculated by using equation (9), listed in subroutine SVAP.
- CC) As in the subroutine BUTANE, the thermodynamic properties at state 2EST are computed after state 1. Again the entropy S2EST is equal to S1 because of the isentropic expansion from P1 to P2. The temperature and specific volume at this

state, T2EST and V2EST, are found by simultaneously solving equations (6) and (9) (from subroutines EQSTAP and SVAP which are supplied through subroutine EF2TV2) using subroutine RQNWT. Subroutine RQNWT is the same one as described in step C of BUTANE. The computed values of T2EST and V2EST are then used with the value of P2 to determine H2EST in subroutine HVAP.

DD) Step DD is used to compute the following values:

QISO - working fluid flow rate

H2 - enthalpy at state 2

REFF - Rankine cycle efficiency

T2 - temperature at state 2

V2 - specific volume at state 2. These values are computed in the same manner as indicated in step D of subroutine BUTANE. QISO is found from equation (17), H2 from equation (18), and REFF from equation (19). The values for T2 and V2 are found from subroutines RQNWT, EQSTAP, and HVAP. The access for subroutines HVAP and EQSTAP is provided to RQNWT by subroutine FFT2V2.

EE) The properties at states 4 and 5 are computed in step EE. The saturation temperature $T_4 = T_5$ is found at pressure P1 using an iterative

technique with subroutine VPRESA (equation (7)). An initial guess for this temperature is made. This temperature guess is used in equation (7) to determine a pressure. This computed pressure is compared with the known value of P_1 . The value of $T_4 = T_5$ is adjusted until the calculated pressure converges to P_1 within 0.5 psia. The specific volume at state 4 (V_4 , saturated liquid) is computed as the inverse of the appropriate density equation listed in Table 4. The specific volume at state 5, V_5 , is determined iteratively using subroutine EQSTAP, similar to the method used to locate SVSATV in step AA. The pressures at states 4 and 5 (P_4 and P_5) are based upon P_1 . Using T_4 and V_4 , the enthalpy and entropy at state 4 are computed using subroutines HVAP and SVAP. The enthalpy and entropy at state 5 (H_5 and S_5) are determined in a similar manner. As in step E of subroutine BUTANE, step II is used with a sub-critical cycle only.

- FF) The calculation of properties at state 3 are determined in the same manner as described in step F of BUTANE.
- GG) This step is identical to step G of subroutine BUTANE. This step is only used for the sub-critical cycle.

In steps AA through GG, all necessary thermodynamic state properties are determined for halogenated hydrocarbons and ammonia. Once BUTANE or FREON computations are completed, the binary cycle plant program passes back through subroutine SETCON to subroutine BINARY to compute the remaining values required to describe the power plant. These remaining values include the power plant component descriptions and the plant operating characteristics.

Physical Fluid Properties - Prior to the discussion of the remaining sections of subroutine BINARY, a description of the method used to determine the physical properties of the fluids in question is necessary. Throughout the entire binary cycle plant program, values for the physical properties of various fluids are required. This information is provided in a single subroutine called FLUID. Each physical property for a particular fluid is expressed as a polynomial function of temperature in its most elementary form.

$$\text{Property value} = K_0 + K_1T + K_2T^2 + K_3T^3 \text{ for} \\ T_x < T < T_y \quad (28)$$

These expressions are valid for the temperature range T_x to T_y , which is dependent upon available data. These data were amassed through a search of numerous reference texts and publications. (67-76) When a physical property is desired, the following statement is made in the program:

Property Value = FLUID (I, J, T)

(29)

where T = temperature at which the property is desired in °F, and:

I = 1, 13 - Property Index

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

J = 1, 10 - 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
- = 10 - NH_3

Within the specified temperature range, there is good agreement between the actual and predicted values. For those properties which are strong functions of pressure as well as temperature, their use should be limited to pressures near the saturation pressure. If a property value is requested outside the specified temperature range, a warning is printed out in output to indicate a possible erroneous result.

Operational Characteristics - The operating characteristics and plant component descriptions are described in this sub-section. A logic flowsheet for this part of the binary cycle plant modeled in subroutine BINARY is shown in Figure 7. Once the thermodynamic state points are determined in subroutines SETCON and BUTANE or FREON, the flow rates, sizes, and other pertinent information are determined for various plant components.

Subroutine ACWRT computes the inlet and outlet cooling water temperatures and cooling water flow rates of the turbine oil cooler and the generator cooler, described in section III-2 of this report under System Cooling. Water flow rates are determined through subroutine QCC. The values computed in subroutine ACWFRT are based on

FIGURE 7. Computer Flow Diagram for Binary Fluid Cycle Plant Model

actual operating data for these components at existing geothermal power plants.

The next step in the BINARY subroutine is to describe the geothermal fluid/working fluid heat exchangers. The physical nature of these heat exchangers is discussed in Section III-2 of this report. Two separate heat exchanger subroutines are provided; SUBHEX for a sub-critical cycle and SUPHEX for a super-critical cycle. Subroutine SUBHEX is described first.

The sub-critical heat exchanger model in SUBHEX consists of three separate heat exchanger types; a preheater, vaporizer, and superheater. Reference to Figure 2 is helpful in visualizing this configuration. The following input information is provided to subroutine SUBHEX from BUTANE

- a) geothermal flow rate; QWAT [lb_m/hr]
- b) working fluid flow rate; QISO [lb_m/hr]
- c) geothermal fluid temperatures; WT1, WT2, WT3, and WT4 [$^{\circ}\text{F}$] where

WT1 = inlet temperature

WT2 = superheater outlet (vaporizer inlet)
temperature

WT3 = vaporizer outlet (preheater inlet)
temperature

WT4 = preheater outlet temperature

- d) geothermal fluid pressures, WP1, WP2, WP3, and WP4 [psia] (same nomenclature as in c)
 - e) working fluid temperatures; T3, T4, T5, and T1 [°F] where
 - T1 = superheater outlet temperature
 - T3 = preheater inlet temperature
 - T4 = preheater outlet (vaporizer inlet) temperature
 - T5 = vaporizer outlet (superheater inlet) temperature
 - f) working fluid enthalpies; H3, H4, H5, and H1 [Btu/lb_m] (same nomenclature as e)
 - g) working fluid pressures; P3, P4, P5, and P1 [psia] (same as nomenclature in 3)
 - h) fouling resistance factors for inside and outside of heat exchanger tubes; FOULFI, FOULFO [hr-ft² °F/Btu]
 - i) inside and outside tube diameters, DIAL, DIAOL, DIAV, DIAOV, DIAS, and DIAOS [ft] where
 - DIAL - preheater inside tube diameter
 - DIAOL - preheater outside tube diameter
 - DIAV - vaporizer inside tube diameter
 - DIAOV - vaporizer outside tube diameter
 - DIAS - superheater inside tube diameter
 - DIAOS - superheater outside tube diameter
-

- j) geothermal fluid velocities through heat exchanger tubes, WVELL, WVELV, and WVELS [ft/hr] where

WVELL - velocity in preheater

WVELV - velocity in vaporizer

WVELS - velocity in superheater

- k) type of working fluid

Using the listed input data, the following characteristics of each heat exchanger are determined

- a) inside heat transfer coefficient
- b) outside heat transfer coefficient
- c) overall heat transfer coefficient
- d) number of tubes
- e) equivalent shell-side flow diameter
- f) total surface area
- g) effective heat exchanger length

Subroutine SUBHEX determined the characteristics of each heat exchanger in a similar step-wise manner. The preheater is defined first, followed by the vaporizer and superheater. The following steps indicate the logical flow in subroutine SUBHEX for one exchanger:

- 1) The average of the inlet and outlet temperatures is computed for both the geothermal and working fluids. These average temperatures are determined in subroutine AVGTMP and are used solely for the determination of physical properties.
- 2) The thermal conductivity, density, specific heat,

and viscosity of the geothermal fluid are calculated at the average temperature determined in step 1 using subroutine FLUID.

- 3) The Reynolds number and Prandtl number for the geothermal fluid are calculated in subroutines REYNO and PRNDTL, respectively. The equations for these quantities are as follows:

$$Re = \frac{\rho V D}{\mu} \quad (30)$$

$$Pr = \frac{C_p \mu}{K} \quad (31)$$

where V is the geothermal fluid velocity and D the inside tube diameter.

- 4) The inside heat transfer coefficient is determined in subroutine HTCDEF, which contains the Dittus-Boelter equation for fully developed turbulent flow; (77)

$$h_i = \frac{0.023}{D} K (Re)^{0.8} (Pr)^n, \quad (32)$$

where $n = 0.3$ for cooling

$n = 0.4$ for heating.

- 5) The number of tubes required for the heat exchanger in question is determined using subroutine TUBES, which contains the following equation:

$$\text{no. of tubes} = \frac{4\dot{m}}{\pi \rho V D^2} \quad (33)$$

The number of tubes is converted to an integer value.

- 6) The total flow area and equivalent shell-side flow diameter are calculated in subroutine QAREA. The total flow area equals the unit flow area times the number of tubes. The unit flow area is the flow passage area for the working fluid calculated as

$$A_{\text{unit}} = \frac{s^2 \sin 60^\circ}{2} - \frac{1}{2} \frac{D^2}{4} \quad (34)$$

In this expression, s is the equilateral tube spacing in ft and D is the outside tube diameter. The equivalent flow diameter is a hydraulic diameter, as follows

$$D_e = \frac{4 * A_{\text{unit}}}{\text{Wetted Perimeter}} \quad (35)$$

- 7) Step 7 is unique for each heat exchanger type. In the preheater, the specific heat, thermal conductivity, density, and viscosity of the liquid working fluid are computed using subroutine FLUID with the average temperature found in step 1. In the vaporizer, the saturated liquid specific heat, saturated liquid thermal conductivity, saturated liquid and vapor densities, and saturated liquid and vapor viscosities are computed for the boiling

working fluid in subroutine FLUID. In the superheater, the specific heat, thermal conductivity, and viscosity are determined for the gaseous working fluid using average temperatures and subroutine FLUID. The density is also calculated, but is determined iteratively for hydrocarbons using subroutine PEQST and for the Freons and ammonia using subroutine EQSTAP.

- 8) The velocity of the working fluid in the shell side of the heat exchanger is determined using subroutine VELOCITY, using this equation

$$V = \frac{\dot{m}}{\rho \times \text{total flow area}} \quad (36)$$

For the vaporizer, the velocity of the saturated liquid is computed.

- 9) The Reynolds number and Prandtl number are determined for the working fluid using equations (30) and (31), respectively. In the vaporizer, the Reynolds and Prandtl number are based on liquid working fluid properties.
- 10) The outside heat transfer coefficient for the preheater and superheater is computed using equation (32). The vaporizer involves boiling, and a special technique for computing the heat transfer coefficient is required. Chen's correlation based upon two-phase properties is used

for this coefficient. (81) Chen states that a reasonable boiling coefficient can be determined using the following equation

$$h_{\text{boiling}} = F \times .023 \times \frac{K_L}{D} \times (Re_L)^{0.8} (Pr_L)^{0.4} \quad (37)$$

where K_L , Re_L , and Pr_L are all based on saturated liquid properties. F is a function of the Martinelli parameter which is defined as

$$\text{Martinelli Parameter} = M = \left(\frac{\rho_l}{\rho_v} \right)^{\frac{1}{2}} \left(\frac{\mu_v}{\mu_l} \right)^{\frac{1}{10}} \quad (38)$$

The factor F has been reduced to a polynomial;

$$F = 1.77 + 1.179M - .0075M^2 + .000029M^3 \quad (39)$$

- 11) The overall heat transfer coefficient for each heat exchanger is computed as a function of the inside and outside heat transfer coefficients, the tube wall resistance, and the fouling factors. Subroutine OHTCOE handles this computation and includes the equation;

$$U = \frac{1}{\frac{D_o}{D_i h_i} + \frac{D_o}{2K_{\text{wall}}} \ln\left(\frac{D_o}{D_i}\right) + \frac{1}{h_o} + FOULFO + FOULFI} \quad (40)$$

- 12) The required outside heat transfer area and the resulting effective heat exchanger length are computed in step 12. The heat transfer area is

found in subroutine HTAREA with the following equation;

$$A = \frac{Q_{ISO} \times (H_{out} - H_{in})}{U \times LMTD} \quad (41)$$

The factor LMTD is the log mean temperature difference, defined as;

$$LMTD = \frac{(WT_i - T_j) - (WT_j - T_i)}{\ln[(WT_i - T_j)/(WT_j - T_i)]} \quad (42)$$

where indices i and j correspond to specific state points in question. The heat transfer length is computed in subroutine HTLNGH with the following expression;

$$\text{Length} = \frac{A}{\pi D_o \times \text{no. of tubes}} \quad (43)$$

These twelve steps outline the procedure used to determine the characteristics of the sub-critical heat exchangers. A typical temperature profile for the sub-critical heat exchanger assembly is shown in Figure 8.

The super-critical heat exchanger is modeled in subroutine SUPHEX. No boiling section exists in this heat exchanger, indeed, the working fluid is always considered to be in the gaseous phase because its pressure is above the critical pressure. This heat exchanger cannot be modeled as a single unit with only end-point data known

FIGURE 8. Typical Temperature Profiles in Sub-Critical Heat Exchangers

since the thermodynamic and physical properties of the working fluid are quite variable and non-linear throughout the heat exchanger. For this reason, the supercritical heat exchanger was assumed to be a series of twenty-five separate heat exchangers, with each unit having an equal working fluid temperature difference. An accurate calculation of total outside heat transfer area can then be obtained by summing the 25 individual areas.

The input to subroutine SUPHEX is similar to the SUBHEX input, except for the lack of intermediate state points for the working and geothermal fluids. Also, the variable names for tube diameters and fluid velocities are different. The input list is as follows:

QWAT - geothermal flow rate

QISO - working fluid flow rate

WT1, WT4 - inlet and outlet geothermal fluid
temperatures

WP1, WP4 - inlet and outlet geothermal fluid
pressures

T3, T1 - inlet and outlet working fluid temperatures

H3, H1 - inlet and outlet working fluid enthalpies

P3, P1 - inlet and outlet working fluid pressures

FOULFI, FOULFO - inside and outside fouling factors

DIASC, DIAOSC - inside and outside tube diameters

DVEL - geothermal fluid velocity.

Seven different heat exchanger characteristics are determined in SUPHEX for each of the 25 heat exchanger sections:

- 1) inside heat transfer coefficient
- 2) outside heat transfer coefficient
- 3) overall heat transfer coefficient
- 4) number of heat exchanger tubes
- 5) equivalent shell-side flow diameter
- 6) heat transfer area
- 7) effective heat exchanger length.

For each of the heat exchanger sections, the following computational steps are taken.

S-1) The temperature TT3 and enthalpy HH3 at the inlet of the heat exchanger section are known from the previous section. If the section under consideration is the first one, TT3 and HH3 are equal to the end-point conditions T3 and H3. Temperature TT1 at the section outlet is set equal to TT3 plus the equal temperature increment $(\frac{T1-T3}{25})$. If a hydrocarbon working fluid is used, the density DD1 at the section outlet is determined by iteration with subroutine PEQST. The enthalpy HH1 is then computed with subroutines HEQST and HIDEAL using the values of TT1 and DD1. If the working fluid is a Freon or ammonia, the specific volume VV1 at the section outlet is determined by iteration with subroutine EQSTAP

and known pressure P1. The enthalpy HH1 is then calculated with subroutine HVAP using the values of TT1 and VV1.

S-2) In this step, the geothermal fluid inlet and outlet temperatures are set. The outlet geothermal fluid WTT4 for a section is set equal to the inlet temperature from the last section. If the section being analyzed is the first, this temperature WTT4 is set equal to the end-point outlet temperature WT4. The inlet geothermal fluid temperature for a section is defined as follows:

$$WTT1 = \frac{QISO (HH1 - HH3)}{QWAT C_p} + WTT4 \quad (44)$$

With steps S-1 and S-2, the inlet and outlet temperatures for both the geothermal and working fluids are known for the heat exchanger section in question.

S-3) The average temperature for the geothermal and working fluids in the section are computed in subroutine AVGTMP.

S-4) The thermal conductivity, density, specific heat, and viscosity of the geothermal fluid are determined from subroutine FLUID using the average temperature from step S-3.

S-5) The Reynolds and Prandtl numbers are computed for the geothermal fluid in subroutines REYNO and PRNDTL, using equations (30) and (31).

S-6) The inside heat transfer coefficient is determined from subroutine HTCDEF using equation (32).

S-7) The number of tubes required for the heat exchanger section is found in subroutine TUBES using equation (33).

S-8) The total flow area and equivalent flow diameter for the shell side of the heat exchanger section are determined in subroutine QAREA with equations (34) and (35).

S-9) The specific heat, thermal conductivity, density, and viscosity of the working fluid are calculated in subroutine FLUID using the average temperature computed in step S-3.

S-10) The Reynolds and Prandtl numbers are computed for the working fluid in this section from equations (30) and (31), respectively. The working fluid velocity used in equation (30) is found in subroutine VELOCITY.

S-11) The outside heat transfer coefficient is determined with subroutine HTCDEF using equation (32).

S-12) The overall heat transfer coefficient is computed using subroutine OHTCOE with equation (40).

S-13) The minimum temperature difference in this section is computed. The lowest value of this temperature difference for all 25 heat exchanger sections is the pinch point, and is controlled by adjusting the exit geothermal fluid temperature, WT4.

S-14) The heat transfer area and length of this section are computed in subroutines HTAREA and HTLNHG, respectively.

After the calculations in steps S-1 through S-14 are completed for all 25 heat exchanger sections, the heat transfer areas and lengths are summed to obtain total values. Average values for the three heat transfer coefficients are determined.

This procedure describes the model of the supercritical heat exchanger. A typical temperature profile for this heat exchanger is shown in Figure 9.

After describing the heat exchangers in subroutine SUBHEX or SUPHEX, subroutine BINARY calls CNDNSR to determine the characteristics of the desuperheater and condenser. These components are physically described in section III-2. The desuperheater uses cooling water to bring the working fluid to a saturated vapor state from its superheated state at the turbine exhaust. The condenser simply condenses the working fluid to a saturated liquid state.

FIGURE 9. Typical Temperature Profiles in Super-Critical Heat Exchangers

The input data to subroutine CNDNSR is listed as follows:

CWT1, CWT2 - inlet and outlet cooling water

temperatures, °F.

T2, P2, H2 - temperature, enthalpy, and pressure of the working fluid at turbine exhaust.

T2SATL, H2SATL - temperature and enthalpy of the working fluid at condenser outlet.

DIAC, DIAOC - inside and outside tube diameters, ft.

FOULFO, FOULFI - inside and outside fouling factors.

QISO - working fluid flow rate, lb/hr.

CWVEL - cooling water velocity in tubes, ft/hr.

NFLUID - type of working fluid.

The output and computational steps used in subroutine CNDNSR are the same as those in the vaporizer and superheater sections of subroutine SUBHEX, and are not repeated here. The single difference between subroutines CNDNSR and SUBHEX is the use of cooling water in the condenser tubes rather than geothermal fluid. Typical temperature profiles for the desuperheater and condenser are shown in Figure 10.

After subroutine CNDNSR, BINARY calls subroutine TOWER to obtain a description of the cooling tower. The cooling tower assumptions are discussed in section III-2 of this report. The input required for subroutine TOWER is as follows:

FIGURE 10. Typical Temperature Profiles in Desuperheater and Condenser

QCWAT - cooling water flow rate, lb/hr (computed
in subroutines ACWFRT and CNDNSR)

CWT2, CWT1 - inlet and outlet cooling water temperatures, °F

WBAT, DBAT - wet bulb and dry bulb air temperatures,
°F

RI - relative humidity, %/100

B - barometric pressure, in Hg.

The basic cooling tower characteristics determined in subroutine TOWER include the air flow rate, evaporation rate, make-up flow rate, and total head load. The cooling water temperature into the tower is the mass-weighted average of the exit cooling water temperatures from the condenser, desuperheater, turbine oil cooler, and generator cooler as shown in Figure 1. The cooling tower is modeled with the requirement that the approach temperature, i.e., temperature difference between the water exit temperature and the inlet wet bulb air temperature, be at least 8°F. The characteristics of the cooling tower are calculated in the steps described below:

- 1) The saturated vapor pressure of the entering air is computed at the dry bulb temperature.
- 2) The actual vapor pressure is determined from the relative humidity and the saturated vapor pressure.
- 3) Based upon the actual vapor pressure, the inlet air humidity is determined.

- 4) Assuming saturated air at the outlet, the outlet air enthalpy is determined.
- 5) Based upon water-weight balance and energy balance equations, the required air flow rate and water evaporation rate are computed.
- 6) The make-up water flow rate is determined based upon the water evaporation and blow-down rates.
- 7) The heat load is computed based upon the equation:

$$Q_T = QCWAT \times C_p \times (CWT_{in} - CWT_{out}) \quad (45)$$

These steps identify the characteristics of a cooling tower for the binary cycle power plant.

The final subroutine called from BINARY is LOSSES. This subroutine determines the internal power requirements for the cooling tower fans, working fluid pumps, cooling water pumps, and the geothermal reinjection pumps. Flow rates, pressure drops, and component efficiencies are supplied to subroutine LOSSES internally or from other subroutines. The total internal power consumed is obtained by summing the individual component power requirements. The net electrical output of the geothermal power plant is then calculated by subtracting the total internal power from the gross power output of the turbine-generator.

In summary, the binary cycle power plant is simulated through the use of the major subroutines BINARY,

OPBIN, SETCON, ACWFRT, SUBHEX, SUPHEX, CNDNSR, TOWER, and LOSSES. All of these subroutines save OPBIN have been described in this section in addition to numerous minor subroutines. Prior to describing subroutine OPBIN and the optimization technique, sample input and output of the binary fluid cycle plant are provided.

Sample Input/Output - Numerous case runs have been made with the binary cycle plant model for a variety of input parameters. Output from these cases has been in good agreement with available data on similar power plants. The computer has been exercised for the following parameter ranges and configurations:

- a) use of all available working fluids
- b) power plant sizes from 10 to 220 MWe
- c) inlet geothermal fluid temperatures from 200-600°F
- d) use of both sub-critical and super-critical cycles
- e) varying heat exchanger approaches and pinch points
- f) varying meteorological conditions
- g) varying heat exchanger geometries
- h) operation of a regenerative Rankine cycle
- i) operation of condensing geothermal steam in the geothermal/working fluid heat exchangers.

Input to this model is passed through subroutine DBINARY. Sample input for a 55 MWe plant using isobutane as a working fluid with a geothermal fluid temperature of 356°F is shown in Table 5. Output for this power plant is shown in Table 6 and includes information on thermodynamic state points, heat exchanger characteristics, condenser and cooling tower characteristics, internal losses, and overall results.

TABLE 5. Typical Input to Binary Cycle Plant Model

TYPE OF PLANT	BINARY
SUPERCRITICAL CYCLE	FALSE
FLUID	ISOBUTANE
PRESSURE FACTOR	2.0
FLUID TEMPERATURE	356°F
FLOW RATE PER WELL	500,000 lb/hr
PLANT SIZE	55,000 kWe
WELL SPACING	20 ACRES
AVERAGE WELL LIFE	15 YEARS
RESERVOIR SIZE	275,000 kW
WET BULB AIR TEMPERATURE	50.0°F
DRY BULB AIR TEMPERATURE	70.0°F

**TABLE 6. Typical Computer Output for
Binary Cycle Plant Model**

**ISOBUTANE SUBCRITICAL BINARY CYCLE FOR 55 MWE PLANT
BINARY POWERPLANT SYSTEM AND TRANSMISSION MODEL**

THE WORKING FLUID IS ISOBUTANE
THE CYCLE IS SUB-CRITICAL
COOLING WATER

TURBINE OIL COOLER	
INLET TEMPERATURE (DEG. F)	6.200000E+01
OUTLET TEMPERATURE (DEG. F)	1.200000E+02
FLOW RATE (LB/HR)	1.616947E+04
HYDROGEN COOLER	
INLET TEMPERATURE (DEG. F)	6.200000E+01
OUTLET TEMPERATURE (DEG. F)	1.200000E+02
FLOW RATE (LB/HR)	6.142880E+04

FINAL CYCLE CONDITIONS

WORKING FLUID CONDITIONS

TEMPERATURES (DEG. F)	
APPHX-HEAT EXCHANGER APPROACH	3.000000E+01
T1-TURBINE INLET	3.257216E+02
T2EST-TURBINE OUTLET (IDEAL)	2.068035E+02
T2-TURBINE OUTLET (ACTUAL)	2.241968E+02
T2SATL-CONDENSER OUTLET	9.200000E+01
T3-PREHEATER INLET	9.590698E+01
T4-VAPORIZER INLET	2.549341E+02
T5-SUPERHEATER INLET	2.549341E+02
PRESSURES (PSIA)	
P1-TURBINE INLET	4.421796E+02
P2-TURBINE EXHAUST	6.493994E+01
P3-PREHEATER INLET	4.421796E+02
P4-VAPORIZER INLET	4.421796E+02
P5-SUPERHEATER INLET	4.421796E+02

SPECIFIC VOLUMES (CU.FT./LB.)

V1-TURBINE INLET	2.392818E-01
V2EST-TURBINE OUTLET (IDEAL)	1.784288E+00

ENTHALPY (BTU/LB.)

H1-TURBINE INLET	-5.576894E+02
H2-TURBINE EXHAUST	-5.895865E+02
H2EST-TURBINE OUTLET (IDEAL)	-5.982943E+02
H3-PREHEATER INLET	-7.886563E+02
H4-VAPORIZER INLET	-6.661859E+02
H5-SUPERHEATER INLET	-6.109138E+02

ENTROPY (BTU/LB/DEG. F)

S1-TURBINE INLET	1.251203E+00
S2EST-TURBINE OUTLET (IDEAL)	1.251203E+00

GEO THERMAL FLUID CONDITIONS

WATER TEMPERATURES (DEG. F)

WT1-SUPER HEATER INLET	3.557216E+02
WT2-VAPORIZER INLET	3.164122E+02
WT3-PREHEATER INLET	2.749341E+02
WT4-PREHEATER EXIT (PRESET)	1.800000E+02
EST-T4-PREHEATER EXIT (ACTUAL)	1.837415E+02
DELPHX-TOTAL PRESSURE DROP	7.500000E+01

TURBINE EFFICIENCY

IDEAL RANKINE EFFICIENCY	1.739092E+01
ACTUAL RANKINE EFFICIENCY	1.367489E+01
OVERALL TURBINE EFFICIENCY	7.863236E+01

HEAT EXCHANGERS SUR-CRITICAL CYCLE
TEMPERATURE (DEG. F) - PRESSURE (PSIA) - ENTHALPY (BTU/LB)

WORKING FLUID

INLET TEMPERATURE
INLET PRESSURE
INLET ENTHALPY
OUTLET TEMPERATURE
OUTLET PRESSURE
OUTLET ENTHALPY

GEO THERMAL FLUID

INLET TEMPERATURE
INLET PRESSURE
OUTLET TEMPERATURE
OUTLET PRESSURE
INSIDE TUBE DIAMETER (FT)
OUTSIDE TUBE DIAMETER (FT)
NUMBER OF TUBES
EQUIVALENT OUTSIDE DIAMETER (FT)
INSIDE HEAT TRANSFER COEFFICIENT (BTU/HR/SQ.FT./DEG. F)
OUTSIDE HEAT TRANSFER COEFFICIENT (BTU/HR/SQ.FT./DEG. F)
OVERALL HEAT TRANSFER COEFFICIENT (BTU/HR/SQ.FT./DEG. F)
HEAT TRANSFER AREA (SQ.FT.)
HEAT TRANSFER LENGTH (FT.)

SUPER HEATER

2.549341E+02
4.421796E+02
-6.109138E+02
3.257216E+02
4.421796E+02
-5.576894E+02

VAPORIZER

2.549341E+02
4.421796E+02
-6.661859E+02
2.549341E+02
4.421796E+02
-6.109138E+02

PREHEATER

9.590698E+01
4.421796E+02
-7.886563E+02
2.549341E+02
4.421796E+02
-6.661859E+02

3.557216E+02
2.297970E+02
3.164122E+02
2.047970E+02
1.000000E-01
1.104000E-01
6.190000E+02
1.078814E-01
3.075677E+03
5.874672E+02
2.356702E+02
3.030672E+04
1.411654E+02

3.144122E+02
2.047970E+02
2.749341E+02
1.797970E+02
1.000000E-01
1.104000E-01
6.040000E+02
1.078814E-01
2.800756E+03
2.451740E+03
3.351092E+02
2.628090E+04
1.254966E+02

2.749341E+02
1.797970E+02
1.837415E+02
1.547970E+02
1.000000E-01
1.104000E-01
5.850000E+02
1.078814E-01
2.341804E+03
7.199818E+02
2.473775E+02
6.358126E+04
3.133670E+02

MAIN CONDENSER
TEMPERATURE (DEG. F) - PRESSURE (PSIA) - ENTHALPY (HTU/LB)

WORKING FLUID

INLET TEMPERATURE
INLET PRESSURE
INLET ENTHALPY
OUTLET TEMPERATURE
OUTLET PRESSURE
OUTLET ENTHALPY

COOLING WATER

INLET TEMPERATURE
INLET PRESSURE
OUTLET TEMPERATURE
OUTLET PRESSURE
INSIDE TUBE DIAMETER (FT)
OUTSIDE TUBE DIAMETER (FT)
NUMBER OF TUBES
EQUIVALENT OUTSIDE DIAMETER (FT)
INSIDE HEAT TRANSFER COEFFICIENT (BTU/HR/SQ.FT./DEG. F)
OUTSIDE HEAT TRANSFER COEFFICIENT (BTU/HR/SQ.FT./DEG. F)
OVERALL HEAT TRANSFER COEFFICIENT (BTU/HR/SQ.FT./DEG. F)
HEAT TRANSFER AREA (SQ.FT.)
HEAT TRANSFER LENGTH (FT.)

DESUPERHEATER

2.241968E+02
6.493994E+01
-5.895865E+02
9.200000E+01
6.493994E+01
-6.531033E+02

CONDENSER

9.200000E+01
6.493994E+01
-6.531033E+02
9.200000E+01
6.493994E+01
-7.886663E+02

7.908658E+01
5.145814E+01
8.700000E+01
4.500000E+01
7.500000E-02
8.330000E-02
5.957000E+03
8.139963E-02
2.014879E+03
1.316453E+02
9.715224E+01
7.318839E+04
4.694818E+01

6.200000E+01
5.500000E+01
7.908658E+01
5.145814E+01
7.500000E-02
8.330000E-02
5.944000E+03
8.139963E-02
1.997433E+03
1.583436E+03
3.000001E+02
1.334799E+05
8.579133E+01

SURROUTINE TOWER -- COOLING TOWER

APPROACH TO COOLING TOWER (F) 1.200000E+01
COOLING TOWER AIR FLOW RATE (CFM) 0.322239E+06
COOLING TOWER EVAPORATION RATE (LB/HR) 9.445719E+05
WET BULB AIR TEMPERATURE (F) 5.000000E+01
DRY BULB AIR TEMPERATURE (F) 7.000000E+01
BAROMETRIC PRESSURE (IN. HG) 2.992000E+01
HUMIDITY RATIO 2.000000E-01
HEAT LOAD (BTU/HR) 1.154438E+09

COOLING WATER REQUIREMENTS (LR/HR)

TURBINE OIL COOLER 1.616547E+04
HYDROGEN COOLER 0.142880E+04
APPROACH TO CONDENSER (F) 5.000000E+01
RANGE-COOLING WATER RANGE (F) 2.500000E+01
FLOW RATE IN CONDENSER 4.720735E+07
MAKEUP FLOW RATE TO COOLING TOWER 9.445719E+05

KILOWATT LOSSES

FANS 1.220970E+03
CONDENSATE PUMPS TO COOLING TOWER 1.046766E+03
COOLING WATER PUMPS 2.146313E+03
REINJECTION PUMPS 4.857460E+02
BOOSTER PUMPS TO OVERPRESSURIZE WATER 7.993104E+01
DOWNHOLE PUMPS TO PREVENT TWO PHASE FLOW 0.
BOOSTER PUMPS 2.191199E+02
FEED PUMPS 3.913915E+03

OVERALL RESULTS

GROSS KILOWATT OUTPUT OF THE PLANT 5.500000E+04
GEOTHERMAL FLUID FLOW RATE (LB/HR) 0.057497E+06
(INCLUDING SALT)
FLOW RATE (LR/HR) OF ISOBUTANE 5.887471E+06
COOLING WATER FLOW RATE (LR/HR) 4.728494E+07
TOTAL KILOWATT LOSSES 9.112761E+03
NET KILOWATT OUTPUT 4.588724E+04
NET HEAT RATE (BTU/KW HR) 5.259807E+04

IV. OPTIMIZATION OF THE BINARY FLUID CYCLE

IV-1. Introduction

Geothermal energy has been identified as a potentially large source for electrical power production, and efforts in plant construction have helped in the realization of this potential. It is unfortunate that the conversion process from heat to electricity is limited by the temperature of the resource and the prevailing ambient conditions for heat rejection. This is especially true in geothermal resources where the temperatures are usually below 250°C. For this reason, there are strong economic incentives for optimizing the performance of the conversion processes used for geothermal energy.

Developments in recent years have suggested that the use of working fluids other than water would be more suitable to electrical power consumption from low-temperature heat sources. Even though these alternate working fluids may be superior, the conversion process must be carried out in a manner such that the geothermal resource is utilized as efficiently as possible. In the following paragraphs, a technique to obtain an optimum cycle for a binary power plant is discussed. The optimization technique is described, and the results of optimizing a binary fluid cycle using isobutane are given.

IV-2. Discussion of Theory

The maximum work that can be produced from any heat source is specified by the second law of thermodynamics. The second law states that the efficiency by which work ΔW can be produced by transferring heat ΔQ from a high temperature source at T_h in a Carnot heat engine and rejecting heat to a low temperature sink at T_o is as follows:

$$\eta = \frac{\Delta W}{\Delta Q} = \frac{T_h - T_o}{T_h} \quad (46)$$

More specifically, the maximum work available can be expressed in an integral form as a summation of heat transferred to an infinite number of small Carnot heat engines between the source temperature T_h and the sink temperature T_o . The net maximum work for a geothermal liquid source at temperatures varying between $T_{h,in}$ and $T_{h,out}$ can be expressed as:

$$W_{\max} = - \int_{T_{h,in}}^{T_{h,out}} \left(\frac{T_h - T_o}{T_h} \right) dQ \quad (47)$$

By expressing dQ as a function of enthalpy, the following equation can be written, assuming constant specific heat

$$dQ = dH|_p = C_p dT_h \quad (48)$$

Substituting equation (48) into equation (47);

$$W_{\max} = -\int_{T_{h,in}}^{T_{h,out}} \left(\frac{T_h - T_o}{T_h} \right) C_p dT_h \quad (49)$$

$$\text{or } W_{\max} = -\int_{T_{h,in}}^{T_{h,out}} C_p \left[1 - \frac{T_o}{T_h} \right] dT_h$$

$$\text{or } W_{\max} = -\int_{T_{h,in}}^{T_{h,out}} C_p dT_h + C_p T_o \int_{T_{h,in}}^{T_{h,out}} \frac{dT_h}{T_h}$$

At constant pressure, by the definition of enthalpy and entropy, equation (49) can be expressed as:

$$W_{\max} = - \left[\Delta H - T_o \Delta S \right]_{T_{h,in}}^{T_{h,out}} \quad (50)$$

This quantity represents the maximum reversible work that can be produced, and is commonly referred to as the change in availability, ΔB . (82-84) Knowing $\Delta H = C_p dT_h$ and

$\Delta S = C_p \ln \frac{T_{h,in}}{T_{h,out}}$, equation (50) can be reduced to

$$W_{\max} = C_p (T_{h,in} - T_{h,out}) - T_o C_p \ln \frac{T_{h,in}}{T_{h,out}} \quad (51)$$

The term $W_{\max}$ is expressed in Btu/lb_m. The maximum total work that can be produced by a geothermal reservoir is equal to $Q_{WAT} \times W_{\max}$. The actual work produced by the conversion process is the gross turbine output minus the internal power consumption, of P_{net} . By dividing P_{net} by $(Q_{WAT} \times W_{\max})$ an expression for the efficiency of the

geothermal system can be found, and is identified as the utilization efficiency:

$$\eta_u = \frac{P_{net}}{QWAT \times W_{max}} = \frac{P_{total} - P_{internal\ use}}{QWAT \times C_p (T_{h,in} - T_{h,out}) - T_o \ln \frac{T_{h,in}}{T_{h,out}}} \quad (52)$$

This expression is the basis for the optimization study. Temperatures in this expression are in units of °R. This efficiency should not be confused with the thermodynamic cycle efficiency, η_{cycle} , defined as

$$\eta_{cycle} = \frac{P_{net}}{QWAT \times C_p \times (T_{h,in} - T_{h,out})} \quad (53)$$

The cycle efficiency does not accurately reflect the efficient use of the available energy in a geothermal resource because the geothermal fluid which is reinjected after use in the power plant is not considered as a loss in η_{cycle} . Because of high cost of drilling wells and laying transmission piping to and from the power plant, one cannot afford to disregard the available energy in the rejected geothermal fluid.

The utilization efficiency was used in conjunction with the binary fluid cycle model to determine the optimum cycle for any particular resource temperature, sink temperature, and working fluid.

VI-3. Optimization Results

The utilization efficiency can be employed to establish an optimum cycle by locating the particular values of

turbine inlet pressure, turbine inlet "approach," and heat exchanger pinch point which result in the highest η_u . The values of the inlet geothermal temperature and sink temperature are required to locate these cycle characteristics. This location can be found by operating the computer code to calculate η_u for a range of the values of P_1 , APP (APP = $WT_1 - T_1$), and PINCHP. The results of these iterations can be illustrated on a plot of η_u versus P_1 , with APP as the parametric variable. The following figures -- 11, 12, 13, 15 and 16 -- indicate the results of these computations for the inlet geothermal temperatures of 200, 250, 300, 350, 400 and 500°F.

An example of the effect of the turbine inlet pressure P_1 and the "approach" to the heat exchanger APP is shown in Figure 11. A geothermal fluid temperature of 300°F with a minimum working fluid (isobutane) temperature of 100°F was chosen. Other pertinent parameters include a heat exchanger pinch point of 15°F, a turbine efficiency of 78.5%, and a plant size of 100 MWe. For all five "approach" values, the utilization efficiency increases rapidly with increasing turbine inlet pressure at lower values of P_1 . Each of the "approach" curves were terminated when the state point 1 reached an isentrope of $S = 1.18$ Btu/lb/°F on a pressure versus enthalpy diagram for isobutane. This was done to prevent operation in the two phase region during expansion. For the "approach"

FIGURE 11. Utilization Efficiency versus Turbine Inlet Pressure; Geothermal Temperature = 300°F

values of 60, 40, and 20°F, a maximum utilization efficiency is found around $P_1 = 312$ psia. The optimum cycle for this example exists at an "approach" of 80°F and a turbine inlet pressure of 312 psia, where the utilization efficiency is approximately 27.3%.

In Figure 11, the curve which represents an "approach" of 20°F decreases after reaching a maximum at $P_1 = 312$ psia, but then begins to increase again at around 480 psia. The explanation for this behavior is in the definitions for utilization efficiency and the geothermal flow rate.

$$\eta_u = \frac{P_{net}}{QWAT (\Delta B)} \quad (54)$$

$$QWAT = \frac{Q_{ISO} (H_1 - H_4)}{C_p (WT1 - WT3)} \quad (55)$$

In the pressure region between $P_1 = 480$ and $P_1 = 529$, the enthalpy of the saturated liquid H_4 is rapidly increasing as the saturated liquid curve approaches a zero slope at the critical point. The quantity $(H_1 - H_4)$ of equation (55) is decreasing rapidly in this region, resulting in a decreased geothermal flow rate. This causes an increase in the utilization efficiency.

The example case of geothermal fluid at 300°F indicated an optimum operating point can be selected for a binary cycle. This point establishes the optimum turbine inlet pressure, P_1 , and inlet temperature, $T_1 = WT1 - APP$.

In Figure 12, "approach" curves on a plot of utilization efficiency versus turbine inlet pressure are shown for geothermal temperatures of 200°F and 250°F. These curves are similar to those in Figure 11, showing an optimum cycle. For a geothermal temperature of 200°F, an optimum exists at an "approach" of 40°F and a pressure of 142 psia, where $\eta_u = 10.3\%$. For a geothermal temperature of 250°F, an efficiency of 18.6% is found at APP = 60°F and $P_1 = 212$ psia. The curve with a 20° "approach" does not increase again after reaching a maximum as in Figure 11, because the pressures are far below the critical pressure of 529 psia.

When the geothermal temperature is raised to 350°F, the plot of utilization efficiency versus inlet turbine pressure changes dramatically. Figure 13 displays this plot with the same fixed variables as Figures 11 and 12. None of the five "approach" curves reach a maximum before the critical pressure is reached. However, they do exhibit the same behavior as in Figure 11 where the slopes increase rapidly as the pressure nears the critical point. At 350°F for the geothermal temperature, the first supercritical points are shown. The 20°F "approach" curve reaches a maximum at 862 psia where the $S = 1.18$ Btu/lb/°F isentrope is met. This is the recommended operating point rather than selecting a 60°F "approach" at 492 psia, even though the latter point has a slightly higher utilization

FIGURE 12. Utilization Efficiency versus Turbine Inlet Pressure; Geothermal Temperature = 200, 250°F

FIGURE 13. Utilization Efficiency versus Turbine Inlet Pressure; Geothermal Temperature = 350°F

efficiency. This point is selected for two reasons. The equations of state are inaccurate near the critical point, resulting in questionable computer output. Also, the vaporizing section of the heat exchangers may cease to operate as a boiling heat exchanger if slight fluctuations in the working fluid state exist, i.e., the working fluid may exist at a super-critical state with a slight rise in pressure.

Figure 13 illustrates a key difference between the sub-critical and super-critical cycles. There exists a discontinuity at the critical point for the "approach" curves of 20, 40 and 60°F. The explanation for this discontinuity lies in the location of the heat exchanger pinch point. In the sub-critical cycle, the pinch point (minimum temperature difference between the working fluid and the geothermal fluid) always exists at the point where the working fluid is a saturated liquid. This is indicated in the sub-critical heat exchanger temperature profile of Figure 8. The pinch point exists at the pre-heater exit or vaporizer inlet. As the inlet turbine pressure nears the critical point from below, the pinch point location rapidly moves toward the critical isotherm. Very near the critical point, the slope of the vapor pressure curve is close to zero, and the pinch point approaches the critical isotherm. This is depicted on a P-H (pressure versus enthalpy) diagram for isobutane in Figure 14, where the locus of pinch points are shown for

FIGURE 14. Pressure versus Enthalpy Diagram for Isobutane; Pinch Point Location

increasing working fluid pressure. As soon as the cycle goes super-critical, the pinch point location tends to follow the critical isotherm. In directional terms, it is moving rapidly back in the opposite direction as the sub-critical pinch point, mainly because the slope of the critical isotherm is near zero around the critical point. When Figure 11 was discussed, it was pointed out that η_u increases rapidly as the critical pressure is approached. The opposite effect is exhibited for the super-critical cycle. The geothermal flow rate increases rapidly as the pressure is increased just above the critical point, thus causing a sudden decrease in the utilization efficiency. However, the recovery is rapid, and the utilization efficiency again rises as the pressure P_1 is increased a bit more. This is explained by the rapidly increasing slope of the critical isotherm to the left of the critical point in Figure 14.

Plots of utilization efficiency versus turbine inlet pressure with varying "approach" temperatures are shown in Figures 15 and 16, for inlet geothermal fluid temperatures of 400 and 500°F. At these high temperatures, the logical choice is a super-critical cycle. Both figures show that a maximum utilization efficiency is not reached when the maximum turbine inlet pressure is set at 1500 psia. However, the highest value of η_u is reached at this pressure; therefore, this pressure is the suggested turbine inlet pressure. For the geothermal temperature of 400°F

FIGURE 15. Utilization Efficiency versus Turbine Inlet Pressure; Geothermal Temperature = 400°F

FIGURE 16. Utilization Efficiency Versus Turbine Inlet Pressure; Geothermal Temperature = 500°F

in Figure 15, the recommended operating point is an "approach" of 20°F at 1500 psia, where $\eta_u = 46.4\%$. At a geothermal temperature of 500°F, the recommended operating point is at an "approach" of 40°F and 1500 psia, where $\eta_u = 43.4\%$, as shown in Figure 16. At these high geothermal temperatures, a general rule can be followed for the selection of a proper turbine inlet operating point. The highest allowable turbine inlet pressure and "approach" values should be selected to locate the cycle with the highest utilization efficiency.

The turbine inlet pressure is limited by a maximum, in this case 1500 psia. The "approach" temperature is limited by a phenomenon that cannot be shown on a plot of η_u versus P_1 . The general rule states that the highest "approach" temperature possible should be used. As the "approach" temperature increases, the difference between the inlet geothermal temperature and the maximum working fluid temperature increases by definition. At the same time, the temperature difference between the exhaust geothermal fluid temperature, WT_4 and the inlet working fluid temperature T_3 is decreasing. There is a limit to how small this temperature difference ($WT_4 - T_3$) can be, and when this limit is reached, higher values of "approach" are not practical. This phenomenon dictates the maximum practical value of "approach." For Figure 15, this limit is 20°F; for Figure 16, it is 40°F.

In Figures 15 and 16, the "approach" curves for the geothermal temperatures of 400 and 500°F are similar to those in previous figures, with the exception of their increased slope. The discontinuity near the critical pressure is evident.

Prior to a discussion of the implications of Figures 11-16, some mention of the effect of minimum working fluid temperature T_{2SATL} (related to the sink temperature T_0) and the pinch point is necessary. In Figures 11-16 the values of T_{2SATL} and $PINCHP$ were set at 100°F and 15°F, respectively. Selecting a base curve with $WT1 = 300^\circ\text{F}$, $APP = 40^\circ\text{F}$, and $PINCHP = 15^\circ\text{F}$, the effect of the minimum working fluid temperature on the utilization efficiency and cycle selection is shown in Figure 17. Three different values of T_{2SATL} are shown: 90, 100, and 120°F. As expected, the highest utilization efficiency occurs at the lowest value of T_{2SATL} . At low sink temperatures, more energy per pound of working fluid can be extracted by the turbine due to the higher temperature difference between the source and sink temperatures. In addition, the efficiency maximum gradually shifts to a higher turbine inlet pressure as T_{2SATL} (and the sink temperature) rise; the optimization technique is increasing turbine inlet working fluid enthalpy to compensate for the higher amount of wasted energy at the turbine exhaust. A high utilization

FIGURE 17. Utilization Efficiency Versus Turbine Inlet Pressure; Effect of Sink Temperature Variation

efficiency can be best achieved by using low temperature sinks, and by keeping condenser and cooling tower temperature differences low.

The effect of heat exchanger pinch point PINCHP on utilization efficiency and cycle selection is shown in Figure 18. The same base curve as in Figure 17 was used. As the pinch point increases from 15°F to 25°F, the utilization efficiency decreases rapidly. The optimum pressure selection also decreases gradually at the same time. Operation of heat exchangers at low pinch points is recommended, although a value less than about 15°F causes heat exchangers to become more costly due to small temperature difference.

Figures 11-18 indicate how the values of turbine inlet pressure, turbine inlet temperature, and heat exchanger pinch point should be selected for an optimum utilization efficiency, knowing the inlet geothermal fluid temperature and the prevailing sink temperature. Figures 11, 12, 13, 15 and 16 indicate the optimum turbine inlet temperature and pressure for the geothermal fluid temperatures of 200, 250, 300, 350, 400 and 500°F. Figure 17 indicates the importance of maintaining a low value for the minimum working fluid temperature. Figure 18 illustrates the importance of minimizing the heat exchanger pinch point. If the optimum values for turbine inlet temperature and pressure are plotted on a pressure versus

FIGURE 18. Utilization Efficiency versus Turbine Inlet Pressure; Effect of Pinch Point Variation

enthalpy diagram of isobutane, the locus of optimum turbine inlet points can be seen. With selected values for turbine and pump efficiency, the entire Rankine cycle is fixed by the determination of turbine inlet conditions. The locus of optimum turbine inlet operating points are shown in Figure 19.

Using a PINCHP of 15°F , $T_{2\text{SATL}} = 100^{\circ}\text{F}$, $\eta_{\text{turbine}} = .785$, and isobutane as a working fluid, these optimum points are shown for each geothermal temperature considered on a P-H diagram. All optimums lie on the isentrope $S = 1.18 \text{ Btu/lb/}^{\circ}\text{F}$ except those cases where the maximum pressure limit was reached. This isentrope is also the limit for operating in the superheated region. The operating point for state 1 (turbine inlet) cannot shift any further to the left in Figure 19 and still prevent two-phase flow during turbine expansion. For those cases that reach the maximum pressure limit (400°F and 500°F), the optimum point is at 1500 psia and the maximum "approach" that satisfies the minimum geothermal fluid exhaust temperature criterion discussed with Figures 15 and 16.

The summary in Figure 19 indicates that a binary fluid cycle for a geothermal system should contain as little superheat for the working fluid as possible. The geothermal flow rate tends to increase as working fluid superheat increases, because of the extra energy to be

FIGURE 19. Pressure versus Enthalpy Diagram
for Isobutane; Cycle Optimums

added between states 3 and 1 (compressor outlet to turbine inlet). Also, as superheat increases at a constant turbine inlet pressure, the turbine operates at a slightly higher working fluid enthalpy difference due to the decreasing slope of the isentropes. However, this increase in extracted energy of the turbine does not compensate for the increase in geothermal flow rate. The available energy in the geothermal fluid must be used conservatively, and a minimum of working fluid superheat accomplishes this.

IV-4. Optimization Subroutine

The optimization subroutine OPBIN discussed in section III is based upon the results of sub-section IV-3. The subroutine OPBIN selects an operating point on the isentrope $S = 1.18 \text{ Btu/lb/}^\circ\text{F}$ which results in a maximum utilization efficiency, within established internal boundaries. These boundaries are

- (1) maximum turbine inlet pressure = 1500 psia
- (2) minimum possible heat exchanger approach = 20°F
- (3) minimum exit temperature of the geothermal fluid = $T_3 + 25^\circ\text{F}$ (T_3 is the working fluid temperature at the inlet to the heat exchangers)
- (4) maximum "approach" = 100°F

The optimum value is reached by slowly decreasing "approach" from a maximum of 100°F until a maximum value

of utilization efficiency is reached. The selection of a turbine inlet pressure within 20 psia of the critical pressure is avoided to prevent heat exchanger operation near the critical point. If the subroutine passes the critical pressure while attempting to locate a maximum η_u , and "approach" of 20°F is selected, and the operating point is located at $T1 = WT1 - 20^\circ\text{F}$ and $S = 1.18 \text{ Btu/lb/}^\circ\text{F}$. If this point results in a pressure $P1$ exceeding the maximum allowable pressure, the operating point is selected at $P1 = 1500 \text{ psia}$ and the minimum value of $T1$, limited by boundary condition(3).

Input to subroutine OPBIN is as follows:

- (a) geothermal fluid temperature at inlet, $WT1$
- (b) minimum working fluid temperature, $T2SATL$

Output of this subroutine is as follows:

- (a) turbine inlet pressure, $P1$
- (b) heat exchanger "approach", APP
- (c) turbine inlet temperature, $T1$
- (d) heat exchanger pinch point, $PINCHP$

Subroutine OPBIN provides the necessary information to subroutine BUTANE for completion of the binary fluid cycle design. This optimization scheme is currently applicable to the working fluid isobutane only.

V. SUMMARY AND CONCLUSIONS

Currently, 380,000 MW of electricity are produced in the United States. Of this total, only 396 MWe are generated from geothermal sources. The prospects for wide-scale use of geothermal resources are however quite promising as evidenced by the existing power plants around the world, the proposed power plants, and the presence of numerous geothermal reservoirs. The utilization of the large number of low temperature reservoirs with the binary fluid cycle plant will significantly increase the power generation capabilities of the United States and other nations.

A computer model which describes a geothermal power plant using the binary fluid cycle has been developed. This model can be used to assess the potential of binary cycle power plants, and to assist in designing power plants for specific geothermal sites. The computer model provides information on the major power plant components, fluid properties throughout the power plant, type of power cycle used, expected net power output, and plant performance and design.

The computer solutions have indicated that a binary fluid cycle can be designed to generate electricity from a geothermal reservoir containing water at a wide range of temperatures. A sample case with a geothermal fluid temperature of 356°F is shown in Table 6. Using isobutane

in a sub-critical cycle, this sample case indicates that 45,890 kilowatts of net power can be generated with a geothermal flow rate of 8,000,000 lb/hr. This quantity of power is determined by subtracting internal power requirements from the gross output of a 55,000 kWe turbine-generator. The utilization efficiency for this plant is 32%.

The computer model has been exercised over a wide range of design conditions. Power plant sizes from 10 to 220 MWe have been considered for geothermal source temperatures ranging from 200 to 600°F. All the available working fluids have been used to generate geothermal power plant information. Varying cycle designs and heat exchanger configurations have been considered during the operation of this computer code. When data have been available, comparison of the computer program output with published plant designs has resulted in good agreement.

This computer model is useful for identifying the potential of the binary fluid cycle in a geothermal power plant. Due to the relatively low energy state of the available geothermal fluid, it is important that this computer program generates a power plant design which efficiently converts the geothermal energy to electricity. A technique has been developed which maximizes the use of available energy contained in the geothermal fluid. This technique has been used with the computer program to

generate an optimum plant design on the basis of maximizing the following quantity called the utilization efficiency:

$$\eta_u = \frac{P_{net}}{Q_{WAT} \times C_p \times [WT1 - T_0 - T_0 \ln(WT1/T_0)]}$$

The utilization efficiency is based upon the second law of thermodynamics, which specifies the maximum work that can be produced from a heat source. The utilization efficiency should not be confused with the thermodynamic cycle efficiency, η_{cycle} , defined as

$$\eta_{cycle} = \frac{P_{net}}{Q_{WAT} \times C_p \times (WT1 - T_0)}$$

The thermodynamic cycle efficiency does not accurately reflect the efficient use of the geothermal resource because the geothermal fluid which is reinjected after use in the power plant is not considered as a loss in η_{cycle} . This loss must be considered due to the large cost of providing the geothermal fluid and the relatively low energy state of that fluid. The utilization efficiency accurately reflects this loss.

Using the utilization efficiency as the basis for maximizing the conversion of energy from the geothermal fluid, the computer program was used to determine optimum binary fluid cycles for a variety of geothermal resource conditions. Two quantities were required to perform this optimization: the inlet geothermal fluid temperature and the prevailing minimum temperature for heat rejection.

The minimum heat rejection temperature dictates the lowest working fluid temperature and the turbine exhaust pressure for a particular working fluid. The utilization efficiency was used to provide the following information on the optimum cycle.

- working fluid pressure at the turbine inlet
- working fluid temperature at the turbine inlet
- pinch point temperature difference of the heat exchanger

These computed values dictate the binary cycle selected.

Optimized cycles were found for inlet geothermal fluid temperatures ranging from 200 to 500°F using the utilization efficiency. Isobutane was the working fluid used in these computations, mainly because of its widespread use in geothermal plant design. The following conclusions can be made concerning these optimum cycle selections:

- (a) The utilization efficiency increases with increasing geothermal fluid temperature. At a geothermal temperature of 200°F, the utilization efficiency of an optimized cycle is 10.3%. The highest utilization efficiency for a power plant using geothermal fluid entering at 400°F is 46.4%. The rise in utilization efficiency is due to the increasing temperature of the geothermal fluid.

- (b) The turbine inlet pressure and temperature computed for an optimized cycle both rise as the inlet geothermal fluid temperature increases.

The following results indicate these trends.

<u>Inlet Geothermal Fluid Temperature</u>	<u>Selected Turbine Inlet Pressure</u>	<u>Selected Turbine Inlet Temperature</u>
200°F	142 psia	160°F
250	212	190
300	312	220
350	862	290
400	1500	380
500	1500	460

At the higher geothermal fluid temperatures, the turbine inlet pressure was limited to 1500 psia.

- (c) Below an inlet geothermal fluid temperature of 350°F, the most efficient cycle is of a sub-critical type, i.e., the turbine inlet pressure selected is below the critical pressure of 529 psia. At inlet geothermal fluid temperatures of 350°F or above, the super-critical cycle is recommended when using isobutane as a working fluid.

- (d) The maximization of utilization efficiency for a wide range of inlet geothermal fluid temperatures has indicated a general rule concerning the recommended working fluid state at the turbine inlet. The working fluid should contain a minimum of superheat at the turbine

inlet. In other words, it is not advantageous to add sufficient energy to the working fluid to create a highly superheated vapor before expansion through the turbine. Even though a highly superheated working fluid results in a more efficient extraction of energy in the turbine, this increase in energy extraction does not compensate for the increased geothermal fluid flow rate required to raise the working fluid to that high energy state. It is costly to provide geothermal fluid to the power plant; therefore it should be used as conservatively as possible.

This general rule results in the state of the working fluid at the turbine inlet always being fixed at a point slightly to the right of the saturated vapor line on an isobutane P-H diagram, with a small amount of superheat. This rule applies to all inlet geothermal fluids considered.

- (e) The results of cycle optimization with the utilization efficiency indicate the best pinch point temperature difference to use in the geothermal fluid/working fluid heat exchangers. The pinch point temperature difference should

be as low as possible without adversely affecting the design of the heat exchanger surfaces. If this pinch point is too low (below 15°F) the surface area required for heat exchange will become excessively high. A pinch point temperature difference of 15°F is recommended.

- (f) The prevailing minimum temperature for heat rejection, or the sink temperature, is of course fixed for a particular power plant. However, results of the computer code optimization indicate that the difference between the sink temperature and the minimum working fluid temperature should be as small as possible. The closer the minimum working fluid temperature is to the sink temperature, the higher the utilization efficiency will be. At low values for the minimum working fluid temperature, the energy per pound of working fluid that can be extracted by the turbine is near its maximum.

If the guidelines suggested in the conclusions of (a) through (f) are followed, the optimum binary fluid cycle can be obtained for a wide range of geothermal fluid temperatures, with isobutane as the selected working fluid. Although extensive investigations on the optimization of other working fluids have not been performed, the guidelines established in this section are most likely still applicable.

The overall computer code of the binary cycle power plant for geothermal applications can be used to obtain plant design information under optimized conditions. The code is applicable to a wide range of input data on geothermal resources. With minor additions the computer model is applicable to more complex types of binary fluid cycle designs and to plant optimizations using a variety of working fluids.

NOMENCLATURE

a	Freon, NH_3 , or hydrocarbon constant
A	Freon or NH_3 constant
A_L	Freon or NH_3 constant
A_0	hydrocarbon constant
A_{00}	hydrocarbon constant
A_2	Freon or NH_3 constant
A_3	Freon or NH_3 constant
A_4	Freon or NH_3 constant
A_5	Freon or NH_3 constant
A_6	Freon or NH_3 constant
A_{10}	hydrocarbon constant
A_{20}	hydrocarbon constant
A_{30}	hydrocarbon constant
A_{40}	hydrocarbon constant
APP	heat exchanger "approach," °F
b	Freon, NH_3 , or hydrocarbon constant
B	barometric pressure, in. Hg.
B	Freon or NH_3 constant
B_L	Freon or NH_3 constant
B_0	hydrocarbon constant
B_{00}	hydrocarbon constant
B_2	Freon or NH_3 constant
B_3	Freon or NH_3 constant
B_4	Freon or NH_3 constant

B_5	Freon or NH_3 constant
B_6	Freon or NH_3 constant
B_{10}	hydrocarbon constant
B_{20}	hydrocarbon constant
B_{30}	hydrocarbon constant
B_{40}	hydrocarbon constant
c	Freon, NH_3 , or hydrocarbon constant
C	Freon or NH_3 constant
C'	Freon or NH_3 constant
C_L	Freon or NH_3 constant
C_p	specific heat at constant pressure, Btu/lbm/°F
C_0	hydrocarbon constant
C_2	Freon or NH_3 constant
C_3	Freon or NH_3 constant
C_4	Freon or NH_3 constant
C_5	Freon or NH_3 constant
C_6	Freon or NH_3 constant
CWT1	inlet cooling water temperature, °F
CWT2	outlet cooling water temperature, °F
CWVEL	cooling water velocity in condenser, ft/hr
d	Freon, NH_3 , or hydrocarbon constant
D	Freon or NH_3 constant
D_e	hydraulic diameter, ft
D_i	inside tube diameter, ft
D_o	outside tube diameter, ft
D_L	Freon or NH_3 constant

D_0	hydrocarbon constant
DBAT	dry bulb air temperature, °F
DD1	iterative density value, lb-mole/ft ³
DD2	iterative density value, lb-mole/ft ³
DIAL	inside tube diameter in preheater, ft
DIAOL	outside tube diameter in preheater, ft
DIAV	inside tube diameter in vaporizer, ft
DIAOV	outside tube diameter in vaporizer, ft
DIAS	inside tube diameter in superheater, ft
DIAOS	outside tube diameter in superheater, ft
DIASC	inside tube diameter in super-critical heat exchanger, ft
DIAOSC	outside tube diameter in super-critical heat exchanger, ft
DIAC	inside tube diameter in condenser, ft
DIAOC	outside tube diameter in condenser, ft
DSAT	saturated liquid density, lbm/ft ³
D1	density at state 1, lb-mole/ft ³
D2	density at state 2, lb-mole/ft ³
D2EST	density at state 2EST, lb-mole/ft ³
D2SATL	density at state 2SATL, lb-mole/ft ³
D4	density at state 4, lb-mole/ft ³
D5	density at state 5, lb-mole/ft ³
E	Freon or NH ₃ constant
E_0	hydrocarbon constant
E_L	Freon or NH ₃ constant
f	Freon or NH ₃ constant

f	fugacity
F	Freon or NH_3 constant
FOULFI	inside fouling factor, $\text{hr} \cdot \text{ft}^2 \cdot ^\circ\text{F}/\text{Btu}$
FOULFO	outside fouling factor, $\text{hr} \cdot \text{ft}^2 \cdot ^\circ\text{F}/\text{Btu}$
GRKW	gross kilowatt power plant output, KWe
h_i	inside heat transfer coefficient, $\text{Btu}/\text{hr}/\text{ft}^2/^\circ\text{F}$
h_o	outside heat transfer coefficient, $\text{Btu}/\text{hr}/\text{ft}^2/^\circ\text{F}$
H_1	enthalpy at state 1, Btu/lbm
H2EST	enthalpy at state 2EST, Btu/lbm
H2SATL	enthalpy at state 2SATL, Btu/lbm
H2SATV	enthalpy at state 2SATV, Btu/lbm
H2	enthalpy at state 2, Btu/lbm
H3	enthalpy at state 3, Btu/lbm
H4	enthalpy at state 4, Btu/lbm
H5	enthalpy at state 5, Btu/lbm
HH1	iterative enthalpy value, Btu/lbm
HH3	iterative enthalpy value, Btu/lbm
J	conversion factor from work to heat units, = 0.185053

$$\frac{\text{in}^2}{\text{ft}^2} \frac{\text{Btu}}{\text{ft} \cdot \text{lb}_f}$$

k	ratio of specific heats
k_{wall}	thermal conductivity, $\text{Btu}/\text{hr}/\text{ft}/^\circ\text{F}$
K	Freon, NH_3 , or hydrocarbon constant
LMTD	log mean temperature difference, $^\circ\text{F}$
$\dot{m}$	mass flow rate, lbm/hr

NFLUID	type of working fluid
OEFF	overall turbine/generator efficiency, %/100
P	pressure, psia
P_r	Prandtl number
P_{net}	net power output, Btu/hr
PVEL	velocity in super-critical heat exchanger tubes, ft/hr
PINCHP	heat exchanger pinch point, °F
P1	working fluid pressure at state 1, psia
P2	working fluid pressure at state 2, psia
P3	working fluid pressure at state 3, psia
P4	working fluid pressure at state 4, psia
P5	working fluid pressure at state 5, psia
Q_T	heat load, Btu/hr
QCWAT	cooling water flow rate, lb/hr
QISO	working fluid flow rate, lb/hr
QWAT	geothermal fluid flow rate, lb/hr
R	gas constant
Re	Reynolds number
REFF	Rankine cycle efficiency, %/100
RI	relative humidity, %/100
S	entropy, Btu/lbm/°R
SVSATV	saturated vapor specific volume, ft ³ /lbm
SDPDT	derivative of pressure w.r.t. temperature
SDHVAP	heat of vaporization, Btu/lbm
S1	entropy at state 1, Btu/lbm/°R

S2EST	entropy at state 2EST, Btu/lbm/°R
S4	entropy at state 4, Btu/lbm/°R
S5	entropy at state 5, Btu/lbm/°R
T	temperature, °F
T _c	critical temperature, °R
T _h	heat source temperature, °F
T ₀	heat sink temperature, °F
TTD	condenser terminal temperature difference, °F
TT1	iterative temperature value, °R
TT3	iterative temperature value, °R
T1	temperature at state 1, °R
T2	temperature at state 2, °R
T2EST	temperature at state 2EST, °R
T2SATL	temperature at state 2SATL, °R
T3	temperature at state 3, °R
T4	temperature at state 4, °R
T5	temperature at state 5, °R
U	overall heat transfer coefficient, Btu/hr/ft ² /°F
V	specific volume, ft ³ /lbm
V	velocity, ft/hr
VV1	iterative specific volume value, ft ³ /lbm
VV3	iterative specific volume value, ft ³ /lbm
V1	specific volume at state 1, ft ³ /lbm
V2EST	specific volume at state 2EST, ft ³ /lbm
V2	specific volume at state 2, ft ³ /lbm
V4	specific volume at state 4, ft ³ /lbm

V5	Specific volume at state 5, ft^3/lbm
W_{max}	maximum work
WT1	working fluid temperature at point 1, $^{\circ}\text{F}$
WT2	working fluid temperature at point 2, $^{\circ}\text{F}$
WT3	working fluid temperature at point 3, $^{\circ}\text{F}$
WT4	working fluid temperature at point 4, $^{\circ}\text{F}$
WP1	working fluid pressure at point 1, psia
WP2	working fluid pressure at point 2, psia
WP3	working fluid pressure at point 3, psia
WP4	working fluid pressure at point 4, psia
WBAT	wet bulb air temperature, $^{\circ}\text{F}$
WTT1	iterative working fluid temperature value, $^{\circ}\text{F}$
WTT4	iterative working fluid temperature value, $^{\circ}\text{F}$
WVELL	flow velocity in preheater tubes, ft/hr
WVELV	flow velocity in vaporizer tubes, ft/hr
WVELS	flow velocity in superheater tubes, ft/hr
X	reference enthalpy, Btu/lbm
Y	reference entropy, $\text{Btu/lbm}/^{\circ}\text{F}$
Z	constant, $= 0.185053 \text{ in}^2/\text{ft}^2 \cdot \text{Btu}/\text{ft} \cdot \text{lb}_f$
α	Freon, NH_3 , or hydrocarbon constant
γ	hydrocarbon constant
ΔB	net availability
ΔH	net enthalpy, Btu/lbm
ΔQ	net heat
ΔW	net work
η_c	Carnot efficiency

η_{cycle}	cycle efficiency
η_{pump}	pump efficiency
η_{Rankine}	Rankine cycle efficiency
η_u	utilization efficiency
μ	viscosity, lbm/ft/hr
ρ	density, lbm/ft ³
ϕ	conversion factor, = $313.611 \frac{\text{psia cu.ft}}{\text{lb-mole}} \cdot \frac{\text{lb}}{\text{Btu}}$
1	state at turbine inlet (see Figure 6 for state points)
2EST	state at ideal turbine exhaust
2	state at actual turbine exhaust
2SATV	state at saturated vapor (condenser)
2SATL	state at saturated liquid (condenser)
3	state at compressor outlet
4	state at saturated liquid (heat exchanger)
5	state at saturated vapor (heat exchanger)

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APPENDIX

COEFFICIENTS FOR LIGHT HYDROCARBON,
FREON, AND AMMONIA EQUATIONS

APPENDIX

Coefficients for Light Hydrocarbon Equations

	<u>Isobutane</u>	<u>N-Butane</u>
<u>Coefficients for Equations (10), (11), (13), & (15)</u>		
A_0	3.7264×10^4	3.25447×10^4
B_0	1.8789	1.56588
C_0	1.01413×10^{10}	1.37436×10^{10}
D_0	8.53176×10^{11}	3.33159×10^{11}
E_0	8.4086×10^{13}	2.30902×10^{12}
a	4.7991×10^4	7.11818×10^4
b	8.58663	9.14066
c	4.06763×10^{10}	7.00044×10^{10}
d	2.16863×10^7	3.64238×10^7
α	4.23987	4.00985
γ	7.11486	7.54122
R	10.7335	10.7335

Coefficients for Equation (12)

A_{00}	-776.11	-739.77
A_{10}	0.06872	0.22389
A_{20}	3.071×10^{-4}	-0.601×10^{-4}
A_{30}	0.0	3.93×10^{-7}
A_{40}	0.0	-1.594×10^{-10}

Coefficients for
Equation (14)IsobutaneN-Butane B_{00}

0.86401

0.9008

 B_{10} 9.079×10^{-4} 10.060×10^{-4} B_{20} -2.073×10^{-7} -3.006×10^{-7} B_{30} 0.885×10^{-10} 0.846×10^{-10} B_{40} -0.241×10^{-13}

0.0

Coefficients for Freon and Ammonia Equations

Coefficients for Equations (6), (8), & (9)	<u>R-11</u>	<u>R-12</u>	<u>R-21</u>	<u>R-22</u>	<u>R-113</u>	<u>R-114</u>	<u>NH₃</u>
R	.078117	.088734	.10427	.124098	.05728	.06278	.63006
b	.0019	.006509	0.0	.002	0.0	.005915	.001247
A ₂	-3.1268	-3.4097	-7.316	-4.3535	-4.035	-2.3857	-96.1215
B ₂	1.3185×10^{-3}	.001594	.004642	.00241	.00262	.00108	.0711
C ₂	-35.77	-56.7628	0.0	-44.0669	0.0	-6.5644	-2.5418
A ₃	-0.02534	.06024	-.20382	-.01746	-.0214	.03406	11.04125
B ₃	4.8751×10^{-5}	-1.8796×10^{-5}	.000359	7.6279×10^{-5}	0.5×10^{-4}	-5.3336×10^{-6}	-.011
C ₃	1.22037	1.3114	0.0	1.48376	0.0	.16367	-5.2232
A ₄	1.6873×10^{-3}	-5.4874×10^{-4}	0.0	.00231	0.0	-3.8575×10^{-4}	-0.8923
B ₄	-1.8051×10^{-6}	0.0	0.0	-3.6057×10^{-6}	0.0	0.0	9.7686×10^{-4}
C ₄	0.0	0.0	0.0	0.0	0.0	0.0	93.744
A ₅	-2.359×10^{-5}	0.0	0.0	-3.724×10^{-5}	0.0	1.6018×10^{-6}	.02435
B ₅	2.4483×10^{-8}	3.4688×10^{-9}	0.0	5.355×10^{-8}	0.0	6.2632×10^{-10}	-2.691×10^{-5}
C ₅	-1.4784×10^{-4}	-2.5439×10^{-5}	0.0	-1.845×10^{-4}	0.0	-1.0165×10^{-5}	-3.1814
A ₆	1.0575×10^8	0.0	0.0	1.3634×10^8	0.0	0.0	0.0
B ₆	-9.472×10^{-4}	0.0	0.0	-1.6726×10^5	0.0	0.0	0.0
C ₆	0.0	0.0	0.0	0.0	0.0	0.0	0.0

Equations (6), R-11
(8) & (9)
(Continued)

ctions (6), R-11 & (9) tinued)	<u>R-12</u>	<u>R-21</u>	<u>R-22</u>	<u>R-113</u>	<u>R-114</u>	<u>NH₃</u>	
K	4.5	5.475	0.0	4.2	0.0	3.0	6.424
α	580.0	0.0	0.0	548.0	0.0	0.0	210.691
C	0.0	0.0	0.0	0.0	0.0	0.0	0.0
X	50.5418	39.5566	76.27	62.4	25.198	25.34	195.933
Y	-0.0918	-.01654	-.11057	-.0453	-.4055	-.1151	-1.4

Coefficients
for Equations
(7) and (26)

A	42.147	39.8838	42.7908	29.3575	33.0655	27.0713	25.5744
B	-4344.344	-3436.6322	-4261.34	-3845.193	-4330.98	-5113.702	-3295.125
C	-12.8460	-12.4715	-13.0295	-7.8610	-9.2635	-6.3087	-6.4012
D	4.0084×10^{-3}	4.7305×10^{-3}	3.985×10^{-3}	2.1909×10^{-3}	2.654×10^{-3}	6.913×10^{-4}	-4.1483×10^{-4}
E	.03136	0.0	0.0	.44574	0.0	.78142	1.476×10^{-6}
F	862.07	0.0	0.0	686.1	0.0	768.35	0.0

Coefficients
for Equations
in Table IV

	<u>R-11</u>	<u>R-12</u>	<u>R-21</u>	<u>R-22</u>	<u>R-113</u>	<u>R-114</u>	<u>NH₃</u>
AL	34.57	34.84	116.38	32.76	122.872	36.321	14.7
BL	57.638	.02696	-.03107	54.6344	-.0128	.02178	22.7685
CL	43.6322	.8349	-.00005	36.749	-.0000636	.6365	28.9739
DL	-42.8236	6.0268	0.0	-22.2926	0.0	6.7183	-23.456
EL	36.7066	-0.6555	0.0	20.4733	0.0	1.97×10^{-6}	15.1727

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