AN ABSTRACT OF THE THESIS OF

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Title MODELING OF GEOTHERMAL POWER PLANTS USING THE BINARY FLUID CYCLE

Abstract approved

In light of the current world-wide energy situation, the utilization of the immense reserves of geothermal energy is being actively pursued. The thermal energy contained in a geothermal reservoir is mainly stored in rocks, but is also held in liquid water and/or steam. The generation of electrical power from these geothermal reserves is currently limited. The majority of operating geothermal power plants either use available dry steam or steam flashed from hot water. If this resource is to make a significant impact on the world energy market, successful methods must be found to utilize the larger quantity of lower temperature and high salinity reservoirs. One of the most promising methods proposed to date for efficient use of these reservoirs is the binary fluid cycle. The topic of this thesis is the formulation of a computer model of a geothermal power plant using the binary fluid cycle.

In the binary fluid cycle plant, geothermal fluid is extracted from a geothermal reservoir and transmitted to
the power plant in a liquid state. At the power plant, the geothermal fluid is passed through a series of heat exchangers where thermal energy is transferred to a working, or power fluid such as a Freon or a light hydrocarbon. This working fluid is brought to a superheated state and then expanded through a turbine which is coupled to a generator. After expansion, the working fluid is condensed in a surface condenser supplied with cooling water from a cooling tower. This fluid is subsequently repressurized and returned to the heat exchangers. The expended geothermal fluid is returned to the underground reservoir.

The objective of this thesis was to develop a computer program which effectively simulates the binary fluid cycle power plant and all its components. The geothermal power plant model includes the following:

- determination of the size and nature of power plant components
- determination of geothermal fluid flow rates, working fluid flow rates, and heat rejection requirements for a variety of geothermal source temperatures, pressures and water conditions, and plant sizes
- calculation of geothermal fluid and working fluid thermodynamic and physical properties
- thermodynamic cycle calculations
- overall design performance with the ability to perform sensitivity analyses on various operating parameters.
After the completion of this computer model, a particular working fluid, isobutane, was selected for an in-depth study of the binary fluid cycle power plant. The optimum power plant designs for a range of available geothermal fluid temperatures were found based on economics and efficient plant performance.
Modeling of Geothermal Power Plants
Using the Binary Fluid Cycle

by
Ronald A. Walter

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April 5, 1976

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NOMENCLATURE

a  Freon, NH\textsubscript{3}, or hydrocarbon constant
A  Freon or NH\textsubscript{3} constant
A\textsubscript{L} Freon or NH\textsubscript{3} constant
A\textsubscript{0} hydrocarbon constant
A\textsubscript{00} hydrocarbon constant
A\textsubscript{2} Freon or NH\textsubscript{3} constant
A\textsubscript{3} Freon or NH\textsubscript{3} constant
A\textsubscript{4} Freon or NH\textsubscript{3} constant
A\textsubscript{5} Freon or NH\textsubscript{3} constant
A\textsubscript{6} Freon or NH\textsubscript{3} constant
A\textsubscript{10} hydrocarbon constant
A\textsubscript{20} hydrocarbon constant
A\textsubscript{30} hydrocarbon constant
A\textsubscript{40} hydrocarbon constant
APP heat exchanger "approach," °F
b  Freon, NH\textsubscript{3}, or hydrocarbon constant
B  barometric pressure, in. Hg.
B  Freon or NH\textsubscript{3} constant
B\textsubscript{L} Freon or NH\textsubscript{3} constant
B\textsubscript{0} hydrocarbon constant
B\textsubscript{00} hydrocarbon constant
B\textsubscript{2} Freon or NH\textsubscript{3} constant
B\textsubscript{3} Freon or NH\textsubscript{3} constant
B\textsubscript{4} Freon or NH\textsubscript{3} constant
<table>
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<tr>
<th>Code</th>
<th>Description</th>
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<td>Freon or NH\textsubscript{3} constant</td>
</tr>
<tr>
<td>B_6</td>
<td>Freon or NH\textsubscript{3} constant</td>
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<td>B_{10}</td>
<td>hydrocarbon constant</td>
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<td>B_{20}</td>
<td>hydrocarbon constant</td>
</tr>
<tr>
<td>B_{30}</td>
<td>hydrocarbon constant</td>
</tr>
<tr>
<td>B_{40}</td>
<td>hydrocarbon constant</td>
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<tr>
<td>c</td>
<td>Freon, NH\textsubscript{3}, or hydrocarbon constant</td>
</tr>
<tr>
<td>C</td>
<td>Freon or NH\textsubscript{3} constant</td>
</tr>
<tr>
<td>C'</td>
<td>Freon or NH\textsubscript{3} constant</td>
</tr>
<tr>
<td>C_L</td>
<td>Freon or NH\textsubscript{3} constant</td>
</tr>
<tr>
<td>C_P</td>
<td>specific heat at constant pressure, Btu/lbm/°F</td>
</tr>
<tr>
<td>C_0</td>
<td>hydrocarbon constant</td>
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<td>C_2</td>
<td>Freon or NH\textsubscript{3} constant</td>
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<td>C_3</td>
<td>Freon or NH\textsubscript{3} constant</td>
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<td>C_4</td>
<td>Freon or NH\textsubscript{3} constant</td>
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<tr>
<td>C_5</td>
<td>Freon or NH\textsubscript{3} constant</td>
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<td>C_6</td>
<td>Freon or NH\textsubscript{3} constant</td>
</tr>
<tr>
<td>CWT_1</td>
<td>inlet cooling water temperature, °F</td>
</tr>
<tr>
<td>CWT_2</td>
<td>outlet cooling water temperature, °F</td>
</tr>
<tr>
<td>CWVEL</td>
<td>cooling water velocity in condenser, ft/hr</td>
</tr>
<tr>
<td>d</td>
<td>Freon, NH\textsubscript{3}, or hydrocarbon constant</td>
</tr>
<tr>
<td>D</td>
<td>Freon or NH\textsubscript{3} constant</td>
</tr>
<tr>
<td>D_e</td>
<td>hydraulic diameter, ft</td>
</tr>
<tr>
<td>D_i</td>
<td>inside tube diameter, ft</td>
</tr>
<tr>
<td>D_o</td>
<td>outside tube diameter, ft</td>
</tr>
<tr>
<td>D_L</td>
<td>Freon or NH\textsubscript{3} constant</td>
</tr>
</tbody>
</table>
\( D_0 \) hydrocarbon constant
\( DBAT \) dry bulb air temperature, \(^\circ\)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 \( \text{NH}_3 \) constant
\( E_0 \) hydrocarbon constant
\( E_L \) Freon or \( \text{NH}_3 \) constant
\( f \) Freon or \( \text{NH}_3 \) constant
f  fugacity
F  Freon or NH₃ constant
FOULFI inside fouling factor, hr·ft²·°F/Btu
FOULFO outside fouling factor, hr·ft²·°F/Btu
GRKW gross kilowatt power plant output, KWe
hᵢ  inside heat transfer coefficient, Btu/hr/ft²/°F
hₒ  outside heat transfer coefficient, Btu/hr/ft²/°F
H₁  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
   \[
   \text{in}^2 \text{Btu} / \text{ft}^2 \text{ft-lb}_f
   \]
k  ratio of specific heats
kₜₙₜₜ thermal conductivity, Btu/hr/ft/°F
K  Freon, NH₃, or hydrocarbon constant
LMTD log mean temperature difference, °F
ṁ  mass flow rate, lbm/hr
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>NFLUID</td>
<td>type of working fluid</td>
</tr>
<tr>
<td>OEFF</td>
<td>overall turbine/generator efficiency, %/100</td>
</tr>
<tr>
<td>P</td>
<td>pressure, psia</td>
</tr>
<tr>
<td>Pr</td>
<td>Prandtl number</td>
</tr>
<tr>
<td>P_{net}</td>
<td>net power output, Btu/hr</td>
</tr>
<tr>
<td>PVEL</td>
<td>velocity in super-critical heat exchanger tubes, ft/hr</td>
</tr>
<tr>
<td>PINCHP</td>
<td>heat exchanger pinch point, °F</td>
</tr>
<tr>
<td>P1</td>
<td>working fluid pressure at state 1, psia</td>
</tr>
<tr>
<td>P2</td>
<td>working fluid pressure at state 2, psia</td>
</tr>
<tr>
<td>P3</td>
<td>working fluid pressure at state 3, psia</td>
</tr>
<tr>
<td>P4</td>
<td>working fluid pressure at state 4, psia</td>
</tr>
<tr>
<td>P5</td>
<td>working fluid pressure at state 5, psia</td>
</tr>
<tr>
<td>QT</td>
<td>heat load, Btu/hr</td>
</tr>
<tr>
<td>QCWAT</td>
<td>cooling water flow rate, lb/hr</td>
</tr>
<tr>
<td>QISO</td>
<td>working fluid flow rate, lb/hr</td>
</tr>
<tr>
<td>QWAT</td>
<td>geothermal fluid flow rate, lb/hr</td>
</tr>
<tr>
<td>R</td>
<td>gas constant</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>REFF</td>
<td>Rankine cycle efficiency, %/100</td>
</tr>
<tr>
<td>RI</td>
<td>relative humidity, %/100</td>
</tr>
<tr>
<td>S</td>
<td>entropy, Btu/lbm/°R</td>
</tr>
<tr>
<td>SVSATV</td>
<td>saturated vapor specific volume, ft³/lbm</td>
</tr>
<tr>
<td>SDPDT</td>
<td>derivative of pressure w.r.t. temperature</td>
</tr>
<tr>
<td>SDHVAP</td>
<td>heat of vaporization, Btu/lbm</td>
</tr>
<tr>
<td>S1</td>
<td>entropy at state 1, Btu/lbm/°R</td>
</tr>
</tbody>
</table>
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
Tc     critical temperature, °R
Th     heat source temperature, °F
T0     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/ft²/°F
V      specific volume, ft³/lbm
V      velocity, ft/hr
VV1    iterative specific volume value, ft³/lbm
VV3    iterative specific volume value, ft³/lbm
V1     specific volume at state 1, ft³/lbm
V2EST  specific volume at state 2EST, ft³/lbm
V2     specific volume at state 2, ft³/lbm
V4     specific volume at state 4, ft³/lbm
<table>
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<th>Symbol</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>V5</td>
<td>Specific volume at state 5, ft$^3$/lbm</td>
</tr>
<tr>
<td>$W_{max}$</td>
<td>maximum work</td>
</tr>
<tr>
<td>WT1</td>
<td>Working fluid temperature at point 1, °F</td>
</tr>
<tr>
<td>WT2</td>
<td>Working fluid temperature at point 2, °F</td>
</tr>
<tr>
<td>WT3</td>
<td>Working fluid temperature at point 3, °F</td>
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<tr>
<td>WT4</td>
<td>Working fluid temperature at point 4, °F</td>
</tr>
<tr>
<td>WP1</td>
<td>Working fluid pressure at point 1, psia</td>
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<td>WP2</td>
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<td>WP4</td>
<td>Working fluid pressure at point 4, psia</td>
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<td>WBAT</td>
<td>Wet bulb air temperature, °F</td>
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<td>Iterative working fluid temperature value, °F</td>
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<tr>
<td>WTT4</td>
<td>Iterative working fluid temperature value, °F</td>
</tr>
<tr>
<td>WVELL</td>
<td>Flow velocity in preheater tubes, ft/hr</td>
</tr>
<tr>
<td>WVELV</td>
<td>Flow velocity in vaporizer tubes, ft/hr</td>
</tr>
<tr>
<td>WVELS</td>
<td>Flow velocity in superheater tubes, ft/hr</td>
</tr>
<tr>
<td>X</td>
<td>Reference enthalpy, Btu/lbm</td>
</tr>
<tr>
<td>Y</td>
<td>Reference entropy, Btu/lbm/°F</td>
</tr>
<tr>
<td>Z</td>
<td>Constant, = 0.185053 in$^2$/ft$^2$ . Btu/ft·lb_f</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>Freon, NH$_3$, or hydrocarbon constant</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>Hydrocarbon constant</td>
</tr>
<tr>
<td>$\Delta B$</td>
<td>Net availability</td>
</tr>
<tr>
<td>$\Delta H$</td>
<td>Net enthalpy, Btu/lbm</td>
</tr>
<tr>
<td>$\Delta Q$</td>
<td>Net heat</td>
</tr>
<tr>
<td>$\Delta W$</td>
<td>Net work</td>
</tr>
<tr>
<td>$n_C$</td>
<td>Carnot efficiency</td>
</tr>
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</table>
\( \eta_{\text{cycle}} \quad \text{cycle efficiency} \)

\( \eta_{\text{pump}} \quad \text{pump efficiency} \)

\( \eta_{\text{Rankine}} \quad \text{Rankine cycle efficiency} \)

\( \eta_{\text{u}} \quad \text{utilization efficiency} \)

\( \mu \quad \text{viscosity, lbm/ft/hr} \)

\( \rho \quad \text{density, lbm/ft}^3 \)

\( \phi \quad \text{conversion factor, } = \frac{313.611 \text{ psia cu.ft}}{\text{lb-mole}} \cdot \frac{\text{lb}}{\text{Btu}} \)

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)
MODELING 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 thesis was motivated by the needs of the Energy Research and Development Administration to rapidly develop new energy sources that can fulfill the ever increasing demands for electrical power in the United States.

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 results in the optimum utilization of a geothermal energy source require development, in both an economic and a thermodynamic sense. This thesis 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-condensible 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 subcritical 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. The fluid is extracted from this tank, repressurized with booster and feed pumps and returned to the heat exchangers for repetition of the cycle.

Cooling water is supplied to 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 blowdown, 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. 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.
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 thesis 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 I. 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

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

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 proceeded 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-condensible 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 directly expanded 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-condensible 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 field 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. A later design called the Hutchinson-Holt Process has been developed using isobutane, but the details are not known for proprietary reasons. 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., Lawrence Berkeley Laboratories, the University of Hawaii, Aerojet Nuclear Company, San Diego Gas and Electric, and Roger's Engineering. 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>
<thead>
<tr>
<th>Plant</th>
<th>Size (MWe)</th>
<th>Working Fluid</th>
<th>Geothermal Fluid Temp (°F)</th>
<th>Max Working Fluid Temp (°F)</th>
<th>Max Working Fluid Pressure (psia)</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>TRW, Inc.</td>
<td>10.0</td>
<td>Isobutane</td>
<td>350°F</td>
<td>310°F</td>
<td>450</td>
<td>subcritical cycle with regeneration; heat rejection with air cooled condenser.</td>
</tr>
<tr>
<td>Lawrence Berkeley Labs</td>
<td>10.0</td>
<td>Isobutane</td>
<td>392°F</td>
<td>370°F</td>
<td>600</td>
<td>simple supercritical Rankine cycle.</td>
</tr>
<tr>
<td>U. of Hawaii</td>
<td>10.0</td>
<td>Isobutane</td>
<td>350°F</td>
<td>320°F</td>
<td>400</td>
<td>subcritical cycle with regeneration.</td>
</tr>
<tr>
<td>Aerojet Nuclear</td>
<td>10.0</td>
<td>Isobutane</td>
<td>300°F</td>
<td>250°F</td>
<td>350</td>
<td>dual boiling system with high pressure and low pressure turbine, series geothermal flow through high, then low temperature side, regeneration used.</td>
</tr>
<tr>
<td>SDG&amp;E</td>
<td>10.0</td>
<td>Isobutane</td>
<td>370°F</td>
<td>300°F</td>
<td>500</td>
<td>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.</td>
</tr>
<tr>
<td>Rogers Engr.</td>
<td>10.0</td>
<td>Isobutane</td>
<td>325°F</td>
<td>290°F</td>
<td>500</td>
<td>simple subcritical Rankine cycle.</td>
</tr>
</tbody>
</table>
These studies, and the power plant design, are in general proposing 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 thesis 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 geopressed 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. 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. 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. 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. 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.

\[ \eta_{\text{Rankine, isentropic}} = \frac{H_1 - H_{2\text{EST}}}{H_1 - H_{2\text{SATL}}} \]  
(1)

\[ \eta_{\text{Rankine, actual}} = \frac{H_1 - H_2}{H_1 - H_{2\text{SATL}}} \]  
(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.

\[
\text{kW}_{\text{pump}} = \frac{\dot{m}(P_a - P_b)(144)}{\rho(1.98 \times 10^6)(.745)\eta_{\text{pump}}}
\]  

(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 vaporization 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.


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>
<thead>
<tr>
<th>Compound</th>
<th>Critical Temperature ($^\circ$F)</th>
<th>Critical Pressure (psia)</th>
<th>Critical Density (lb/ft$^3$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Isobutane</td>
<td>274.96</td>
<td>529.06</td>
<td>13.80</td>
</tr>
<tr>
<td>n-butane</td>
<td>305.62</td>
<td>550.66</td>
<td>14.24</td>
</tr>
<tr>
<td>R-11; trichlorofluoromethane</td>
<td>388.4</td>
<td>640.0</td>
<td>34.6</td>
</tr>
<tr>
<td>R-12; dichlorodifluoromethane</td>
<td>233.6</td>
<td>597.0</td>
<td>34.84</td>
</tr>
<tr>
<td>R-21; dichlorodifluoromethane</td>
<td>353.3</td>
<td>750.0</td>
<td>36.0</td>
</tr>
<tr>
<td>R-22; chlorodifluoromethane</td>
<td>204.8</td>
<td>721.9</td>
<td>32.8</td>
</tr>
<tr>
<td>R-113; trichlorotrifluoroethane</td>
<td>417.4</td>
<td>498.9</td>
<td>36.0</td>
</tr>
<tr>
<td>R-114; dichlorotetrafluoroethane</td>
<td>294.3</td>
<td>473.0</td>
<td>36.3</td>
</tr>
<tr>
<td>Ammonia</td>
<td>271.4</td>
<td>1657.0</td>
<td>14.6</td>
</tr>
</tbody>
</table>
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

\[ P = \sum_{i=1}^{5} f_i (V-b)^{-i} \]  

where \( f_i \) is a temperature dependent function

\[ f_i = A_i + B_i T + C_i \exp \left( -\frac{K_i T}{T_c} \right). \]  

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

\[ P = \frac{RT}{(V-b)} + \frac{A_2 + B_2 T + C_2 \exp(-K T/T_c)}{(V-b)^2} \]
\[ + \frac{A_3 + B_3 T + C_3 \exp(-K T/T_c)}{(V-b)^3} + \frac{A_4 + B_4 T + C_4 \exp(-K T/T_c)}{(V-b)^4} \]
\[ + \frac{A_5 + B_5 T + C_5 \exp(-K T/T_c)}{(V-b)^5} + \frac{A_6 + B_6 T + C_6 \exp(-K T/T_c)}{\exp(aV)(1 + C \exp(aV))} \]

The letters \( A_i, B_i, C_i, a, \) 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.

\[
\log_{10} P_{\text{saturation}} = A + \frac{B}{T} + C \log_{10}(T) + DT \\
+ E\left(\frac{F-T}{T}\right)\log_{10}(F-T)
\]  

(7)

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

\[
H = aT + \frac{bT^2}{2} + \frac{cT^3}{3} + \frac{dT^4}{4} - \frac{f}{T} + JPV + J\left[\frac{A_2}{(V-b)} + \frac{A_3}{2(V-b)^2} + \frac{A_4}{3(V-b)^3} + \frac{A_5}{4(V-b)^4} + \frac{A_6}{\alpha}\left(\frac{1}{\exp(\alpha V)} - C \ln\left(1 + \frac{1}{C \exp(\alpha V)}\right)\right)\right]\]

+ \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]

\[+ \frac{C_5}{4(V-b)^4} + \frac{C_6}{\alpha\exp(\alpha V)} - \frac{C_6C^-}{\alpha} \ln\left(1 + \frac{1}{C^- \exp(\alpha V)}\right)\] + \chi
\]  

(8)

where J = .1851

X = reference enthalpy (based on -40°F for the saturated liquid)
\[ S = a \ln T + bT + \frac{cT^2}{2} + \frac{dT^3}{3} - \frac{f}{2T^2} + JR \ln(V-b) \]

\[- J \left[ \frac{B_2}{(V-b)} + \frac{B_3}{2(V-b)^2} + \frac{B_4}{3(V-b)^3} + \frac{B_5}{4(V-b)^4} \right] \]

\(+ \frac{B_6}{a} \left( \frac{1}{\exp(\alpha V)} - C \ln \left( 1 + \frac{1}{C \exp(\alpha V)} \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] \]

\(+ \frac{C_6}{a \exp(\alpha V)} - \frac{C_6 C^-}{a} \ln \left( 1 + \frac{1}{C \exp(\alpha V)} \right) \right) + \gamma \] (9)

where \( \gamma \) = 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. 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:

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

The letters \(B_0, A_0, C_0, D_0, E_0, a, b, c, d, \alpha,\) and \(\gamma\) 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.

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

where \(\phi\) is the conversion from Btu/lb\(_m\) to psia ft\(^3\)/lb mole and \(H_0\) is the ideal gas enthalpy at the temperature in question;

\[
H_0 = A_{00} + A_{10}T + A_{20}T^2 + A_{30}T^3 + A_{40}T^4
\]
\[ S = - \frac{R}{\phi} \ln(\rho RT) - \left( B_0 R + \frac{2C_0}{T^3} - \frac{3D_0}{T^4} + \frac{4E_0}{T^5} \right) \frac{\rho}{\phi} - \frac{1}{2} \left( bR + \frac{d}{T^2} \right) \frac{\rho^2}{\phi} \]
+ \frac{\alpha \rho}{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 \]

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

\[ S_0 = B_{00} + B_{10} T + B_{20} T^2 + B_{30} T^3 + B_{40} T^4 \]

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,

\[ \ln f = \ln(\rho RT) + 2 \left( B_{0RT} - A_0 - \frac{C_0}{T^2} + \frac{D_0}{T^3} + \frac{E_0}{T^4} \right) \frac{\rho}{RT} \]
+ \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] \]

(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 thesis, 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.
The working fluid state points are indicated; this nomenclature will be used throughout this thesis. 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.
constant pressure removal of the heat of vaporization from the working fluid.

compression of the liquid working fluid.

constant pressure addition of heat to the working fluid from the geothermal fluid.

(sub-critical only) heat addition to the working fluid between pump exhaust and the saturated liquid state at constant pressure.

(sub-critical only) additions of the heat of vaporization to the working fluid at constant pressure.

(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.
A) The minimum working fluid temperature $T_{2\text{SATL}}$, 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, $T_{2\text{SATL}}$, the saturation pressure $P_2$ is determined in subroutine PTWO. PTWO 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 $T_2\text{SATL}$, the enthalpy at this state point, $H_2\text{SATL}$, 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," $\text{APP}$, 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, $W_1$. 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 P1. If the difference between P1 and the computed pressure is less than 0.5 psia, the density D1 has been found. If this difference is greater, a new estimate for D1 is found based on the following equation,

$$D_{1\text{new}} = D_{1\text{old}} * (1.0 + (P1 - P_{\text{guess}})/P1). \quad (16)$$

This new value of D1 is then used to compute a new calculated value for pressure.

The iteration is continued until the convergence criteria is satisfied. The enthalpy H1 at state 1 is computed with subroutines HEQST and HIDEAL (equations (11) and (12)). The entropy, S1, 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 P2. The entropy at state 2EST, S2EST, is therefore equal to S1 computed in step B. The temperature and density, T_{2\text{EST}} and D_{2\text{EST}}, are calculated by simultaneously solving the equations (10) and (13) with (14). These equations are of the form
\begin{align*}
P &= f(T, D) & \text{(equation 10)} \\
S &= f(T, D) + f(T) & \text{(equation 13 & 14)}
\end{align*}

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

\begin{equation}
QISO = GRKW \times 3414.43 / (H1 - H2EST) / \text{OEFF} \quad (17)
\end{equation}

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

\begin{equation}
H2 = H1 - (H1 - H2EST)(\text{OEFF}) \quad (18)
\end{equation}

In addition, the actual Rankine cycle efficiency REFF is determined according to the equation
REFF = (H1 - H2) / (H1 - H2SATL)  \hspace{1cm} (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 RQNWT, 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 RQNWT 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 $P_4$ and $P_5$ are determined based on $P_1$.

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 $P_1$. 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:

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

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

The temperature at state 3 is determined from $\Delta H$ and the temperature at 2SATL as follows:

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

\[ QWAT = \frac{QISO(HI-H4)}{Cp(WT1-WT3)} \]  

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.
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**

<table>
<thead>
<tr>
<th>Working Fluid of Interest</th>
<th>Subroutine Called</th>
<th>Density Equation [\text{lb m/ft}^3]</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-11, R-22, NH₃</td>
<td>DNLIQA</td>
<td>[\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} ]</td>
</tr>
<tr>
<td>R-12, R-114</td>
<td>DNLIQB</td>
<td>[\rho = AL + BL(T_c - T) + CL(T_c - T)^{1/2} + DL(T_c - T)^{1/3} + EL(T_c - T)^2 ]</td>
</tr>
<tr>
<td>R-21, R-113</td>
<td>DNLIQC</td>
<td>[\rho = AL + BL(T_c) + CL(T_c)^2 ]</td>
</tr>
</tbody>
</table>

The coefficients A1, B1, C1, D1, and E1 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, $H_{2\text{SATL}}$, is as follows:

$$H_{2\text{SATL}} = H_{2\text{SATV}} - SDHVAP$$  \hfill (24)

where $H_{2\text{SATV}} = \text{saturated vapor enthalpy at P2}$ and $SDHVAP = \text{heat of vaporization at P2}$.  

The first quantity to be determined in equation (23) is $H_{2\text{SATV}}$. In order to compute $H_{2\text{SATV}}$, the specific volume at this state, $SV_{\text{SATV}}$, must first be determined. An initial guess based on ideal gas laws is made for $SV_{\text{SATV}}$. This estimated specific volume and the known temperature $T_{2\text{SATL}}$ 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 $P_2$. If the absolute value of the difference between these two pressures is within 0.5 psia, a suitable value for $SV_{\text{SATV}}$ has been found. If the convergence criterion is not satisfied, a new estimate for $SV_{\text{SATV}}$ is found based on the following equation

$$SV_{\text{SATV}}_{\text{new}} = SV_{\text{SATV}}_{\text{old}} \times (1 + (P_{\text{calc}} - P_2)/P_2) \hfill (25)$$
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:

\[ SDHVAP = Z \times SDPDT \times (T2SATL + 459.67)(SVSATV - \frac{1}{DSAT}) \]  

(26)

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

0.185053 \( \text{in}^2 \cdot \frac{\text{Btu}}{\text{ft}^2 \cdot \text{ft-lb}_f} \), 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
\[
SDPDT = P \left[ \frac{-\ln 10 \times BFR + EF \times \log_{10}(F - T) - C}{T^2} + \frac{E}{T} + D \ln 10 \right] \tag{27}
\]

The constants B, C, D, E and F are provided in subroutine SETCON. The enthalpy of saturated liquid, \(H_{2\text{SATL}}\) 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 \(P_1\) is known from subroutine OPBIN. The temperature \(T_1\) is the difference between the entering geothermal fluid temperature \(W_{T1}\) and the heat exchanger "approach" \(APP\). The specific volume at state 1, \(V_1\), is computed through an identical technique used in step AA for \(SV_{SATV}\). The enthalpy \(H_1\) is computed from subroutine HVAP using equation (8). The entropy \(S_1\) 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 \(S_{2EST}\) is equal to \(S_1\) because of the isentropic expansion from \(P_1\) to \(P_2\). The temperature and specific volume at this
state, $T_{2\text{EST}}$ and $V_{2\text{EST}}$, 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 $T_{2\text{EST}}$ and $V_{2\text{EST}}$ are then used with the value of $P_2$ to determine $H_{2\text{EST}}$ in subroutine HVAP.

DD) Step DD is used to compute the following values:
- $Q_{\text{ISO}}$ - working fluid flow rate
- $H_2$ - enthalpy at state 2
- $\text{REF}_f$ - Rankine cycle efficiency
- $T_2$ - temperature at state 2
- $V_2$ - specific volume at state 2. These values are computed in the same manner as indicated in step D of subroutine BUTANE. $Q_{\text{ISO}}$ is found from equation (17), $H_2$ from equation (18), and $\text{REF}_f$ from equation (19). The values for $T_2$ and $V_2$ 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 $P_1$ 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 P1. The value of T4 = T5 is adjusted until the calculated pressure converges to P1 within 0.5 psia. The specific volume at state 4 (V4, saturated liquid) is computed as the inverse of the appropriate density equation listed in Table 4. The specific volume at state 5, V5, is determined iteratively using subroutine EQSTAP, similar to the method used to locate SVSATV in step AA. The pressures at states 4 and 5 (P4 and P5) are based upon P1. Using T4 and V4, the enthalpy and entropy at state 4 are computed using subroutines HVAP and SVAP. The enthalpy and entropy at state 5 (H5 and S5) 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.

\[
\text{Property value} = K_0 + K_1T + K_2T^2 + K_3T^3 \quad \text{for} \quad T_x < T < T_y
\]

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) \hspace{1cm} (29)

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

\[ I = 1, 13 \quad \text{Property Index} \]

\[ = 1 \quad \text{specific heat of liquid} \quad [\text{Btu/lbm/°F}] \]
\[ = 2 \quad \text{thermal conductivity of liquid} \quad [\text{Btu/hr/ft/°F}] \]
\[ = 3 \quad \text{density of liquid} \quad [\text{lbm/ft}^3] \]
\[ = 4 \quad \text{viscosity of liquid} \quad [\text{lbm/ft/hr}] \]
\[ = 5 \quad \text{specific heat of gas} \quad [\text{Btu/lbm/°F}] \]
\[ = 6 \quad \text{thermal conductivity of gas} \quad [\text{Btu/hr/ft/°F}] \]
\[ = 7 \quad \text{density of gas} \quad [\text{lbm/ft}^3] \]
\[ = 8 \quad \text{viscosity of gas} \quad [\text{lbm/ft/hr}] \]
\[ = 9 \quad \text{heat of vaporization} \quad [\text{Btu/lbm}] \]
\[ = 10 \quad \text{critical pressure} \quad [\text{psia}] \]
\[ = 11 \quad \text{critical temperature} \quad [\text{°F}] \]
\[ = 12 \quad \text{surface tension} \quad [\text{lb}_f/\text{ft}] \]
\[ = 13 \quad \text{molecular weight} \quad [\text{lb}_m/\text{lb mole}] \]

\[ J = 1, 10 \quad \text{Fluid Index} \]

\[ = 1 \quad \text{water} \]
\[ = 2 \quad \text{isobutane} \]
\[ = 3 \quad \text{n-butane} \]
\[ = 4 \quad \text{R-11} \]
\[ = 5 \quad \text{R-12} \]
\[ = 6 \quad \text{R-21} \]
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 [lbm/hr]
b) working fluid flow rate; QISO [lbm/hr]
c) geothermal fluid temperatures; WT1, WT2, WT3, and WT4 [°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
\[ T1 = \text{superheater outlet temperature} \]
\[ T3 = \text{preheater inlet temperature} \]
\[ T4 = \text{preheater outlet (vaporizer inlet) temperature} \]
\[ T5 = \text{vaporizer outlet (superheater inlet) temperature} \]

f) working fluid enthalpies; H3, H4, H5, and H1 [Btu/lbm] (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
\[ DIAL = \text{preheater inside tube diameter} \]
\[ DIAOL = \text{preheater outside tube diameter} \]
\[ DIAV = \text{vaporizer inside tube diameter} \]
\[ DIAOV = \text{vaporizer outside tube diameter} \]
\[ DIAS = \text{superheater inside tube diameter} \]
\[ DIAOS = \text{superheater outside tube diameter} \]
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 e changer 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:

\[
\begin{align*}
Re &= \frac{\rho V D}{\mu} \quad (30) \\
Pr &= \frac{c_p \mu}{K} \quad (31)
\end{align*}
\]

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

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

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

where \( n = 0.3 \) for cooling

\[
n = 0.4 \text{ for heating.}
\]

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

\[ A_{\text{unit}} = \frac{s^2 \sin 60^\circ}{2} - \frac{1}{2} \frac{D^2}{4} \]  

(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

\[ D_e = \frac{4 \times A_{\text{unit}}}{\text{Wetted Perimeter}} \] 

(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 super-
heater, the specific heat, thermal conductivity,
and viscosity are determined for the gaseous work-
ing fluid using average temperatures and sub-
routine 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 VELOCY, using this equation

\[ V = \frac{m}{\rho \times \text{total flow area}} \]  \hspace{1cm} (36)

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

9) The Reynolds number and Prandtl number are deter-
mined 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 corre-
lation based upon two-phase properties is used
for this coefficient. Chen states that a reasonable boiling coefficient can be determined using the following equation

\[ h_{\text{boiling}} = F \times 0.023 \times \frac{K_L}{D} \times (Re_L)^{0.8} (Pr_L)^{0.4} \]  

(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

\[ \text{Martinelli Parameter} = M = \left( \frac{\rho_L}{\rho_v} \right)^{1.9} \left( \frac{\mu_v}{\mu_L} \right)^{10} \]  

(38)

The factor \( F \) has been reduced to a polynomial;

\[ F = 1.77 + 1.179M - 0.0075M^2 + 0.000029M^3 \]  

(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;

\[
U = \frac{1}{\frac{D_0}{D_i h_i} + \frac{D_0}{2K_{\text{wall}}} \ln \left( \frac{D_0}{D_i} \right) + \frac{1}{h_0} + \text{FOULFO} + \text{FOULFI}}
\]  

(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;

\[ A = \frac{QISO \times (H_{\text{out}} - H_{\text{in}})}{U \times \text{LMTD}} \]  

(41)

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

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

(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;

\[ \text{Length} = \frac{A}{\pi D_0 \times \text{no. of tubes}} \]  

(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 HH\textsubscript{1} is then calculated with subroutine HVAP using the values of TT\textsubscript{1} and VV\textsubscript{1}.

S-2) In this step, the geothermal fluid inlet and outlet temperatures are set. The outlet geothermal fluid WTT\textsubscript{4} for a section is set equal to the inlet temperature from the last section. If the section being analyzed is the first, this temperature WTT\textsubscript{4} is set equal to the end-point outlet temperature WT\textsubscript{4}. The inlet geothermal fluid temperature for a section is defined as follows:

\[
WTT1 = \frac{QISO (HH1-HH3)}{QWAT CP} + WTT4
\]  

(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 REYN0 and PRNDTL, using equations (30) and (31).

S-6) The inside heat transfer coefficient is determined from subroutine HTCOEF 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 VELOCY.

S-11) The outside heat transfer coefficient is determined with subroutine HTCOEF 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 HTLNGH, 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.
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:

\[ Q_T = QCWAT \times C_p \times (CWT_{\text{in}} - CWT_{\text{out}}) \]  

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>
<thead>
<tr>
<th><strong>TYPE OF PLANT</strong></th>
<th>BINARY</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>SUPERCRITICAL CYCLE</strong></td>
<td>FALSE</td>
</tr>
<tr>
<td><strong>FLUID</strong></td>
<td>ISOBUTANE</td>
</tr>
<tr>
<td><strong>PRESSURE FACTOR</strong></td>
<td>2.0</td>
</tr>
<tr>
<td><strong>FLUID TEMPERATURE</strong></td>
<td>356°F</td>
</tr>
<tr>
<td><strong>FLOW RATE PER WELL</strong></td>
<td>500,000 lb/hr</td>
</tr>
<tr>
<td><strong>PLANT SIZE</strong></td>
<td>55,000 kWe</td>
</tr>
<tr>
<td><strong>WELL SPACING</strong></td>
<td>20 ACRES</td>
</tr>
<tr>
<td><strong>AVERAGE WELL LIFE</strong></td>
<td>15 YEARS</td>
</tr>
<tr>
<td><strong>RESERVOIR SIZE</strong></td>
<td>275,000 kW</td>
</tr>
<tr>
<td><strong>WET BULB AIR TEMPERATURE</strong></td>
<td>50.0°F</td>
</tr>
<tr>
<td><strong>DRY BULB AIR TEMPERATURE</strong></td>
<td>70.0°F</td>
</tr>
</tbody>
</table>
TABLE 6. Typical Computer Output for Binary Cycle Plant Model

**ISOBUTANE SUBCRITICAL BINARY CYCLE FOR 55 MWE PLANT**

**BINARY POWERPLANT SYSTEM AND TRANSMISSION MODEL**

<table>
<thead>
<tr>
<th>Working Fluid</th>
<th>Isobutane</th>
</tr>
</thead>
<tbody>
<tr>
<td>The cycle is subcritical</td>
<td></td>
</tr>
</tbody>
</table>

**Cooling Water**

<table>
<thead>
<tr>
<th>Component</th>
<th>Inlet Temperature (Deg. F)</th>
<th>Outlet Temperature (Deg. F)</th>
<th>Flow Rate (L/Hr)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbine Oil Cooler</td>
<td>6.200000E+1</td>
<td>1.200000E+2</td>
<td>6.16947E+4</td>
</tr>
<tr>
<td>H2O</td>
<td>6.200000E+1</td>
<td>1.200000E+2</td>
<td>6.14289E+4</td>
</tr>
</tbody>
</table>

**Final Cycle Conditions**

<table>
<thead>
<tr>
<th>Working Fluid</th>
<th>Temperatures (Deg. F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Approx-Heat Exchanger Approach</td>
<td>3.000000E+01</td>
</tr>
<tr>
<td>T1-Turbine Inlet</td>
<td>3.257216E+02</td>
</tr>
<tr>
<td>T2-Exhaust (Inlet)</td>
<td>2.068035E+02</td>
</tr>
<tr>
<td>T3-Condenser Outlet</td>
<td>9.200000E+01</td>
</tr>
<tr>
<td>T4-Superheater Inlet</td>
<td>9.590698E+01</td>
</tr>
<tr>
<td>T5-Supervisor Inlet</td>
<td>2.549341E+02</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Working Fluid</th>
<th>Pressures (PSIA)</th>
</tr>
</thead>
<tbody>
<tr>
<td>P1-Turbine Inlet</td>
<td>4.421796E+02</td>
</tr>
<tr>
<td>P2-Turbine Exhaust</td>
<td>6.449399E+01</td>
</tr>
<tr>
<td>P3-Preface Inlet</td>
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## Heat Exchangers

### Supercritical Cycle

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### Geothermal Fluid

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Subroutine Temperatures

- APPROX APPROACH TO COOLING TOWER (F): 1*200000E+01
- COOLING TOWER AIR FLOW RATE (CFM): 0.322239E+06
- COOLING TOWER EVAPORATION RATE (LB/HR): 4.445719E+05
- NET BULB AT TEMP TEMPERATURE (F): 0.000000E+01
- DRY BULB AT TEMP TEMPERATURE (F): 0.000000E+01
- DRY BULB AT TEMP TEMPERATURE (F): 9.920000E+01
- HUMIDITY RATIO: 0.000000E+01
- HEAT LOAD (BTU/HR): 1.154438E+09

Cooling Water Requirements (LB/HR)

- TURBINE OIL COOLER: 1.61654E+04
- HYDROGEN COOLER: 0.14288E+04
- APPROX APPROACH TO CONDENSER (F): 0.000000E+01
- RANGE-CONDENSER WATER RANGE (F): 5.000000E+01
- FLOW RATE IN CONDENSER: 7.20735E+07
- MAKEUP FLOW RATE TO COOLING TOWER: 4.445719E+05

Kilowatt Losses

- FANS: 1.22097E+03
- CONDENSATE PUMPS TO COOLING TOWER: 1.04676E+03
- COOLING WATER PUMPS: 1.154313E+03
- REINJECTION PUMPS: 0.88746E+02
- ROOSTERED PUMPS TO OVERPRESSURIZE WATER: 7.993104E+01
- DOWNHOLE PUMPS TO PREVENT TWO PHASE FLOW: 0.000000E+01
- BOOSTER PUMPS: 1.91199E+02
- FEED PUMPS: 9.13915E+03

Overall Results

- GROSS KILOWATT OUTPUT OF THE PLANT: 5.000000E+04
- GEOTHERMAL FLUID FLOW RATE (LB/HR) (INCLUDING SALT): 0.057497E+06
- FLOW RATE (LB/HR) OF ISOBUTANE: 8.87471E+06
- TOTAL FLOW RATE (LB/HR) OF ISOBUTANE: 7.28494E+07
- TOTAL KILOWATT LOSSES: 1.12761E+03
- NET KILOWATT OUTPUT: 5.88724E+04
- NET HEAT RATE (HTU/W) HR: 2.5407E+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 $\Delta W$ can be produced by transferring heat $\Delta Q$ 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:

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

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):

\[ W_{\text{max}} = -\int_{T_{h,in}}^{T_{h,out}} \left( \frac{T_h - T_0}{T_h} \right) C_{pd} dT_h \]  

or

\[ W_{\text{max}} = -\int_{T_{h,in}}^{T_{h,out}} C_p \left[ 1 - \frac{T_0}{T_h} \right] dT_h \]

or

\[ W_{\text{max}} = -\int_{T_{h,in}}^{T_{h,out}} C_{pd} dT_h + C_p T_0 \int_{T_{h,in}}^{T_{h,out}} \frac{dT_h}{T_h} \]

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

\[ W_{\text{max}} = -\left[ \Delta H - T_0 \Delta S \right]_{T_{h,in}}^{T_{h,out}} \]  

(50)

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

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

\[ W_{\text{max}} = C_p (T_{h,in} - T_{h,out}) - T_0 C_p \ln \frac{T_{h,in}}{T_{h,out}} \]  

(51)

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

$$\eta_u = \frac{P_{net}}{QWAT \times W_{max}} = \frac{P_{total} - P_{internal \ use}}{QWAT \times C_p \ (T_{h,in} - T_{h,out})} - T_0 \ln \frac{T_{h,in}}{T_{h,out}}$$

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, $\eta_{cycle}$, defined as

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

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 $\eta_{cycle}$. 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 $\eta_u$. 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 $\eta_u$ 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 $\eta_u$ 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 $S = 1.18 \text{ Btu/}lb{/}°\text{F}$ 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

APPROACH TO THE HEAT EXCHANGER (°F)

- 20
- 40
- 60
- 80
- 100

WORKING FLUID - ISOButane
GEOTHERMAL FLUID TEMPERATURE - 300°F
TURBINE EXHAUST PRESSURE - 72 psia
MINIMUM WORKING FLUID TEMPERATURE - 100°F
PINCH POINT - 15°F
TURBINE EFFICIENCY - 78.5%
POWER OUTPUT - 100 MW_e
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.

\[
\eta_u = \frac{P_{\text{net}}}{Q_{\text{WAT}}(\Delta B)} \quad (54)
\]

\[
Q_{\text{WAT}} = \frac{Q_{\text{ISO}}(H_1-H_4)}{C_p(WT_1-WT_3)} \quad (55)
\]

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, \( T_1 = WT_1-\text{APP} \).
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 $\text{APP} = 60\,\text{°F}$ 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 preheater 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

Shift of Pinch Point Location with Increasing Pressure

Super-Critical

Sub-Critical
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 $\eta_u$ 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 $\eta_u$ is reached at this pressure; therefore, this pressure is the suggested turbine inlet pressure. For the geothermal temperature of 400°F
WORKING FLUID - ISOButANE
GEOTHERMAL TEMPERATURE - 400°F
TURBINE EXHAUST PRESSURE - 72 psia
MINIMUM WORKING FLUID TEMPERATURE - 100°F
PINCH POINT - 15°F
TURBINE EFFICIENCY - 78.5%
POWER OUTPUT - 100 MW<sub>e</sub>

APPROACH TEMPERATURE TO
THE HEAT EXCHANGER, °F

--- 20
--- 40
--- 60
--- 80
--- 100

FIGURE 15. Utilization Efficiency versus Turbine Inlet Pressure; Geothermal Temperature = 400°F
WORKING FLUID - ISOButANE
GEOTHERMAL TEMPERATURE - 500°F
TURBINE EXHAUST PRESSURE - 72 psia
MINIMUM WORKING FLUID TEMPERATURE - 100°F
PINCH POINT - 15°F
TURBINE EFFICIENCY - 78.5%
POWER OUTPUT - 100 MWₑ

APPROACH TEMPERATURE TO
THE HEAT EXCHANGER, °F

--- 20
--- 40
--- 60

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 $\eta_u$ 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 T2SATL (related to the sink temperature T₀) and the pinch point is necessary. In Figures 11-16 the values of T2SATL and PINCHP were set at 100°F and 15°F, respectively. Selecting a base curve with WT1 = 300°F, APP = 40°F, and PINCHP = 15°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 T2SATL are shown: 90, 100, and 120°F. As expected, the highest utilization efficiency occurs at the lowest value of T2SATL. 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 T2SATL (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

- PINCH POINT \( ^\circ \text{F} \)
  - 15
  - 20
  - 25

- WORKING FLUID: ISOBUTANE
- GEOTHERMAL TEMPERATURE: 300\(^\circ\)F
- TURBINE EXHAUST PRESSURE: 72 psia
- MINIMUM WORKING FLUID TEMPERATURE: 100\(^\circ\)F
- APPROACH TO THE HEAT EXCHANGER: 40\(^\circ\)F
- TURBINE EFFICIENCY: 78.5%
- POWER OUTPUT: 100 MW\(_e\)
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, T2SATL = 100°F, \( \eta_{\text{turbine}} = 0.785 \), 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 \( S = 1.18 \text{ Btu/lb/°F} \) 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
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 $S = 1.18 \text{ Btu/lb/°F}$ 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 = $T_3 + 25°F$ ($T_3$ 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
\( n_u \), and "approach" of 20°F is selected, and the operating
point is located at \( T_1 = W_{T1} - 20°F \) and \( S = 1.18 \text{ Btu/lb/°F} \).
If this point results in a pressure \( P_1 \) exceeding the maxi-
mum allowable pressure, the operating point is selected at
\( P_1 = 1500 \text{ psia} \) and the minimum value of \( T_1 \), limited by
boundary condition(3).

Input to subroutine OPBIN is as follows:
(a) geothermal fluid temperature at inlet, \( W_{T1} \)
(b) minimum working fluid temperature, \( T_{2SATL} \)

Output of this subroutine is as follows:
(a) turbine inlet pressure, \( P_1 \)
(b) heat exchanger "approach", \( APP \)
(c) turbine inlet temperature, \( T_1 \)
(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:

\[ \eta_u = \frac{P_{\text{net}}}{QWAT \times C_p \times [WT_1 - T_0 - T_0 \times \ln(WT_1/T_0)]} \]

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, \( \eta_{\text{cycle}} \), defined as

\[ \eta_{\text{cycle}} = \frac{P_{\text{net}}}{QWAT \times C_p \times (WT_1 - T_0)} \]

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 \( \eta_{\text{cycle}} \). 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.

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

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.


56. "Binary Fluid Conversion Systems for Low Temperature Geothermal Fluids," internal memo, Battelle-Northwest Laboratories, Richland, WA.


Downing, R. C., Refrigerant Equations, Dupont Co., Freon Products Division, No. 2313.


APPENDIX I

COEFFICIENTS FOR LIGHT HYDROCARBON,
FREON, AND AMMONIA EQUATIONS
## APPENDIX I

### Coefficients for Light Hydrocarbon Equations

#### Coefficients for Equations (10), (11), (13), & (15)

<table>
<thead>
<tr>
<th></th>
<th>Isobutane</th>
<th>N-Butane</th>
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<tbody>
<tr>
<td>$A_0$</td>
<td>$3.7264 \times 10^4$</td>
<td>$3.25447 \times 10^4$</td>
</tr>
<tr>
<td>$B_0$</td>
<td>$1.8789$</td>
<td>$1.56588$</td>
</tr>
<tr>
<td>$C_0$</td>
<td>$1.01413 \times 10^{10}$</td>
<td>$1.37436 \times 10^{10}$</td>
</tr>
<tr>
<td>$D_0$</td>
<td>$8.53176 \times 10^{11}$</td>
<td>$3.33159 \times 10^{11}$</td>
</tr>
<tr>
<td>$E_0$</td>
<td>$8.4086 \times 10^{13}$</td>
<td>$2.30902 \times 10^{12}$</td>
</tr>
<tr>
<td>$a$</td>
<td>$4.7991 \times 10^4$</td>
<td>$7.11818 \times 10^4$</td>
</tr>
<tr>
<td>$b$</td>
<td>$8.58663$</td>
<td>$9.14066$</td>
</tr>
<tr>
<td>$c$</td>
<td>$4.06763 \times 10^{10}$</td>
<td>$7.00044 \times 10^{10}$</td>
</tr>
<tr>
<td>$d$</td>
<td>$2.16863 \times 10^7$</td>
<td>$3.64238 \times 10^7$</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>$4.23987$</td>
<td>$4.00985$</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>$7.11486$</td>
<td>$7.54122$</td>
</tr>
<tr>
<td>$R$</td>
<td>$10.7335$</td>
<td>$10.7335$</td>
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#### Coefficients for Equation (12)

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<td>$-776.11$</td>
<td>$-739.77$</td>
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<td>$A_{10}$</td>
<td>$0.06872$</td>
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<td>$A_{20}$</td>
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<td>$A_{30}$</td>
<td>$0.0$</td>
<td>$3.93 \times 10^{-7}$</td>
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<td>$A_{40}$</td>
<td>$0.0$</td>
<td>$-1.594 \times 10^{-10}$</td>
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<td>Coefficients for Equation (14)</td>
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<td>n-Butane</td>
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<td>-------------------------------</td>
<td>-----------------</td>
<td>-----------------</td>
</tr>
<tr>
<td>$B_{00}$</td>
<td>0.86401</td>
<td>0.9008</td>
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<td>$B_{10}$</td>
<td>$9.079 \times 10^{-4}$</td>
<td>$10.060 \times 10^{-4}$</td>
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<td>$B_{20}$</td>
<td>$-2.073 \times 10^{-7}$</td>
<td>$-3.006 \times 10^{-7}$</td>
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<td>$B_{30}$</td>
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<td>R-12</td>
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<tr>
<td>R</td>
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<td>B₂</td>
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<td>C₂</td>
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<td>A₃</td>
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<tr>
<td>B₃</td>
<td>4.8751 x 10⁻⁵</td>
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</tr>
<tr>
<td>C₃</td>
<td>1.22037</td>
<td>1.3114</td>
</tr>
<tr>
<td>A₄</td>
<td>1.6873 x 10⁻³</td>
<td>-5.4874 x 10⁻⁴</td>
</tr>
<tr>
<td>B₄</td>
<td>-1.8051 x 10⁻⁶</td>
<td>0.0</td>
</tr>
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<tr>
<td>A₅</td>
<td>-2.359 x 10⁻⁵</td>
<td>0.0</td>
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<tr>
<td>B₅</td>
<td>2.4483 x 10⁻⁸</td>
<td>3.4688 x 10⁻⁹</td>
</tr>
<tr>
<td>C₅</td>
<td>-1.4784 x 10⁻⁴</td>
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</tr>
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<td>A₆</td>
<td>1.0575 x 10⁸</td>
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<td>B₆</td>
<td>-9.472 x 10⁻⁴</td>
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<tr>
<td>C₆</td>
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<td>0.0</td>
</tr>
<tr>
<td></td>
<td>R-11</td>
<td>R-12</td>
</tr>
<tr>
<td>----------</td>
<td>------</td>
<td>------</td>
</tr>
<tr>
<td>K</td>
<td>4.5</td>
<td>5.475</td>
</tr>
<tr>
<td>α</td>
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<td>0.0</td>
</tr>
<tr>
<td>C&lt;sup&gt;-&lt;/sup&gt;</td>
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<td>0.0</td>
</tr>
<tr>
<td>X</td>
<td>50.5418</td>
<td>39.5566</td>
</tr>
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<td>Y</td>
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Coefficients for Equations (7) and (26)

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<td>29.3575</td>
<td>33.0655</td>
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<td>B</td>
<td>-4344.344</td>
<td>-3436.6322</td>
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<tr>
<td>D</td>
<td>4.0084 x 10&lt;sup&gt;-3&lt;/sup&gt;</td>
<td>4.7305 x 10&lt;sup&gt;-3&lt;/sup&gt;</td>
<td>3.985 x 10&lt;sup&gt;-3&lt;/sup&gt;</td>
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<td>E</td>
<td>0.03136</td>
<td>0.0</td>
<td>0.0</td>
<td>0.44574</td>
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<td></td>
<td>R-11</td>
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<td>R-22</td>
<td>R-113</td>
<td>R-114</td>
<td>NH₃</td>
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<td>-------</td>
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</tr>
<tr>
<td>AL</td>
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<td>-.0128</td>
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<td>6.0268</td>
<td>0.0</td>
<td>-22.2926</td>
<td>0.0</td>
<td>6.7183</td>
<td>-23.456</td>
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<td>36.7066</td>
<td>-.6555</td>
<td>0.0</td>
<td>20.4733</td>
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APPENDIX II

BINARY FLUID CYCLE COMPUTER PROGRAM
SUBROUTINE DBINRY
COMMON /NAMLST/ LL1, LL2, LL3, LL4, LL5, LL6,
* LL7, LL8, LL9, LL10, LL11, LL12, NUMTF, ITEMS,
* ACRES, PDPHTH, DINJW, CNWCL, FINTPW, RWIA, PF, WDEPTH,
* DEPTH, AMWEPW, AVGWL, WM3S, PWTEMP, PREST, DIA, FLOOR,
* ODS, PM, CAC0, WELPRS, CASFPC, PRORAT, NIHTYP, WELS,
* PVTWTH, SIMFRC, FRCNRM, FRCEPW, NTC, NC, IPLANT, PSALVG,
*NFLAS, PGCOND, GRK1, WBAT, DHR, B, RI, TCI,
*TQINT, TCNAFT, TCO, TLODG, TOCMC, GASEJ, CINLAB, INJW,
*INJP, RINJW, PLINJW, PINJP, WP2S, WPC02, WPCH4, WPONC0G, TOTNCG,
*STSEPR, DMFACT, IJDPUMP, WM2PS, WP3, WPCH4, WPONC0G, TOTNCG,
*IGFOP, MLTHSL, NLTHSL, LTHPS, IDPHUJ, DHIEM, IDPRPNT, IDPRPNT,
*CHARAT, STATUS, SLNCS, IURIGS, IPUNCH, DRPLPR,
*IAVEL, DCWPI, DCPW, DLTNWW, GROUP(50), TMTS(50),
*LAGYM(2502), UNIT(50), FT(50), D(35),
*PERCNT(2), UNIT(50), STRATA(2410), TYPE(10),
*INPUT(701), LAGS(2502), EVALUE(2),
*EQUIVALENCE (PWEMYDM0(8)), (YRS, 0(6)), (BND, 0(4)),
* (PRED0(5)), (RE0(6)), (VOU, 0(10)), (ROYPAY, 0(27)),
*WDEPTU, PWCH4, EQUIVALENCE (FRCWPW, WPNP),
*EQUIVALENCE (AVGWL, WFLIFE), (WELPRS, WPNP),
*EQUIVALENCE (FRCNPWW, WPNP),
*EQUIVALENCE (AVGWL, WFLIFE), (WELPRS, WPNP),
*EQUIVALENCE (AVGWL, WFLIFE), (WELPRS, WPNP),
*EQUIVALENCE (MINLUT, PXNG),
*INTEG3O GROUP, GASEJ, STSEPR,
*REAL N,
*COMMON /FLASH/, TF(5), PPFI(5), HFG(5), X(5), WH, TCOND,
*MCIND, NPNW, JFPLAS, WAKBP, TBIN, PIN, SPBIN, RFFACT,
*COMMON /STEAM/, T, PM, WP, Q, QT, QS,
*QSI(5), QMI, TNHR(4), TKWO, TFWL05, QWSS, SSW,
*NPIN, QNA,
*COMMON /INTRAN/ FLOW, TM2D, WEXTRA, FLOWIN, PAVAIL, PETEPM,
*REPRES, PELFLO, IEFFL, HRATE(6), NTPR, OMPK, SDHDEP,
*ISALT,
*COMMON /OUTRAN/ NTRANS, TBCST, DCCOST, DMCOST, OMCOST, TCCAP,
*OCAPB, AVFLOW(6), TIN(6), PINA, HIN(6), FIN(6), DFLOW(2),
*NPW, NW, NEXW(2), WACPE, PL, ARCP0, BPKLOS, PCC0ST,
*TEQUIP(10), LTYPE(6), LTPY, DTM(2), DNPY(2),
*DFIN(2), NTRAN, NSTATE, SQUAL, PWTAW, PPWAT, PHWAT,
*RCOST,
*COMMON /POINT/, MU, KU, LU, NOD0E, NNAW, LNAWP,
*LSNW, LSW0, LUSW, LSHW, LFW, LTN, LPN, LYN,
*LON, LUP, LTHK, LEA,
*COMMON /FLAGS/, FLAG, NEXT, FLAG1,
*INTGE5O FLAG, NINTARY, SUPER, NPLUID, WT1, WT2, WT3, WT4,
*ESTWT4, WT1, WP3, CT1, CWT2, CWPI, CWP2, T1,
*FIBEST, T2SATL, T3, T4, T5, P1, P2,
*P3, P4, H1, H2, HPST, H2SATL, H3,
*H4, H5, S1, SEST, S3, S4, SVSATL, 01,
*02, 03, EEST, V1, EEST, PINCHP, WELD, DIAL, DIA0L,
*WVEL, DIAV, DIAV, WELS, DIAS, DIAOS, WELS, DIASC,
*DIAONS, PVELC, DIAC, DIOAC, FOULFI, FOULFG, BAR, QISO,
*HTAPX(1), HAPON, TAPPCH, WFOUR, APPCON, APPT, APMHX,
*DELPHX, PINSUB, PINSUP, PUNEP, RANGE,
*LOGICAL SUPER,
*IF (FLAG.EQ.0) GO TO 110
SUBROUTINE DBINARY

CALL PAGE(0)
WRITE(*,100)
FORMAT("BINARY POWER PLANT SYSTEM AND TRANSMISSION MODEL")
IF(FLAG.EQ.1) RETURN
C INITIALIZE VARIABLES
100 PAVAIL = JAN*3*PA*PF
PKLOS = 0.0
NTITER = 0
PFACT = PFACT
MD=PS
KU=PS
LD=1
TWHO = DEW(PWFACT)
TWHO = TWHO*1.0 + 2.
C SET GEOTHERMAL FLUID PRESSURE AT WELLHEAD, INCLUDING ANY OVERPRESSURIZATION
C COMPUTE SATURATION PRESSURE OF GEOTHERMAL FLUID
PSAT = PSATT(TWHO)
C SAVE SATURATION PRESSURE PRIOR TO OVERPRESSURIZATION
SPHOD = PSAT
SPHIN = PSAT
PWHOD = PWFACT * PSAT
PWHIN = PWHOD
C COMPUTE ENTHALPY OF GEOTHERMAL FLUID AT SATURATION
IF(SPWHOD-PSAT).LT.0.101
WH = HPFU(SPWHOD,TWHO)
GO TO 6
6 WH = HPF(TWHO)*(1.-STFRAC) + HGT(TWHO)*STFRAC
GO TO 7
7 WH = HPFL(SPWHOD,TWHO)
C CONTINUE
C COMPUTE ENTHALPY OF GEOTHERMAL FLUID AT OVERPRESSURIZATION
IF(PWHOD-PSAT).LT.0.101
WH = HPFU(PWHOD,TWHO)
GO TO 12
12 WH = HPF(TWHO)*(1.-STFRAC) + HGT(TWHO)*STFRAC
GO TO 12
12 CONTINUE
WP = PWHOD
TIN(1) = TWHO
PIN(1) = PWHOD
C 199 NCOUNT = 0
GOLO = 0.0
200 CONTINUE
NCOUNT = NCOUNT + 1
NTITER = NCOUNT
IF(NCOUNT.EQ.11) GO TO 22
CALL BINARY
C 20 NTITER = NTITER + 1
CALL TTRANS(MD, KU, LD)
IF(NEXT.NE.1) GO TO 999
IF(PFACT.LE.1.0) GO TO 22
C MINIMIZE OVERPRESSURIZATION REQUIRED FOR FLUID LINE TO PLANT
SUBROUTINE DBINRY

IF (NTITER.GT.1) GO TO 210
PSAT = PSATT(TIN(NTRANS))
DELP = P(NTRANS) - PSAT
C SET OVERPRESSURIZATION TO ALLOW FOR GEOTHERMAL FLUID PRESSURE
C DROP ACROSS HEAT EXCHANGERS
DELPOV = DELPHI * 10
PWMO = PWMO - DELTP * DELPOV
PWMO = PWMO / SPWMO
PRES = SPWMO
PRES1 = PWMO

2090 IF (FLAN.GE.0) WRITE (6, 2091) PRES, PRFAC, PFACT, PRES
2091 FORMAT (1X, 'MINIMIZATION OF WATER OVERPRESSURIZATION***/
*1AX,'ORIGINAL WATER PRESSURE' =", 1PE15.6/
*1AX,'REDUCED PRESSURE FACTOR' =", 1PE15.6/
*1AX,'OVERPRESSURIZED WATER PRESSURE' =", 1PE15.6/
*"***RESIMULATE TRANSMISSION MODEL***/
Go TO 20

2100 CONTINUE
CALL CONVNG (OLD*QNA + ICONV)
OLD*QNA
GO TO (200, 240, 250) + ICONV

220 WRITE (6, 221)
221 FORMAT (1X, 'PLANT DRIVER FAILED TO CONVERGE AFTER 10 ITERATIONS***/
*1AX,'GO TO NEXT CASE*****'
NEXT = 3
GO TO 209

C SET FLAG INDICATING NEXT ITERATION WILL BE THE FINAL ITERATION
240 ICONV = 2
FLAG = LL6
GO TO 200
C POWERPLANT-TRANSMISSION ITERATIONS COMPLETED
250 CONTINUE
IF (FLAN .LE. 0) WRITE (6, 251) NCOUNT
251 FORMAT (1X, 'NUMBER OF POWERPLANT - FLUID TRANSMISSION ITERATIONS***/
1 T6A, 12)

PMTH = FLORAT(WHO+HFT(C*N1))/3414.43EJ
CALL INJECT (MD, KD, LD)

999 RETURN
END
SUBROUTINE CONVRG

COMMON /FLGS/ FLAG, NEXT, FLAG1

INTEGRAL FLAG, FLAG1

DATA PCONV, GDLO, GPLANT, ICONV

C TEST FOR POWERPLANT-TRANSMISSION CONVERGENCE BASED UPON NEWLY
C IERATED POWERPLANT GEOTHERMAL FLUID DEMAND RATE ARISING FROM
C NEWLY ITERATED TRANSMISSION FLUID DEGRADATION

DIF = GPLANT - GDLO

PCONV = ABS(DIF)/GPLANT*100.

IF (FLAG.NF.0) WRITE (6,10) GDLO, GPLANT, DIF, PCONV

10 FORMAT (1X "POWERPLANT - FLUID TRANSMISSION"
CONVERGENCE CRITERION"

* RATE DEMANDED BY PLANT (LMH/HR)"+TAB+"PERCENT")"

* CURRENT GEOTHERMAL FLOW RATE DEMANDED BY PLANT (LMH/HR)"

* DIFFERENCE (PERCENT)"

* CONVERGENCE CRITERION (PERCENT)"

* PCONV*100)

IF (PCONV.GT.0) WRITE (6,20)

20 FORMAT (1X "POWERPLANT AND TRANSMISSION MODELS SATISFY"
CONVERGENCE CRITERION"

* POWERPLANT AND FLUID TRANSMISSION MODELS SATISFY CONVE

* PCONV*.0) WRITE (6,60)

60 FORMAT (1X "POWERPLANT AND FLUID TRANSMISSION MODELS SATISFY CONVE

* PCONV*.0) WRITE (6,60)

60 FORMAT (1X "POWERPLANT AND FLUID TRANSMISSION MODELS SATISFY CONVE

* PCONV*.0) WRITE (6,60)

60 FORMAT (1X "POWERPLANT AND FLUID TRANSMISSION MODELS SATISFY CONVE

* PCONV*.0) WRITE (6,60)

60 FORMAT (1X "POWERPLANT AND FLUID TRANSMISSION MODELS SATISFY CONVE
SUBROUTINE ACWFRT

SUBROUTINE ACWFRT(QCOC, QCOC, TCI, TCOOC, TCOHC, GPKW)
COMMON /FLAGS/ FLAG, NEXT, FLAG
INTEGER FLAG, FLAG

ROUTINE CALCULATES THE AUXILIARY COOLING WATER FLOW RATES

OUTPUT VALUES
QCOC = TURBINE OIL COOLER COOLING WATER FLOW RATE
QCOC = COOLING WATER FLOW RATE OF THE HYDROGEN COOLER

INPUT VALUES
TCI = COOLING WATER INLET TEMPERATURE
TCOOC = OIL COOLER EXIT COOLING WATER TEMPERATURE
TCOHC = HYDROGEN COOLER EXIT COOLING WATER TEMPERATURE
GPKW = GROSS KILOWATT

OIL COOLER
TOCKW = GPKW * N*005
QCOC = QCQC(TOCKW, TCI, TCOOC)

HYDROGEN COOLER
HCKW = GPKW * N*019
QCOC = QCQC(HCKW, TCI, TCOHC)

IF(FLAG.EQ.0) RETURN
WRITE(*,1) TCI, TCOOC, QCOC
WRITE(*,2) TCI, TCOHC, QCOC
RETURN

1 FORMAT(T60, "Cooling Water")
2 FORMAT(T10, "inlet Temperature (Deg. F)")
3 FORMAT(T10, "Outlet Temperature (Deg. F)")
4 FORMAT(T10, "flow Rate (lb/hr)")
5 FORMAT(T10, "Hydrogen Cooler")
END
FUNCTION AVGTMP

FUNCTION AVGTHP(TMP1, TMP2)

C
C FUNCTION TO AVERAGE TWO TEMPERATURES
C
C AVGTHP = (TMP1 + TMP2) / 2.0
C
RETURN
END
SUBROUTINE BFT2D2

SUBROUTINE BFT2D2(X, N, K, P, H)
COMMON /FLAGS/ FLAG, NEXT, FLAG1
INTEGER FLAG, FLAG1
DIMENSION X(N), R(K)
C***COMON FOR THE MUTAF CONSTANTS
COMMON /RUTANF/ AR, AZERO, AH1, AH2, AR4, HH, AZERO, H1
= BA2, BA3, ABA, CZERO, DZERO, EZERO, SHA, SRRB, SM, 50, ALPHA,
= GAMMA, PMI, BUTMW, RH
COMMON /RWNCOM/ TERM1, TERM2, SAVT(V), JFLAG, DEL(2), OLD1
R(1)=PSeST(X(1); X(2); TERM1)
R(2)=SeST(X(1); X(2); / PMI * SIEFAL(X(1)) - TERM2
IF (FLAG.EQ.0) PRINT 100
1 FORMAT ('**BFT2D2 RESIDUALS=', 1PE14.6, 34X, 'TV=', 1PE14.6)
RETURN
END
SUBROUTINE BFT2V2(X, N, K, R, P)
C**DETERMINE T2 AND V2
DIMENSION X(2), R(2)
C**COMMON FOR THE MUTANE CONSTANTS
COMMON /AUTANE/ AB, AZERO, AB1, AB2, AB3, AAR, RAR, AZERO, RAR,
* ABP, AB3, RAP, CZERO, DZERO, EZERO, SMA, SMAB, SMC, SM0, ALPMA,
* GAMMA, PMI, BTHI, RH
COMMON /MUTMOM/ TERM1, TERM2, SAVTV(2), JFLAG, DEL(2), OLDR
R(1) = POST(X(1), X(2), TERM1)
R(2) = WPOST(X(1), X(2))/PMI + HIDEAL(X(1)) + TERM2
RETURN
END
SUBROUTINE BINARY

THIS IS THE MAIN SUBROUTINE FOR THE BINARY FLUID CYCLE CALCULATIONS FOR THE GEOOTHERMAL POWER PLANT

REAL IDEF,

COMMON /NAMST/ LL1, LL2, LL3, LL4, LL5, LL6,
*LL7, LL8, LL9, LL10, LL11, LL12, NUMFTF, ITEMS,
*ACRES, PWFTH, DINJW, CMWC, FIUW, PWMDIA, PF, WDEPTH,
*DEPTH, DTFFPW, AVGWL, WPDS, PWTMP, PFST, UTA, FLORAT,
*PS, PH, CACO3, WELPRES, CASFRC, PRPRT, NUMTYP, WELSPC,
*PMWH, STFRC, FRCPNW, PFCNPW, NYC, NC, IPLANT, PSALVG,
*NFLASH, PCOND, GRKX, W0AT, D0AT, B, PI, TCI,
*TCOND, TCONAFT, TCO, TLCOC, TCHOC, GASEJ, CINLAR, INJW,
*INJU, RINJW, PLINJP, PLINJU, ISCRUR, MTX, PCTS, FLASH,
*STSEP, WPFACT, IJUMP, WMPCS, WPCO2, WPCMA, WPONG, TOTNCG,
*IGEOF, MTHSL, DMLHTM, LIPLAS, IOPNPU, DDEP, IORDIL, IORDPT,
*CHARAT, STATUS, SLNCST, IURGTS, IPUNCH, DRILPR,
*IAYNEL, DCPW, DCNPW, DLCINJW, GROUP(SL), ITEX(SL),
*LAGTYM(257), UNIT(40), FTF(40), D(35),
*PERCT(2), UNIT0(50), STRATA(10), TYPE(10),
*DINPUT(77), LAGS(5,2), EVALUE(2),
*EQUIVALENCE (PWW+DMM+0(8)), (YRSD+2(2)), (AND+D(4)),
*(DDJ0+0(5)), (DDJ0+1(10)), (AND+D(27)),
*(WDEPTH+PWRLEN),
*EQUIVALENCE (FRCPNW, NR), (AVGWL+WLFIE), (WELPRES+PWHO),
*EQUIVALENCE (D128+TMaint), (D129+UMaint), (WELSPC+WSP),
EQUIVALENCE (STFRC+STMFRL),
INTEGER GROUP, GASEJ, STSEP,
REAL X, X0,
COMMON /FLASH/ T(5), PFF(5), HFG(5), IX(5), WH, TCOND,
*MCNDF, NPLOW, NLFWL, SPMH, TMN, PIN, SPAB, RPFACT,
COMMON /STEAM/ T, P, CT, Q, QT, Q0,
*OS1(6), QMT, TMC(6), RTW0, TW0, QWSS, SSQW,
*NPITER, QNA,
COMMON /INTRAN/ FLOW, T=0, WEXTRA, FLOWIN, PAVAL, PETEMP,
*PEPRES, PELOW, IEFL(16), NTPITER, DPKW, SMDUP,
*ISALT,
COMMON /OUTH/ VTRANS, TCAST, UCCOST, TMCOST, DCOST, TCAV,
*CAPR, AVFLOW(6), TIN(6), PIN(6), MIN(6), FIN(6), DFLOW(2),
*NPW, NUM, N1W2(2), WACRE, PL, BPACOST, BPKLOS, PCCOST,
*TEQUIP(10), LTYPE(6), LTYP, DINT(2), OPIN(2),
*DFIN(2), NTRAN, NTDATE, SQUAL, PWTW, PPAW, PHWAT,
*RJCOST,
COMMON /FLAGS/ FLAG, NEXT, FLAG1,
INTEGER FLAG, COMMON /RINARY/ SUPER, NPFLUID, WT1, WT2, WT3, WT4,
*ESTUTA, WPI, WP3, CWT1, CWT2, CWP1, CWP2, T1,
*T2, T2EST, T2SATL, T3, T4, T5, P1, P2,
P3, P4, P5, H1, H2, H2EST, H2SAT1, H3,
*H4, H5, SCh, SCh, SCh, SCh, SCh, SCh, SCh, SCh, SCh,
*O2, O2EST, V1, VEFS, PINP, WVEL, DIAT, DIAT, DIAT,
*WVEL, DIAV, DIAC, DIAO, FOULFI, FOULFO, RAR, O150,
*HTAE(13), HTAON, T=PRCH, WFOIR, APPCON, APPCT, APPH
SUBROUTINE BINARY

*DELPMX, PINSUB, PINSUP, PUNEP, RANGE
LOGICAL SUPER
COMMON /WFFRT, REFF /, REFF, OEFF
COMMON /XOUT /, XSLT(I + J)

DESCRIPTION OF COMMON BINARY ELEMENTS

SUPER - LOGICAL VARIABLE SET TO "TRUE" IF SUPER-CRITICAL CYCLE
GRKW - GROSS KILOWATT OUTPUT OF PLANT
NFLI1D - INDEX OF WORKING FLUID BEING USED
WT1 - WATER TEMPERATURE AT INLET TO SUPER HEATER
WT3 - WATER TEMPERATURE AT INLET TO PREHEATER
WT4 - WATER TEMPERATURE AT OUTLET FROM PREHEATER
WP1 - WATER PRESSURE AT INLET TO SUPER HEATER
WP3 - WATER PRESSURE AT INLET TO PREHEATER
CWT1 - COOLING WATER INLET TEMPERATURE
CWT2 - COOLING WATER OUTLET TEMPERATURE
CWP1 - COOLING WATER INLET PRESSURE
CWP2 - COOLING WATER OUTLET PRESSURE
TCOC - OIL COOLER EXIT COOLING WATER TEMPERATURE
TCOC - HYDROGEN COOLER EXIT COOLING WATER TEMPERATURE
T1 - WORKING FLUID TEMPERATURE AT TURBINE INLET
T2 - WORKING FLUID TEMPERATURE AT TURBINE INLET
T3 - WORKING FLUID TEMPERATURE AT PREHEATER INLET
T4 - WORKING FLUID TEMPERATURE AT VAPORIZER INLET
T5 - WORKING FLUID TEMPERATURE AT SUPER HEATER INLET
P1 - WORKING FLUID PRESSURE AT TURBINE INLET
P2 - WORKING FLUID PRESSURE AT TURBINE EXHAUST
P3 - WORKING FLUID PRESSURE AT PREHEATER INLET
P4 - WORKING FLUID PRESSURE AT VAPORIZER INLET
P5 - WORKING FLUID PRESSURE AT SUPER HEATER INLET
H1 - WORKING FLUID ENTHALPY AT TURBINE INLET
H2 - WORKING FLUID ENTHALPY AT TURBINE EXHAUST
H3 - WORKING FLUID ENTHALPY AT PREHEATER INLET
H4 - WORKING FLUID ENTHALPY AT VAPORIZER INLET
H5 - WORKING FLUID ENTHALPY AT SUPER HEATER INLET
S3 - WORKING FLUID ENTROPY AT PREHEATER INLET
S4 - WORKING FLUID ENTROPY AT VAPORIZER INLET
T2ES - WORKING FLUID TEMPERATURE AT SATURATION AND P2
H2ES - WORKING FLUID ENTHALPY AT TURBINE EXHAUST (ESTIMATED)
S2ES - WORKING FLUID ENTROPY AT TURBINE EXHAUST (ESTIMATED)
H2SATL - ENTHALPY OF WORKING FLUID AS A SATURATED LIQUID AT P2
SVSATL - SPECIFIC VOLUME OF THE SATURATED LIQUID AT P2
PINF - DIFFERENCE IN INLET WATER AND OUTLET WORKING FLUID
      TEMPERATURES OF LIQUID HEAT EXCHANGER
WVELL - WATER VELOCITY IN THE LIQUID HEAT EXCHANGER (PREHEATER)
DIALL - INSIDE TUBE DIAMETER IN PREHEATER
DIANL - OUTSIDE TUBE DIAMETER IN PREHEATER
WVELV - WATER VELOCITY IN VAPORIZER
DIAV - INSIDE TUBE DIAMETER IN VAPORIZER
DIANV - OUTSIDE TUBE DIAMETER IN VAPORIZER
WVELS - WATER VELOCITY IN SUPER HEATER
DIAS - INSIDE TUBE DIAMETER IN SUPER HEATER
DIANs - OUTSIDE TUBE DIAMETER IN SUPER HEATER
WVELS - WATER VELOCITY IN SUPER-CYCLE HEAT EXCHANGER
DIACS - INSIDE TUBE DIAMETER IN SUPER-CYCLE HEAT EXCHANGER
DIANSC - OUTSIDE TUBE DIAMETER IN SUPER-CYCLE HEAT EXCHANGER
SUBROUTINE BINARY

C

VE:IC - VELOCITY OF COOLING WATER IN CONDENSER
DIA: C - INSIDE TUBE DIAMETER IN CONDENSER
DIAC - OUTSIDE TUBE DIAMETER IN CONDENSER
FOU:FI - INSIDE FOULING FACTOR FOR HEAT EXCHANGER
FOU:FO - OUTSIDE FOULING FACTOR FOR HEAT EXCHANGER
WBAT = WFT RUBL AIR TEMPERATURE
DBAT = DRY BUBL AIR TEMPERATURE
BAR = BAROMETRIC PRESSURE
RI = HUMIDITY RATIO
WP = PRESSURE OF GEOHERMAL FLUID AT THE GROUND

C

DIMENSION FLU(10)
REAL KWLRP, KWLFP

C

DATA CON / 5.4233E-5 /
C

DATA(FIU(1),#*2+10)/"ISOBUTANE","N-RUTANE","R-11","R-12","R-21","R-22","R-113","R-114","AMMONIA"/
WTI = TIN(1)
WP1 = DIN(1)
CALL ODBIN

C

C***SUBROUTINE SETCON DETERMINE IF SUBROUTINE BUTANE OR FREON SHOULD BE
C***CALLED AND MAKES THE TRANSFER
C

CALL SETCON(NFLUID)

C

C

PRINT DESIGNATED VALUES
C

IF(FLAG .EQ. 0) GO TO 200
WRITE(4+1) FLU(NFLUID)
IF(.NOT. SUPER) WRITE(6,2)
IF(SUPER) WRITE(6,3)
200 CONTINUE

C

C DETERMINE GEOHERMAL FLUID FLOW RATE
C

C

C DETERMINE GEOHERMAL WATER FLOW RATE
WT = EBT4
IF(SUPER) WT = BT4
T = AVT4P(WT1 + WT)
CPET = FLUID(1 + 1 + T, "BINARY AT CPEST CALC")
Q = GICO * (M1-H3) / CPES1 / (WT1-WT)
QNA = Q/1. - WPD9/100.
WP2 = WP3 = DLPNX/3.
WP4 = WP3 = DLPNX

C

C DETERMINE AUXILIARY COOLING WATER FLOW RATE
C

CALL AWFHT(GCOC, GCHC+CW1, TCOOC, TCOMC, GRKW)
SUBROUTINE BINARY

HEAT EXCHANGER CHARACTERISTICS

SUB-CRITICAL CYCLE

IF(SUPR) GO TO 280
CALL SUBHEX(W1, W2, T1, T2, T3, T4, T5, P1, P2, P3, P4, P5, FOUFL, VELV, DIAL, DIAUL, WVELS, DIAOS, NPLUID, H1AEX, P1, GMA, WPDS)

IF(FLA2, EQ. 0) GO TO 370
WRITE(A4) APH, T1, T2EST, T2, T2SATL
WRITE(A4) T3, T4, T5
WRITE(A5) P1, P2, P3, P4, P5
WRITE(A6) V1, V2EST
WRITE(A7) H1, H2, H2EST
WRITE(A7) H3, H4, H5
WRITE(A8) S1, S2EST
WRITE(A9) W1, W2, W3, W4, ESTWT4, DELPHX
WRITE(A10) IMEFF, REFF, VEFF
WRITE(A11)
WRITE(A12)
WRITE(A13) T5, T4, T3
WRITE(A14) P5, P4, P3
WRITE(A15) H5, H4, H3
WRITE(A16) T1, T5, T4
WRITE(A17) P1, P5, P3
WRITE(A18) H1, H5, H4
WRITE(A19)
WRITE(A20) W1, W2, W3
WRITE(A21) WP1, WP2, WP3
WRITE(A22) W2, W3, ESTWT4
WRITE(A23) WP2, WP3, WP4
GO TO 370

280 CONTINUE

SUPER CRITICAL CYCLE

CALL SUBHEX(W1, W2, W3, T1, T2, T3, T4, T5, P1, P2, P3, P4, P5, FOUFL, VELV, DIAL, DIAUL, WVELS, DIAOS, NPLUID, H1AEX, P1, GMA, WPDS, PINSUP)

IF(FLA2, EQ. 0) GO TO 370
WRITE(A4) APH, T1, T2EST, T2, T2SATL
WRITE(A4) T3
WRITE(A5) P1, P2
WRITE(A5) P3
WRITE(A6) V1, V2EST
WRITE(A7) H1, H2, H2EST
WRITE(A7) H3
WRITE(A8) S1, S2EST
WRITE(A9) W1, W2, W3, DELPHX
WRITE(A10) IMEFF, REFF, VEFF
WRITE(A11)
WRITE(A12)
DETERMINATION OF COOLING TOWER CHARACTERISTICS

CALL CHONSR(OCWAT, CWGPM, CWT1, CWT2, M2SATL, T2, T2SATL, P2, NFLUID, *
* MVELC, DIAVC, DIAVC, FOULFI, FOULFO, QISO, HTACON, M2SATV, *
* CWT12, CWP12, CWP12, CWP2, CWP2)

IF(FLAN.EQ.0) GO TO 40

WRITE (*, 11) T2, T2SATL
WRITE (*, 14) P2, P2
WRITE (*, 15) H2, M2SATV
WRITE (*, 16) T2SATL, T2SATL
WRITE (*, 17) P2, PC
WRITE (*, 18) M2SATV, M3
WRITE (*, 19) CWT12, CWT1
WRITE (*, 20) CWP12, CWP1
WRITE (*, 21) CWT2, CWT12
WRITE (*, 22) CWP2, CWP12
WRITE (*, 23) MXRSLT(1, 1) + MXRSLT(1, 2)
WRITE (*, 24) MXRSLT(2, 1) + MXRSLT(2, 2)
WRITE (*, 25) MXRSLT(3, 1) + MXRSLT(3, 2)
WRITE (*, 26) MXRSLT(4, 1) + MXRSLT(4, 2)
WRITE (*, 27) MXRSLT(5, 1) + MXRSLT(5, 2)
WRITE (*, 28) MXRSLT(6, 1) + MXRSLT(6, 2)
WRITE (*, 29) MXRSLT(7, 1) + MXRSLT(7, 2)
WRITE (*, 30) MXRSLT(8, 1) + MXRSLT(8, 2)
WRITE (*, 31) MXRSLT(9, 1) + MXRSLT(9, 2)

CONTINUE
SUBROUTINE BINARY

QC = TOTAL COOLING WATER FLOW RATE
    QC = QCWAT + QCOC + QCCH

TQC = TEMPERATURE OF QC
TQC = (QCWAT * CWT2 + QCOC * TCOC + QCCH * TCHC) / QC

CALL TOWER(CTA, QLA, QC, TQC, BAT, DBAT, BAR, RI, CWT1, APPCT)

IF(FLAG .NE. 0) WRITE(6, 31) QCOC, QCCH, APPCON, RANGE, QCWAT, QLA

CALCULATIONS OF GENERAL ELECTRICAL LOSSES

QRI = REINJECTION RATE
ALL OF THE GEOTHERMAL FLUID IS REINJECTED

QRI = QNA
CALL LOSSES(TKWLOS, CTA, QC, QC, QRI, SPWHDF, IJPUMP, BKLOS, NWP, DHKPW, **, FCPW)

ELECTRICAL LOSSES TO OTHER EQUIPMENT

DEL = FLUID(3 * NFLUID + T3, "BINARY AT DEL CALC")
DEL = DEL * (1.007566 * WPDS + 1.0)

KWLBP = WORKING FLUID BOOSTER PUMPS

PDELTP = 20.0
KWLBP = QISO * CON * PDELTP / DEL / 0.85

KWLFP = WORKING FLUID FEED PUMPS

PDELTP = (P3 - P2 - 20.0)
KWLFP = QISO * CON * PDELTP / DEL / 0.85

NET HEAT RATE

WTWHD = FLUID(1 + NFLUID + T1, "BINARY AT TNHR CALC")
WCWT1 = FLUID(1 + NFLUID + T1, "BINARY AT TNHR CALC")
TNHR(I) = Q * (WTWHD - WT1) - CWT1 / (GHK + KWLBP + KWLFP + TKWLOS)
HRATE(I) = TNHR(I)

TKWLOS = TKWLOS + KWLBP + KWLFP
ANKWO = GHK + TKWLOS
IF(FLAG .EQ. 0) GO TO 450
WRITE(4, 32) KWLBP, KWLFP
WRITE(4, 33) GHK, QNA, FLUID(NFLUID), QISO + QC, TKWLOS, ANKWO, TNHR(I)
CONTINUE

PEFLOW = QRI / 3600.
PETEMP = ESTWT4
IF(SUPER) PETEMP = WT4
P EPRES = MP4

OVERPRESSURIZE SPENT GEOTHERMAL FLUID FLOW PLANT HEAT EXCHANGERS
FOR REINJECTION IF IJPUMP IS NONZERO
IF(IJPUMP .NE. 0) PEPRES = PEPRES + 40.

144
SUBROUTINE BINARY

RETURN
1 FORMAT("THE WORKING FLUID IS " / "A9")
2 FORMAT("THE CYCLE IS SUB-CRITICAL")
3 FORMAT("THE CYCLE IS SUPERCRITICAL")
4 FORMAT("OFINAL CYCLE CONDITIONS" / " WORKING FLUID CONDITIONS")
  1 T60"TEMPERATURES (DEG. F)"
    * T10"APRX. HEAT EXCHANGER APPROX." T40,1PE14.6
    * T20"T1-TURBINE INLET" T40,1PE14.6
    3 T10"T2-TURBINE OUTLET (IDEAL)" T40,1PE14.6
    4 T10"T2-TURBINE OUTLET (ACTUAL)" T40,1PE14.6
    5 T10"T2-CONDENSER OUTLET" T40,1PE14.6
41 FORMAT(T10"T3-PREHEATER INLET" T40,1PE14.6
        * T10"T4-VAPORIZER INLET" T40,1PE14.6
        * T10"T5-SUPERHEATER INLET" T40,1PE14.6
42 FORMAT(T10"T3-HEAT EXCHANGER INLET" T40,1PE14.6
  5 FORMAT(T60"PRESSURES (PSIA)"
    1 T10"T1-TURBINE INLET" T40,1PE14.6
    2 T10"T2-TURBINE EXHAUST" T40,1PE14.6
51 FORMAT(T10"T3-PREHEATER INLET" T40,1PE14.6
        * T10"T4-VAPORIZER INLET" T40,1PE14.6
        * T10"T5-SUPERHEATER INLET" T40,1PE14.6
52 FORMAT(T10"T3-HEAT EXCHANGER INLET" T40,1PE14.6
  6 FORMAT(T60"SPECIFIC VOLUMES (CU. FT./LB)"
    1 T10"V1-TURBINE INLET" T40,1PE14.6
    2 T10"V2-TURBINE OUTLET (IDEAL)" T40,1PE14.6
7 FORMAT(T60"ENTHALPY (BTU/LB)"
    1 T10"H1-TURBINE INLET" T40,1PE14.6
    2 T10"H2-TURBINE EXHAUST" T40,1PE14.6
    3 T10"H2-TURBINE OUTLET (IDEAL)" T40,1PE14.6
71 FORMAT(T10"H3-PREHEATER INLET" T40,1PE14.6
        * T10"H4-VAPORIZER INLET" T40,1PE14.6
        * T10"H5-SUPERHEATER INLET" T40,1PE14.6
72 FORMAT(T10"H3-HEAT EXCHANGER INLET" T40,1PE14.6
  8 FORMAT(T60"ENTROPY (BTU/LB/DEG. F)"
    1 T10"S1-TURBINE INLET" T40,1PE14.6
    2 T10"S2-TURBINE OUTLET (IDEAL)" T40,1PE14.6
9 FORMAT("JOGEOTHERMAL FLUID CONDITIONS" / " WATER TEMPERATURES (DEG F)"
    1 T10"WT1-SUPERHEATER INLET" T40,1PE14.6
    2 T10"WT2-VAPORIZER INLET" T40,1PE14.6
    3 T10"WT3-PREHEATER INLET" T40,1PE14.6
    5 T10"WT4-PREHEATER EXIT (RESET)" T40,1PE14.6
    6 T10"WT4-PREHEATER EXIT (ACTUAL)" T40,1PE14.6
    7 T10"MELPHX-TOTAL PRESSURE DROP" T40,1PE14.6
91 FORMAT("JOGEOTHERMAL FLUID CONDITIONS" / " WATER TEMPERATURES (DEG F)"
        * T10"WT1-HEAT EXCHANGER INLET" T40,1PE14.6
        * T10"WT4-HEAT EXCHANGER EXIT" T40,1PE14.6
        * T10"MELPHX-TOTAL PRESSURE DROP" T40,1PE14.6
10 FORMAT("TURBINE EFFICIENCY"
    1 T10"IDEAL RANKINE EFFICIENCY" T40,1PE14.6
    2 T10"ACTUAL RANKINE EFFICIENCY" T40,0E14.6
    3 T10"OVERALL TURBINE EFFICIENCY" T40,0E14.6
11 FORMAT("WORKING FLUID" / "THERMAL EFFICIENCY"
    5X"SUB-CRITICAL CYCLE"
    1 " TEMPERATURE (DEG. F) = PRESSURE (PSIA) = ENTHALPY (BTU/LB)"
    2 T72"SUPER HEATER", T92"VAPORIZER", T113"PREHEATER" /
SUBROUTINE BINARY

50 FORMAT (T7I"DESUPERHEATER", T9I"CONDENSER", /" WORKING FLUID")
13 FORMAT (6X"INLET TEMPERATURE", T7I0.3('PE14.6X6X))
14 FORMAT (6X"INLET PRESSURE", T7I0.3('PE14.6X6X))
15 FORMAT (6X"INLET ENTHALPY", T7I0.3('PE14.6X6X))
16 FORMAT (6X"OUTLET TEMPERATURE", T7I0.3('PE14.6X6X))
17 FORMAT (6X"OUTLET PRESSURE", T7I0.3('PE14.6X6X))
18 FORMAT (6X"OUTLET ENTHALPY", T7I0.3('PE14.6X6X))
19 FORMAT (6X"COOLING WATER")
20 FORMAT (6X"HEAT EXCHANGER", 6X"SUPERCRITICAL CYCLE")
21 FORMAT (6X"TEMPERATURE (DEG. F) - PRESSURE (PSIA) - ENTHALPY (RTU/LB)")
22 FORMAT (6X"INSIDE TUBE DIAMETER (FT)", T7I0.3('PE14.6X6X))
23 FORMAT (6X"NUMBER OF TUBES", T7I0.3('PE14.6X6X))
24 FORMAT (6X"EQUIVALENT OUTSIDE DIAMETER (FT)", T7I0.3('PE14.6X6X))
25 FORMAT (6X"INSIDE HEAT TRANSFER COEFFICIENT (BTU/HR/SQ.FT./DEG. F")
26 FORMAT (6X"OUTSIDE HEAT TRANSFER COEFFICIENT (BTU/HR/SQ.FT./DEG. F")
27 FORMAT (6X"OVERALL HEAT TRANSFER COEFFICIENT (BTU/HR/SQ.FT./DEG. F")
28 FORMAT (6X"HEAT TRANSFER AREA (SQ.FT.)", T7I0.3('PE14.6X6X))
29 FORMAT (6X"HEAT TRANSFER LENGTH (FT)", T7I0.3('PE14.6X6X))
30 FORMAT (6X"MAIN CONDENSER")
31 FORMAT (6X"TEMPERATURE (DEG. F) - PRESSURE (PSIA) - ENTHALPY (RTU/LB)")
32 FORMAT (6X"CUMULATIVE OIL COOLER", T5I0.1('PE14.6X)/
33 FORMAT (6X"HYDROGEN COOLER", T5I0.1('PE14.6X)/
34 FORMAT (6X"APPROX. APPROACH TO CONDENSER (F)", T5I0.1('F14.0)/
35 FORMAT (6X"RANGE-COOLED WATER RANGE (F)", T5I0.1('F14.0)/
36 FORMAT (6X"MAKING FLOW RATE TO COOLING TOWER", T5I0.1('F14.0)/
37 FORMAT (6X"BOOSTER PUMPS", T5I0.1('PE14.6X)/
38 FORMAT (6X"FED PUMPS", T5I0.1('PE14.6X)/
39 FORMAT (6X"OVERALL RESULTS")
40 FORMAT (6X"NET KILOWATT OUTPUT OF THE PLANT", T5I0.1('PE14.6X)/
41 FORMAT (6X"TOTAL KILOWATT LOSSES", T5I0.1('PE14.6X)/
42 FORMAT (6X"TOTAL KILOWATT OUTPUT", T5I0.1('PE14.6X)/
43 FORMAT (6X"NET HEAT RATE (BTU/KW HR)", T5I0.1('PE14.6X)/
44 FORMAT (6X"OTHER THERMAL FLUID")
END
SUBROUTINE BUTANE
EXTERNAL BFT202, BFT2V2
COMMON /NAMLIST/ LL1, LL2, LL3, LL4, LL5, LL6,
*                 LL7, LL8, LL9, LL10, LL11, LL12, NUIMFTF, ITEMS,
*          ACRES, PWDPH, OINJP, CIANCL, FIWT, RWODIA, PF, WDEPTF,
*          DEPTH, AMWEPW, AVGWL, WPHS, PWTEMP, DFREST, DIA, FLORTA,
*          ODS, PH, CAC03, WELPRS, CASFR, PRDRT, NNUM, WELSPC,
*          PMWTH, STMGRP, FRCNPW, FMPMNPW, FMPMNYC, NC, NC, IPLANT, PSALVG,
*         FLASH, PCOND, GRW, WDET, DBAT, B, RI, TCI,
*          TCINT, TCFRT, TCQ, TCOCC, TCOHC, BASEJ, CINLAR, INJW,
*           INJW, RINJW, PLINJP, PLINJP, ISCRRU, MIX, PCTS, LFLASH,
*       STSEP, PMFACT, IMMRTMP, WPH2S, WBC02, WBC04, WPONCG, T0NCG,
*          IGEOP, WETHSL, DLMET, LIPMA, IDLPUM, DMEP, IDRLJ, IDRTNP,
*          CMARRAT, STATUS, SLMCS, IURIS, IPUNCH, DRILD,
*          AVWEL, DCPW, DCPNW, DCPNPW, GROUP(50), ITLP(50),
*          AGYMP(25,2), UNI(50), FTF(50), D(35),
*           PERCNT(2), UNITO(50), STRATA(2,10), TYPF(10),
*           INPUT(70), LAGS(25,2), EVALUATE(2),
*         EQUIVALENCE (PMW*DMW*D(8)), (YAS*D(2)), (AND*D(4)),
*          (RR0+0(5)), (RE+D(6)), (YOU+D(10)), (ROPAY+D(27)),
*        EQUIVALENCE (FRCNPW*WDRY), (AVGWL*WIFE), (WELPRS*PW0),
*       EQUIVALENCE (D(28)*UMAINT), (D(29)*UMAINT), (WELSPC*WSP),
*          EQUIVALENCE (STFRAC*STMFR),
*          INTEGER GROUP, GASEJ, STSEP,
*          REAL NC, COMMON /FLAGS/ FLAG, NEXT, FLAG1,
*          COMMON /BUTANE/ FLAG, FLAG1,
*          COMMON /RINART/ SUPER, NPLUID, WI, WT2, WT3, WT4,
*         ESTW1, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*          +P3, S2, S3, S4, S5, S6, S7, S8, S9, S10,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
*          +T2, T2EST, T2SATL, T3, T4, T5, T6, P1, P2,
*         ESTW, WP1, WP2, WM1, WM2, WM3, WM4, WM5, WM6,
SUBROUTINE BUTANE

P1 = PONEP
IF(PONEP GT 0.0) GO TO 200

***ASSUME A STARTING VALUE FOR INLET PRESSURE P1 TO BEGIN ITERATION
P1 = 300.
IF(T1 LT 250.0) P1 = 100.

200 CONTINUE
IF(.NOT. SUPER) GO TO 203
IF(PONEP LT 0.0) GO TO 201
P1 = 10.0
IF(1.5*PCRTLT 1000.0) P1 = 1.5*PCRT

PINCHP = MINSUP

201 CONTINUE

NIT1 = NIT2 + 1
IF(FLAG .EQ. 2) PRINT 83, NIT1, P1, PINCHP

83 FORMAT("ITERATION NO.","I5"," FOR P1/"0.","21X,"NEW VALUE OF P1=",
1 IP14.6/31X,"PINCHP=",1P14.6)

NIT1 = 0
D1 = 1.0/(RUT(T1+459.67)/P1)

220 CONTINUE

NIT1 = NIT1 * 1
PESTV = EOST(T1+459.67, D1, 0.0)
IF(ABS(PESTV - P1) LT 0.5) GO TO 240
IF(NIT1 * D1, 4) STOP "D1 FAILED TO CONVERGE"
D1 = D1*(1.0*(P1-PESTV)/P1)
GO TO 220

240 CONTINUE
IF(FLAG .EQ. 2) PRINT 80, NIT1, D1
AN FORMAT("NUMBER OF ITERATIONS TO DETERMINE D1=""I5/35X,"D1="
1 IP14.6/""ITERATION TO DETERMINE TEMPERATURE AND DENSITY AT STAGE
2 2FEST")

M1 = EOST(T1+459.67, D1)/PHI
S1 = EOST(T1+459.67, D1) / PHI * SIDEAL(T1+459.67)

C***SOLVE FOR T2EST AND D2EST(KNOWING P2 AND S2EST)USING THE MATHLIB
S2EST = 1.0
N = 2
FA(1) = 1000.
TOL = 0.01
PRNT = 0
IP(1) = 100
FX(1) = T1 + 459.67 - 100.
FX(2) = 0.4/RUTMW
TERM1 = 0
TERM2 = T2EST
CALL RONWT(FX, N, REF2D, PRNT, TOL, IP, RMS, FA)
T2EST = FX(1) - 459.67
D2EST = FX(2)
IF(FLAG .EQ. 2) PRINT 81, T2EST, D2EST

A1 FORMAT("0", "25X", "T2EST", D2EST, "1P14.6")
H2EST = EOST(T2EST+459.67, D2EST)/PHI*SIDEAL(T2EST+459.67)
CALL WFRP(0150, H2, SUPER, GRK, T1, H1, H2EST, H2SATL, TCRT)
IF(SUPER) GO TO 290

C

C***IF P1 HAS NOT CHANGED, NO NEED TO RECALCULATE T4
IF(POLY .NE. P1) T4 = FOUR(P1, DLIQ, DVAP, NFUID)

280 CONTINUE
T4 = T4
P4 = P1
SUBROUTINE BUTANE

P3=P1
H4=HEQ(T4*459.67,OLIQ)/H4HIDEAL(T4*459.67)
H5=HEQ(T5*459.67, DVAP)/H5HIDEAL(T5*459.67)

CALL PIMP(T3, H3, T2SATL, H2SATL, P2, P3, NFLUID, VSATL)

IF(SUPER) GO TO 400

WT3=WT4*PINCHP

E=WT4*(H1-H3)/(H1-H4)+WT1*(H3-H4)/(H1-H4)

T=AVGTP(WT1=ESTWT4)

C=FLUID(T1, T)

C=GTOS*(H1-H3)/CWT/(WT1=ESTWT4)

IF(FLAG.EQ.0) PRINT 1, "T0, QISO, QWAT, T1, T2EST, T3, T4, T5,
    P1, P2, P3, P4, P5, D2EST"

1 FORMAT("CYCLE CONDITIONS FOR ITERATION No.", "QISO-WORKING FLUID"/
    "FLOW RATE (L/R), "T4*1PE14.6/ "QWAT-GEOTHERMAL FLUID FLOW RATE"/
    "FLUIDS (L/R), "T4*1PE13.6/ "EXCLUDING SALT")/

2 T0"="T1-TURBINE INLET", T4*1PE14.6/

3 T0"="T2EST-TURBINE OUTLET (IDEAL)", T4*1PE14.6/

4 T0"="T3-PREHEATER INLET", T4*1PE14.6/

5 T0"="T4-VAPORIZER INLET", T4*1PE14.6/

6 T0"="T5-SUPER HEATER INLET", T4*1PE14.6/"OPRESURES"/

7 T0"="P1-TURBINE INLET", T4*1PE14.6/

8 T0"="P2-TURBINE EXHAUST", T4*1PE14.6/

9 T0"="P4-VAPORIZER INLET", T4*1PE14.6/

A T0"="P5-SUPER HEATER INLET", T4*1PE14.6/"OFNSITY"/

B T0"="P7-TURBINE INLET", T4*1PE14.6/

C T0"="P8-TURBINE EXIT (IDEAL)", T4*1PE14.6/"OENTHALPY"

IF(FLAG.EQ.2) PRINT 2, "T1, T2, T2EST, T3, T4, T5,
    P1, P2, P3, P4, P5, D2EST"

1 WT1=WT3+WT4*ESTWT4

2 FORMAT(
1 T0"="T1-TURBINE INLET", T4*1PE14.6/

2 T0"="T2-TURBINE EXHAUST", T4*1PE14.6/

3 T0"="T3-PREHEATER EXIT (IDEAL)", T4*1PE14.6/

4 T0"="T4-VAPORIZER EXIT", T4*1PE14.6/

5 T0"="T5-SUPER HEATER EXIT (PRESET)", T4*1PE14.6/

6 T0"="T1-TURBINE EXIT (ACTUAL)", T4*1PE14.6/

C T0"="W1-TURBINE EXIT (IDEAL)", T4*1PE14.6/

D T0"="T1-PREHEATER EXIT (ACTUAL)", T4*1PE14.6/

C IF(PINCHP.EQ.0) GO TO 400

C*** HAS P1 CONVERGED

CALL PNEADELOLD, DELNEW, POLD, P1, PCRT, WT4, ESTWT4, PINCHP,
    IFLAG, IPCNT)

GO TO(300*, 200*, 400*) IFLAG

300 CONTINUE

C*** ASUME PINCHP CAN NOT REACH MAXIMUM WITHOUT P1 ALSO REACHING MAX

PINCHP=(WT4*(H1-H4)-WT1*(H3-H4)-T4*(H1-H3))/(H1-H3)

IF(PINCHP.EQ.30) GO TO 200

C*** SUPERCRITICAL CYCLE

IF(FLAG.EQ.2) PRINT 3, PINCHP, P1
SUBROUTINE BUTANE

3 FORMAT(""N\$SUPERCRITICAL/"/"M\$INCHP\$P1="/"1\$2F14.6")
SUPER=TRUE.
GO TO 500
400 CONTINUE
C***KNOWING O2 AND H2 DETERMINE T2 AND V2
TERM1=O2
TERM2=H2
N=2
FA(1)=100*
TOL=0.01
PRNT=0
IP(1)=100
FX(1)=T2EST=459.67
FX(2)=0.4/FX(1)
CALL RONWT(FX, N, RFT2V2, PRNT, TOL, IP, RMS, FA)
T2=FX(1)-459.67
V2=1.0/FX(2)
V2EST=1.0/D2EST/BUTMW
RETURN
END
SUBROUTINE CNDNSR

SUBROUTINE CNDNSR(OCWAT, CWGPM, CWTI, CWT2, H2SATL, T2, T2SATL, P2,
   NFLUID, PVELC, DIAC, DIANC, FOULFI, FOULFO, GISO, HTACON, H2SATV,
   CWT1, CWPI, CWT2, CWPI2)

ROUTINE DETERMINES BOTH THE COOLING WATER FLOW RATE FOR THE WORKING
FLUID CONDENSER AND THE CONDENSER CHARACTERISTICS. THE WORKING FLUID
IS CONTAINED IN THE CONDENSER TUBES.

INPUT PARAMETERS

CWTI = COOLING WATER INLET TEMPERATURE
CWT2 = WORKING WATER OUTLET TEMPERATURE
H2 = WORKING FLUID ENTHALPY AT TURBINE EXHAUST
H2SATL = WORKING FLUID ENTHALPY AT CONDENSER EXIT
T2 = WORKING FLUID TEMPERATURE AT TURBINE EXHAUST
T2SATL = WORKING FLUID TEMPERATURE AT CONDENSER OUTLET
P2 = WORKING FLUID PRESSURE AT TURBINE EXHAUST
NFLUID = INDEX OF WORKING FLUID
CWT = COOLING WATER VELOCITY
DIAC = INSIDE TUBE DIAMETER
DIANC = OUTSIDE TUBE DIAMETER
FOULFI = INSIDE FOULING FACTOR
FOULFO = OUTSIDE FOULING FACTOR
GISO = WORKING FLUID FLOW RATE
RVELC = VELOCITY OF COOLING WATER IN CONDENSER

OUTPUT PARAMETERS

OCWAT = COOLING WATER FLOW RATE
CWGPM = COOLING WATER FLOW RATE GALLONS PER MINUTE

REAL KALUM, KAY, MRP
COMMON /FLAGS/ FLAG, NEXT, FLAG1
INTEGER FLAG, FLAG1
COMMON /XOUT/ XRSLT(9, 3)
COMMON /BUTANE/ AB, AZERO, AB1, AB2, AB3, AB4, BB, BZERO, HB1,
   * AB2, AB3, AB4, CZERO, DZCNO, EZERO, SMA, SMAB, SMHC, SHD, ALPHAR,
   * GAMMA, PHI, BUTMW, AR
COMMON /BECOM/ AL, ALCL, CLUL, EL, A8FR, C, UFR, E, FR, SB, A2, B2, C2, A3, B3
   * E3, A4, B4, C4, A5, B5, C5, A6, B6, C6, KAY, ALPHA, HCA, HCB, HCC, HCD, HCF, XFR
   * YPR, PP, TL, VC

DATA KALUM/30/>

**CONSTANTS**

******DESUPERHEATER**********

***ION TO YIELD THE COOLING WATER TEMP ASSOCIATED WITH THE
SATURATED VAPOR (CWT12) AND THE AVG SPEC HEAT OF THE COOLING
WATER WHILE THE WORKING FLUID IS SUPERHEATED (CWCP)

CWT = AVGTM(CWT1, CWT2)
CPFEST = FLUID(1, 1, CWT1, "DESUPERHEATER AT CPFEST")
CP1 = FLUID(1, 1, CWT1, "DESUPERHEATER AT CWT1")
CP2 = FLUID(1, 1, CWT2, "DESUPERHEATER AT CWT2")
HFG = FLUID(9,NFLUID,T2SATL,"DESUPERHEATER AT HFG")
SUBROUTINE CNDNSR

H2SATV = H2SATL * MFG
GQWAT = GISO + (H2-H2SATL) / (CP2 - CWI8 - CR1 - CWT1)
TOLD = -(H2-H2SATV) * GISO / GQWAT / CPEST / CWT2

1 CPEST = FLUID(1,1,TOLD,"DESUPERHEATER AT CPEST")
TNW = CWT2-(H2-H2SATV) * GISO / GQWAT / CPEST
IF (ABS(TOLD-TNEW)*LT.0.5) GO TO 100
TOLD = TNEW
GO TO 1
100 CONTINUE
CWT12 = TNEW
CWCP = CPEST

C DETERMINER PROPERTIES OF COOLING WATER AT CWT12

C CWT12 = AVGTMP(CWT12+CWT2)
CWCON = FLUID(2:1+CWT12:"DESUPERHEATER AT CWCON")
CWRO = FLUID(3:1+CWT12:"DESUPERHEATER AT CWRO")
CWVIS = FLUID(4:1+CWT12:"DESUPERHEATER AT CWVIS")
CWPR = MRODTL(CWT12/10.)
CWREYN = REYN(0.0+CWVEL+DIAC+CWVIS)

C CONVERT TO GALLONS PER MINUTE
CWGPM = GQWAT / CWRO 0.1246753

C NUMBER OF TUBES IN DESUPERHEATER

C DSTUBE = TUBES(GQWAT,CWPHO,CWVEL,DIAC)

C INSIDE HEAT TRANSFER COEFFICIENT

C MTCD = MTCOEF(CWCON+DIAC+CWPR+CWREYN+4)
T2VAVG = AVGTMP(T2VAVG+CWT12)

C WORKING FLUID PROPERTIES (VAPORS) AT T2VAVG

PCPV = FLUID(5,NFLUID,T2VAVG:"DESUPERHEATER AT PCPV")
PCONDV = FLUID(6,NFLUID,T2VAVG:"DESUPERHEATER AT PCONDV")
IF (NFLUID.GT.3) GO TO 500
NITA = 0
DB = 1.0 / (RB = (T2VAVG * 459.67) / P2)

480 CONTINUE
NITA = NITA + 1
PESTV = PEST(T2VAVG * 459.67+DB, 0,0)
IF (ABS(PESTV-P2) *LT.0.5) GO TO 490
IF (NITA.GT.4) STOP "SUDHEX-DB FAILED TO CONVERGE"
DB = DB 1.0 + (P2-PESTV)/P2
GO TO 480

490 PRHNO = DB*RUTHW
GO TO 520

500 CONTINUE
VB = R*(T2VAVG * 459.67) / P2
MAXR = 0

510 CONTINUE
MAXR = MAXR + 1
PESTV = EQSTAP(T2VAVG * 459.67+VB+0)
IF (ABS(PESTV-P2) *LT.0.5) GO TO 515
IF (MAXR.GT.4) STOP "SUDHEX-VR FAILED TO CONVERGE"
VB = VB 1.0 + (PESTV-P2)/P2
SUBROUTINE CNDNSR

GO TO 515
515 CONTINUE
PRHOV = 1.0/VB
520 CONTINUE
CALL AREA(DIAOC, DUSTUBE, EODIA, FLAREA)
PVISV = FLUID(q, NFLUID, T2VAVG, "DESUPERHEATER AT PVISV")
PVEL = VELOC(q, QISO, FLAREA, PRHOV)
REYNOV = REYNO(PRHOV, PVEL, EODIA, PVISV)
PVEL = PVEL / 3600.*
PRANOVR = PVISV / PCONDV

C OUTSIDE HEAT TRANSFER COEFFICIENT

HTCO = HTCOEF(FCONDV, EODIA, PRANOVR, REYNOV, 0.3)

C OVERALL HEAT TRANSFER COEFFICIENT

U = 0HTCOEF(DIAOC, HTCI, DIAC, KALUM, HTCO, FOULFI, FOULFO)

C HEAT TRANSFER AREA

T2SATV = T2SATL
WT = T2SATV - CWT12
FT = T2 - CWT2
IF (WT. LE. FT) GO TO 5
HTA = UTAMEA(QISO, H2, H2SATV, U, WT, FT)
GO TO 7
5 WT = -WT
FT = -FT
HTA = UTAMEA(QISO, H2, H2SATV, U, FT, WT)
HTA = -HTA
7 CONTINUE

C HEAT TRANSFER LENGTH

HTL = HYNLNG(HTA, DIAC, DSTUBE)

HXRLT(1+1) = DIAC
HXRLT(2+1) = DIAOC
HXRLT(3+1) = DUSTUBE
HXRLT(4+1) = EODIA
HXRLT(5+1) = HTCI
HXRLT(6+1) = HTCO
HXRLT(7+1) = U
HXRLT(8+1) = HTA
HXRLT(9+1) = HTL
HTA1 = HTA
HTL1 = HTL
FLAREA = FLAREA
EQDIA1 = EODIA

C
**SUBROUTINE CNDNSR**

---

**PROPERTIES OF COOLING WATER AT CWT112**

- `CWT112 = AVGTMP(CWT1+CWT14)`
- `CWFCON = FLUID(2+1+CWT112utzer) CONDENSER AT CWFCON`
- `CWRHO = FLUID(3+1+CWT112utzer) CONDENSER AT CWRHO`
- `CWVIS = FLUID(4+1+CWT112utzer) CONDENSER AT CWVIS`
- `CWPRAN = PRNDL(CWT112/10V.)`  
- `CWEYNN = KEYNQ(CWRHO,CWVEL*DIAC,CWVIS)`

**INSIDE HEAT TRANSFER COEFFICIENT**

- `HTCI = HTCOEF(CWCON+DIAC+CWPRAN+CWEYNN*4)`

**NUMBER OF TUBES IN CONDENSER**

- `CTUBE = TURES(CWAT+CWRH+CWVEL*DIAC)`

**PROPERTIES OF THE TWO PHASE WORKING FLUID AT T2SATL**

- `PEPL = FLUID(1+NFLUID+2SATLutzer) CONDENSER AT PEPL`  
- `PCOND = FLUID(2+NFLUID+2SATLutzer) CONDENSER AT PCOND`  
- `PRHOV = FLUID(7+NFLUID+T2SATLutzer) CONDENSER AT PRHOV`  
- `PVISV = FLUID(9+NFLUID+T2SATLutzer) CONDENSER AT PVISV`  
- `PCPRIT = FLUID(12+NFLUID+T2SATLutzer) CONDENSER AT PCPRIT`  
- `SIGMA = FLUID(12+NFLUID+T2SATLutzer) CONDENSER AT SIGMA`  
- `DELTAP = PCPRIT - P2`  

**TOTAL FLOW AREA**

- `CALL QAREA(DIAOC,CTUBE,EQUID,FLAREA)`  
- `PVEL= VELOCY(QIS0,FLAREA,PRHO)`  
- `REYNOL = REYNQ(PRHO,PVEL,EQUID,PRISL)`  
- `PRANOL = PEPL*PVISL/PCPRIT`  

**BOILING HEAT TRANSFER**

- Assume macroconvective heat transfer only

**MARTINELI PARAMETER**

- `MARTPA = (PRHOV/PRHO)**.5 * (PVISV/PVISL)**.5`

**MULTIPLICATION FACTOR**
SUBROUTINE CNDNSR

C
F = 1.76098 * 1.179312 * MARTMA * 0.007466 * MARTPA ** 0 * 2.6585565 - 5
C
C
MACRO
CONVECTIVE HEAT TRANSFER COEFFICIENT

C
FOTHICO = F * 0.0234 * REYNOL * 0.05 * PRANOL * 0.4 * PCOND / EODIA
CONTINUE

C
OVERALL HEAT TRANSFER COEFFICIENT
C
U = ONRTOL(DIANC, HTC, DIAL, KALUM, FTHTCO, FOULFI, FOULFO)
WT = T2SATV - CWT12
FT = T2SATL - CWT1

C
AVERAGE HEAT TRANSFER AREA

C
HTA = UTAMFA(QISO, H2SATV + H2SATL, U, WT, FT)

C

C
HEAT TRANSFER LENGTH

C
HTL = UTLNGH(HTA, DIANC, CLURE)

450 CONTINUE
MXRSLT(1 + 2) = DIA
MXRSLT(2 + 2) = DIAOC
MXRSLT(3 + 2) = CTUBE
MXRSLT(4 + 2) = EODIA
MXRSLT(5 + 2) = HTC
MXRSLT(6 + 2) = FTHTCO
MXRSLT(7 + 2) = U
MXRSLT(9 + 2) = HTA
MXRSLT(18 + 2) = HTL
TOTHTA = HTA + HTA1
TOTHTL = HTL + HTL1
AVGFLA = (FLAREA + FLAP1) / 2
AVGFOO = (EODIA + EODIA1) / 2
HTACON = TOTHTA
CWP12 = CWP1 - (CWP1 - CWP2) * (HTA1 / TOTHTA)
RETURN
END
FUNCTION DNLIQA

FUNCTION DNLIQA(T)
COM**USED BY FREON-11, FReON-22 (NFLUID=4,7)
REAL KAY
COM**COMMON FOR THE FREON CONSTANTS
**C3*A4*4*C4*A5*C5*A6*C6*KAY*ALPHA*HCA*MCB*HCC*HCD*HCF*XFR
**YFR*PR*TU*VC
AA=1.0/T/TC
DNLIQA=AL*BL*A**((1./3.)*CL*A**(2./3.)*DL*A*EL*A**((4./3.))
RETURN
END
FUNCTION DNLIQB

FUNCTION DNLIQB(T)
C***USED BY FREON-12, FREON-114 (NFLUID=5, 9)
REAL KAY
C***COMMON FOR THE FREON CONSTANTS
COMMON /FRFCON/ AL,BL,CL,VL,EL,AL,BFR,CFR,F,E,F,R,SH,A2,A2,B2,C2,A3,B3,
* C3,A4,B4,C4,A5,B5,C5,A6,B6,C6,KAY,ALPHA,HCA,HCB,HCC,HCD,MCF,XFR
* YFR,PC,TL,VC
TCMT=TC - T
DNLIQB=AL + BL*TCMT + CL*SQRT(TCMT) + DL*TCMT**1.5 + EL*TCMT**2
RETURN
END
FUNCTION DNLIQC

FUNCTION UNLIGC(T)

*** USED BY FREON-21, FREON-113 (FLUID=6,8)

*** COMMON FOR THE FREON CONSTANTS

COMMON /FRECON/ AL,RL,CL,PL,EL,A,BFR,C+VRP+E,FR+SR,A2,B2,C2,A3,B3

* E,F,G,H,A4,A5,A6,B6,C6,K,CY,ALPHA,HC,HCB,HC,HCC,HC0,HC,F,FR

* YFR,P,R,TL,VC

DNLIQC=AL + BL*T + CL*T**2

RETURN

END
SUBROUTINE EFT2V2(X, N, K, R, P)

C***DETERMINES T2EST AND V2EST
COMMON /FLAGS/ FLAG, NEXT, FLAG1
INTEGER FLAG, FLAG1
COMMON /RUNCOM/ TERM1, TERM2, SAVTV(2), JFLAG, DEL(2), OLDR1
DIMENSION X(2), R(d)
IF(ABS(X(1)) GT .0000) GO TO 240
IF(X(2) LT .007) GO TO 220
R(1) = ENSTAP(X(1), X(2), TERM1)
R(2) = ENSTAP(X(1), X(2), TERM2)
SAVT(1) = A(1)
SAVT(2) = A(2)
200 CONTINUE
IF(FLAq .LT. 2) PRINT 1, X
1 FORMAT('0', 20X, 'EFT2V2 REBONAL51', 1122E14.6, 34X, 'I, V='1P2E14.6)
RETURN
220 CONTINUE
C**MINUS VOLUME OR SLOW CONVERGANCE
JFLAG=1
DEL(1) = (SAVT(1) - X(1))/2*U
DEL(2) = SAVTV(2)/2*U
GO TO 700
240 CONTINUE
X(2) = SAVTV(2)
X(1) = SAVTV(1)
250 CONTINUE
GRE = ENSTAP(X(1), X(2), TERM1)
IF(ABS(GR) LE 0.5) GO TO 300
X(1) = X(1) + (1 - GR/TERM1)
GO TO 297
300 CONTINUE
R(1) = 0.0
R(2) = 0.0
IF(FLAG LE 2) PRINT 1, X
RETURN
END
FUNCTION EQSTAP(T*V*P)

COMMON /BINARY/ SUPER, NFLUID, WT1, WT2, WT3, WT4,
* ESTWT4, WP1, WP3, CWT1, CWT2, CWPT1, WP2, T1,
* T2, T2EST, T2SATL, T3, T4, T5, P1, P2,
* P3, P4, P5, M1, M2, M2EST, H3, H5,
* H4, H5, S1, SEST, S3, S4, SVSATL, D1,
* D2, D2EST, V1, VEEST, PICHP, WVELL, DIAL, DIAPL,
* WVELV, DIAV, DIAOV, WVELS, DIAS, DIAOS, WVELSC, DIASC,
* DIASC, DVELC, DIAE, FOULF, FOULFQ, BAR, GISO,
* HTAEEx, HTACON, TAPRCH, WTFOUR, APPCON, APPCT, APPFX,
* DELPHX, PINSUB, PINSUP, PUEP, RANGE

LOGICAL SUPER
REAL KAY

C***COMMON FOR THE FREON CONSTANTS
INTEGER FLAG
COMMON /FLAGS/ FLAG, NEXT, FLAG1
COMMON /FRECON/ A10L, CL0, E1, A.BFR, C.UBF, E1, RSR, A2, R2, C2, A3, B3
,** C3, A4, B4, C4, A5, B5, C5, A6, B6, C6, KAY, ALPWA, HCA, HCB, HCC, HCD, HCF, XFR
,** YFR, PT, TC, VC
VMS8=W-8R
PART1=A*T/VMS8
EXPWAE=EXP(-KAY*T/TC)
PART2=(A2 + B2*T + C2*EXPWAE)/VMS8**2
PART3=(A3 + B3*T + C3*EXPWAE)/VMS8**3
IF (NFLU1D.EQ.1) GO TO 10U
PART4=(A4 + B4*T) / VMS8**4
GO TO 150
100 PART5=(A4*R4*T+C4*EXPWAE)/VMS8**4
150 CONTINUE
PART6=(A5 + B5*T + C5*EXPWAE)/VMS8**5
PART6=0.0
EXPWA=0.0
AV=ALPHA = V
IF(AV .GT. 120.) GO TO 200
C***FOR A12, A21, A113 AND A116, A6, H6 AND ALPHA=0.0
IF(A6 .EQ. 0.0) GO TO 200
EXPWA=EXP(AV)
PART6=(A6 + B6*T) / EXPWA
200 CONTINUE
EOESTAP=PART1 + PART2 + PART3 + PART4 + PART5 + PART6 - P
RETURN
END
SUBROUTINE FFT2V2

SUBROUTINE FFT2V2(X, N, K, P)
COMMON /FLAGS/ FLAG, NEXT, FLAG1
INTEGER /FLAGS/ FLAG1
COMMON /RUNCOM/ TERM1, TERM2, SAVTV(2), JFLAG, DEL(2), OLDR1
DIMENSION X(2), R(2), RESITV(2)
IF (ABS(X(1)) GT 20.0) UC TO 240
IF (X(2) .LT. 0.0) GO TO 240
C***E Termites T2 and V2
R(1)=ESTAP(X(1), X(2), TERM1)
R(2)=MVAP(X(1), X(2), TERM1) - TERM2
SAVTV(1)=A(1)
SAVTV(2)=A(2)
ABS1=ABS(R(1))
IF (ABS1 .GT. 0.0) GO TO 200
OLDR1=ABS1
BESTTV(1)=X(1)
BESTTV(2)=X(2)
200 CONTINUE
RETURN
220 CONTINUE
JFLAG=1
DEL(1)=(SAVTV(1) - X(1)) / 2.0
DEL(2)=(SAVTV(2) - X(2)) / 2.0
GO TO 340
240 CONTINUE
X(1)=BESTTV(1)
X(2)=BESTTV(2)
250 CONTINUE
GR=EOSTAP(X(1), X(2), TERM1)
IF (ABS(GR) .LE. 0.5) GO TO 300
X(1)=X(1) + (X(1) - GR/TERM1)
GO TO 250
300 CONTINUE
R(1)=0.0
R(2)=0.0
RETURN
END
FUNCTION FLUID(NPROP, NFLUID, TEMP, LOCAT)
COMMON /FLAGS/ FLAG, NEXT, FLAG
INTERESN FLAG, FLAG1

LOGICAL WARN

DIMENSION LOCAT(2)
DIMENSION COLLAJ(10,14,8)
DIMENSION FLUP(10), PROP(13,3)

COLL AJ(NFLUID, NPROP, ICLA)

NFLUID = 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 = AMMONIA

NPROP = PROPERTY INDEX
1 = SPECIFIC HEAT (LIQUID) (BTU/LB DEGREES F)
2 = THERMAL CONDUCTIVITY (LIQUID) (BTU/HR FT DEGREES F)
3 = DENSITY (LIQUID) (LB/CF FT)
4 = VISCOSITY (LIQUID) (LH MASS/FT HR)
5 = SPECIFIC HEAT (GAS) (BTU/LB DEGREES F)
6 = THERMAL CONDUCTIVITY (BTU/HR FT DEGREES F) (GAS)
7 = DENSITY (GAS) (LB MASS/CF FT)
8 = VISCOSITY (GAS) (LB MASS/FT HR)
9 = HEAT OF VAPORIZATION (BTU/LB)
10 = CRITICAL PRESSURE (PSIA)
11 = CRITICAL TEMPERATURE (DEGREES F)
12 = SURFACE TENSION (LB FORCE/FT)
13 = MOLECULAR WEIGHT (LB/LB MOLE)

COLL AJ(NFLUID, ICLA) CONTAINS ADDITIONAL COEFFICIENTS. THESE
ARE NECESSARY BECAUSE OF TWO SETS OF COEFFICIENTS TO DETERMINE
THE DENSITY OF GAS FOR ALL OF THE FLUIDS EXCEPT WATER AND R-21.

ICLA = COEFFICIENTS, LIMITS AND ADJUSTMENT FACTORS
1 = FIRST COEFFICIENT
2 = SECOND COEFFICIENT
3 = THIRD COEFFICIENT
4 = FOURTH COEFFICIENT
5 = LOWER TEMPERATURE LIMIT
6 = UPPER TEMPERATURE LIMIT
7 = FIRST ADJUSTMENT FACTOR, ADDED TO TEMPERATURE
8 = SECOND ADJUSTMENT FACTOR, DIVIDED INTO TEMPERATURE

VALUES FOR WATER
DATA ((COLL AJ(I,1), I=1,10), J=1,14, I=1,8) /
FUNCTION FLUID

C LIQUID SPECIFIC HEAT (1.1 J)
  * 0.991428
  * 0.921649
  * -0.015377
  * 0.003724
  * 32.0

C LIQUID THERMAL CONDUCTIVITY (1.2 J)
  * 0.312721
  * 0.253946
  * -1.055776
  * 9.0
  * 32.0

C LIQUID DENSITY (1.3 J)
  * 62.87418
  * 0.0
  * -3.01818
  * 1.0
  * 50.0

C LIQUID VISCOSITY (1.4 J)
  * 2.1997
  * 0.26103
  * 0.0
  * 1.0
  * 100.0

C THERMAL CONDUCTIVITY OF GAS (1.5 J)
  * 0.011731
  * 0.13058
  * -0.12242
  * 0.013467
  * 200.0

C SPECIFIC HEAT OF GAS (1.6 J)
  * 0.0200242
  * 4.01325
  * 1.0
  * 200.0

C THERMAL CONDUCTIVITY OF GAS (1.7 J)
  * 0.0
  * 0.0
  * 0.0
  * 1.0

C HEAT OF VAPORIZATION (1.9 J)
  * 1.089
  * 13
  * -0.429748
  * -3.40511
  * 0.0
  * 100.0

C CRITICAL PRESSURE (1.10 J)
  * 320.8
  * 320.8
  * 0.0
  * 1.0

C CRITICAL TEMPERATURE (1.11 J)
  * 705.47
  * 705.47
  * 0.0
  * 1.0

C SURFACE TENSION (1.12 J)
  * 4.03577
  * 4.03577
  * 0.0
  * 1.0

C MOLECULAR WEIGHT (1.13 J)
  * 12.0
  * 12.0
  * 0.0
  * 0.0

C DATA ((COLLAJ(1,1,J),J=1,6), I=9,13) /

C SPECIFIC HEAT OF LIQUID (2.1 J)
  * 0.43838
  * 0.256242
  * -0.1343026
  * 0.0
  * 75.0

C THERMAL CONDUCTIVITY OF LIQUID (2.2 J)
  * 6.44842
  * 2.461118
  * -2.10477
  * -2.51871
  * 75.0

C DENSITY OF LIQUID (2.3 J)
  * 37.8239
  * -6.21837
  * 2.90839
  * -1.15089
  * 0.0

C VISCOSITY OF LIQUID (2.4 J)
  * 0.1
  * 0.0
  * 0.0
  * -50.0

C DATA ((COLLAJ(2,1,J),J=1,6), I=1,8) /

C SPECIFIC HEAT OF LIQUID (2.1 J)
  * 0.44838
  * 0.256242
  * -0.1343026
  * 0.0
  * 75.0

C THERMAL CONDUCTIVITY OF LIQUID (2.2 J)
  * 6.44842
  * 2.461118
  * -2.10477
  * -2.51871
  * 75.0

C DENSITY OF LIQUID (2.3 J)
  * 37.8239
  * -6.21837
  * 2.90839
  * -1.15089
  * 0.0

C VISCOSITY OF LIQUID (2.4 J)
  * 0.1
  * 0.0
  * 0.0
  * -50.0
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**Data for M=11**

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</tr>
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<td>Viscosity of gas</td>
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</table>
HEAT OF VAPORIZATION (4.9, J)
- 19.75977  - 19.405426  - 12.253346  - 6.112320  - 0.0

C

CRITICAL PRESSURE (4.10, J)
- 8.769  0.0  1.0  0.0  388.4

C

CRITICAL TEMPERATURE (4.11, J)
- 388.4  0.0  0.0  0.0

C

SURFACE TENSION (4.12, J)
- 1.253346  0.0  0.0  100.0

C

MOLECULAR WEIGHT (4.13, J)
0.0  1.0

C

DENSITY OF GAS T > GT (4.14, J)
- 52.148845  0.065767  - 25.858909  3.845847  18.0

C

VALUES FOR M=12

DATA (COLLAJ(5,1, J) = 1, I=1...8)

C

SPECIFIC HEAT OF LIQUID (5.1, J)
- 0.0993113  2.42424E-3  - 9.72960E-6  1.49532E-8  - 140.0

C

THERMAL CONDUCTIVITY OF LIQUID (5.2, J)
- 0.1199956  - 2.44840E-4  0.0  0.0  - 190.0

C

DENSITY OF LIQUID (5.3, J)
- 9.360742  - 16.595114  8.18744  - 4.011931  0.0

C

VISCOITY OF LIQUID (5.4, J)
- 0.2299986  - 2.41973E-3  0.0  0.0  - 5.0

C

SPECIFIC HEAT OF GAS (5.5, J)
- 0.087828  5.59820E-4  - 7.04869E-7  3.29112E-10  - 280.0

C

THERMAL CONDUCTIVITY OF GAS (5.6, J)
- 4.32443E-3  - 1.54487E-5  4.49688E-9  0.0  - 10.0

C

DENSITY OF GAS (5.7, J) T > LE, 120

C

VISCOITY OF GAS (5.8, J)
- 1.42332E-4  - 1.0202E-4  - 6.52732E-9  0.0  - 20.0

C

DATA (COLLAJ(5,1, J) = 1, I=9...16)

C

HEAT OF VAPORIZATION (5.9, J)
- 69.75977  - 19.405426  - 12.253346  - 6.112320  - 0.0

C

CRITICAL PRESSURE (5.10, J)
- 582.0  0.0  0.0  232.7

C

CRITICAL TEMPERATURE (5.11, J)
- 582.0  0.0  0.0  232.7
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Values for H=21:

Data (COLLAJ(A+I,J)+J=1+8), I=I+8) /
FUNCTION FLUID

VALUES FOR R=72

DATA ((COLL\(7,1\), J=1), I=1, 8) /

SPECIFIC HEAT OF LIQUID (7\(1\), J) =
- 0.26765
- 3.22637E+5
- 1.93112E-6
- 2.40.0
- 3.86552E+9
- 2.40.0

DATA ((COLL\(7,1\), J=1), I=1, 8) /

DATA ((COLL\(7,1\), J=1), I=1, 8) /

DATA ((COLL\(7,1\), J=1), I=1, 8) /

DATA ((COLL\(7,1\), J=1), I=1, 8) /

VALUES FOR R=113

DATA ((COLL\(7,1\), J=1), I=1, 8) /
FUNCTION FLUID

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Note: The table format is slightly altered for better readability, but the information remains the same.
FUNCTION FLUID

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&290.0 & 0.0 & 100.0 & \\

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& R6.0 & 0.0 & 1.0 & \\

C & -0.027243A & 1.14849E-3 & -1.8265E-6 & 1.5928E-9 & 6.0 & \\
& 460.0 & 460.0 & 1.8 & \\

C & 9.18242E-3 & 4.39676E-5 & 1.5285E-7 & 0.0 & 1.0 & \\
& 440.0 & 460.0 & 1.8 & \\

C & 0.266450Y & 0.5628975 & 0.292512 & 0.750146 & 0.0 & \\
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& 500.0 & 460.0 & 1.8 & \\

C & \text{DATA (COLLAJ(9J+J)), J=1 to 9, I=9+14) /} \\

C & 62.3000A & -14.82155 & 5.44204 & -2.24406 & 0.0 & \\
& 290.0 & 0.0 & 100.0 & \\

C & 474.0 & 0.0 & 0.0 & 1.0 & \\
& 294.4 & 0.0 & 0.0 & 1.0 & \\

C & 294.4 & 0.0 & 0.0 & 1.0 & \\
& 0.0 & 0.0 & 1.0 & \\

C & 8.2222E-4 & 0.0 & 0.0 & 77.0 & \\
& 77.0 & 0.0 & 1.0 & \\

C & 170.9 & 0.0 & 0.0 & 0.0 & \\
& 0.0 & 0.0 & 1.0 & \\

C & 95.46959 & 14.950850 & 76.087139 & 45.025255 & 150.0 & \\
& 290.0 & 0.0 & 100.0 & \\

C & \text{VALUES FOR AMMONIA /} \\

C & \text{DATA (COLLAJ(10J+J), J=1 to 10, I=1+8) /} \\

C & 1.43544063 & 3.8939744 & 532.2260957 & 4391624728 & 35.0 & \\
& 465.0 & 0.0 & 100.0 & \\

C & 0.521 & -4.25E-4 & -0.0 & 80.0 & \\
& 220.0 & 469.0 & -1.0 & \\

C & 45.811364 & -2.2797384 & 0.3693810 & -4.06246371 & 40.0 & \\
& 270.0 & 0.0 & 100.0 & \\

C & 63339937 & -1.7049357 & 4.334106820 & -4.0031940153 & 32.0 & \\
& 400.0 & 0.0 & 100.0 & \\

C & 465740 & -1.09343E-1 & 4.01395E-7 & -1.4539E+10 & -58.0 &
FUNCTION FLUID

C * 675.0 469.0 1.0
C THERMAL CONDUCTIVITY OF GAS (10^6 J)
C 3.935273703E-2-6.2E7683E-3 2.73493078E-3-3.19723D3F-4 40.0
C 200.0 100.0
C DENSITY OF GAS (10^6 J)
C 4.58842441 1.46188464E1 4.460.8526E1 3.91848841 40.0
C 270.4 100.0
C VISCOSITY OF GAS (10^8 J)
C 2.0786234E2 2.3037181E-3 3.0762930E-4 40.0 2.7125531E-5 32.0
C 500.0 100.0
C DATA (COLLAP(10,J+I,9,13)/

C DATA (PROP(I,J+1,J+3); I=13)/
C "SPECIFIC M" "HEAT OF LIQUID" "DENSITY OF" "LIQUID"
C "VISCOSITY OF LIQUID" "SPECIFIC M" "HEAT OF LIQ"
C "THERMAL CONDUCTIVITY OF LIQUID" "DENSITY OF" "DENSITY OF" "VISCOSITY"
C "DENSITY OF" "VISCOSITY OF" "HEAT OF VAPORIZATION"
C "CRITICAL PRESSURE" "CRITICAL TEMPERATURE" "SURFACE TENSION"
C "MOLECULAR WEIGHT" "MOLECULAR WEIGHT"

C CHECK FOR THOSE FLUIDS THAT HAVE TWO SETS OF COEFFICIENTS FOR THE
C DENSITY OF GAS
C N= NPRP
C NF= NFLUID
C IF(NF.NF=9.08. NF.E0.6 .UR. NF.E0.10) GO TO 200
C IF(I=7) GO TO 260
C IF(TEND .RT. COLLAP(NF,NPT)NP:14
C 200 CONTINUE
FUNCTION FLUID

C CHECK RANGE: DO NOT CHECK CRITICAL PRESSURE OR CRITICAL TEMPERATURE
C
C WARN=.FALSE.
C IF(NP .EQ. 10 .OR. NP .EQ. 11) GO TO 220
C IF(TEMP .LT. COLLAJ(NF,NP,4)) WARN=.TRUE.
C IF(TEMP .GT. COLLAJ(NF,NP,6)) WARN=.TRUE.

220 CONTINUE

C ADJUST TEMPERATURE
C
C T = (TEMP+COLLAJ(NF,NP,7))/COLLAJ(NF,NP,8)
C
C FLUID = COLLAJ(NF,NP,1) + COLLAJ(NF,NP,2)*T + COLLAJ(NF,NP,3)*T^2
C + COLLAJ(NF,NP,4)*T^3
C
C IF(.NOT. WARN) RETURN

C TEMPERATURE OUT OF RANGE, PRINT WARNING
C
C IF(FLAG .NE. 2) RETURN
C WRITE(*,1) TEMP, (PROP(NPROP(i),i=1,3), FLUID, FLU(NFLUID), COLLAJ
C = (NF,NPROP(5), COLLAJ(NF,NPROP(6)),LOCAT
C 1 FORMAT(*"**WARNING**/
C "' THE INPUT TEMPERATURE ("'F8.3"," FOR DETERMINING THE "'A10,
C = "'(F8.3)"")/
C " FOR "'A90" IS OUTSIDE THE RANGE OF "'F8.3" TO "'F8.3/
C " THE CALL TO FLUID WAS IN "'A10/
C
C RETURN
C END
SUBROUTINE FREON
EXTERNAL FFT2V2, FFTV2
REAL XAY
COMMON /NAMLST/ LL1, LL2, LL3, LL4, LL5, LL6
*LL7, LL8, LL9, LL10, LL11, LL12, NUMFTF, ITEMS
*ACFRS, PNDUPT, OINJW, CINWCL, FIVTPW, PWIOA, PF, WDEPTH
*DEPTH, AMWEPW, AVGML, WPS, PWTEMP, PREST, DIA, FLORAT
*DPS, PM, CAC03, WELPRS, CASFRC, PRDRAT, NUMTYP, WELSPC
*PMWTH, STMFR, FRCPW, PCCPW, NYC, NC, IPMT, PSLAVG
*NFLASH, PCOD, GRKW, WRAT, DBAT, E, RI, TCI
*POINT, TPDAFT, TCO, TCODC, TCMHC, GASEJ, CINLAB, INJW
*INJP, RINJW, PLINJP, PLINJW, ISCRUB, MIX, PCTS, LFLASH
*STSPER, PONFACT, IJPPUMP, WMHAS, WPCO2, WPCH4, WPONCG, TOTNCG
*IOEEP, METH, DLMETH, LIPHAS, IDNPUM, BDNEP, IORILL, IDPRNT
*CHARAT, STATUS, SLNCST, URPSTG, IPUNCH, DRLPR
*IAWEL, DCPW, DCNPW, DLINJW, GROUP(50)
*LAGSTM(2542), UNIT(50), FTF(50), D(35)
*PPCNT(2), UNIT(50), STRATA(210), TYPE(10)
*FINPUT(70), LAGS(2542), EVALUATE(2)
*EQUIVALENCE (PMWE*DMW(0)), (YRS*D(2)), (BND*D(4))
*ERRS(0), D(5), (MRR*D(6)), (YOU*D(10)), (ROYPAY*D(27))
*DDEPTH, PMECLEM
*EQUIVALENCE (FRCPW, WDRY), (AVGLW*WFLF), (WELPRS, PWMD)
*EQUIVALENCE (D(128), MAINT), (D(29), MAINT), (WELSPC, WSP)
*EQUIVALENCE (STFRAC, STMFR), INTEGRATED GROUP, GASJ, STSPER
REAL NC
COMMON /FLAGS/ FLAG, NEXT, FLAG1
INTEGRATED FLAG, FLAG1
COMMON /BINARY/ SUPER, NFUID, WT1, WT2, WT3, WT4
*ESTWT4, WP1, WP2, CMT1, CMT2, CW1, CW2, TP1, TP2
*T2, T2EST, T5ATL, T3, T4, T5, TC, TP2
*PS, PA, PS, M1, M2, M2EST, M2ATL, M3
*M2, M3, M4, S1, S2EST, S3, S4, SVSATL, D1
*D2, D2EST, V1, V2EST, PINCHP, WVELL, DIAL, DIAL
*WVEL, DIAV, DIAO, WVELS, DIA5, DIAOS, WVELSC, DIAASC
*DIAASC, PVELC, DIAC, DIAOC, FOILFI, FOULFG, BAR, QISO
*HTAVE(3), HYACON, TAPRCH, WTPF, APPCON, APPCT, APPFX
*DELPHX, PINSUB, PINSUP, PUNEP, RANGE
LOGICAL SUPER
C***COMMON FOR THE FREON CONSTANTS
*C3, A4, B4, C4, A5, B5, C5, A6, B6, C6, K1, ALPHA, HCA, HCB, HC0, HCO, MCF, XFR
*YFR, PC, TL, VTC
COMMON /FRCON/ TERM, TERM2, SAVTV(2), JFLAG, DEL(2), OLDR1
DIMENSION FX(2), IR(3), FA(287)
IPCN=0
DELLOG=0.0
Z=0.181253
PINCH=PINSUB
PERR=FLUID(10), NFUID, 0.0, "SUBROUTINE, " FREON"
TERR=FLUID(11), NFUID, 0.0, "SUBROUTINE, " FREON"
C***DETERMINE P2, SVSATL AND M2ATL
PLOG=IPRTA(T5ATL*459.67)
P2=1.0**P2LOG
IF(FLAG*EQ.2) PRINT 12, P2
12 FORMAT("0P2=TURBINE EXHAUST PRESSURE (PSIA)"*,T65,1PE14.6)
SUBROUTINE FREON

IF(NFLUID.EQ.4 .OR. NFLUID.EQ.7 .OR. NFLUID.EQ.10)
  * DSAT = DNLQA(T2SATL*459.67)
IF(NFLUID.EQ.5 .OR. NFLUID.EQ.9) DSAT=DNLIGB(T2SATL*459.67)
IF(NFLUID.EQ.6 .OR. NFLUID.EQ.8) DSAT=DNLIQC(T2SATL*459.67)

SVSATL=1.0/DSAT
SVSATVR*(T2SATL*459.67)/P2
MAX4=0

140 CONTINUE
ESTP2=ESTP(T2SATL*459.67, SVSATV, 0.0)
DELP=ESTP2-P2
IF(ABS(DELP) .LT. 0.0) GO TO 185
MAX4=MAX4+1
IF(MAX4 .LT. 40) STOP "FREON MAX4"

SVSATV=SVSATV*(1.0+DELP/P2)
GO TO 180

185 CONTINUE
H2SATV=HVAP(T2SATL*459.67, SVSATV, P2)
SDPDT=SPRST(P2, T2SATL*459.67)
SDHVAP=SDPDT*(T2SATL*459.67)*(SVSATV-SVATL)

H2SATL=H2SATV+SDHVAP
IF(FLAG .EQ. 0) PRINT 13, H2SATL

13 FORMAT("M2SATL=SATURATED L-IQUID ENTHALPY OF WORKING FLUID (BTU/LB *), T65,1PE14.8")
P1 = PANE
IF(PONP .LT. 0.0) GO TO 199

C**ASSUME A VALUE OF INLET PRESSURE P1 TO BEGIN ITERATION
P1=100.0
IF(NFLUID.EQ.4) P1=200.
IF(NFLUID.EQ.10) P1=600.
IF(NFLUID.EQ.8) AND. T1 .LT. 250.) P1=30.

199 MAX5=0

200 CONTINUE
IF(.NOT. SUPER) GO TO 203
IF(PONP .LT. 0.0) GO TO 201
P1=100.0
IF(1.5*PCR/T1 .GT. 1000.0) P1=1.5*PCR

201 PINCH=PINCUS

203 CONTINUE
MAX = MAX4+1
IF(FLAG .EQ. 2) PRINT 3, MAX, P1, PINCH

3 FORMAT("ITERATION NO."14." FOR P1/ 
"TOP = TURBINE INLET PRESSURE (PSIA)" , T65,1PE14.6/
"PINCH=HEAT EXCHANGER RICH POINT (DEG. F)" , T65,1PE14.6)

C**REQUIRE DETERMINING M1, NEED TO FIND V1
C**ESTIMATE V1 USING IDEAL GAS LAW
V1=R*(T1*459.67)/P1

MAX1=0

220 CONTINUE
PESTV=ESTP(T1*459.67, V1, 0.0)
IF(ABS(PESTV-P1) .LT. 0.5) GO TO 240
MAX1=MAX1+1
IF(MAX1 .LT. 40) STOP "MAX1"
V1=V1*(1.0+(PESTV-P1)/P1)
GO TO 220

240 CONTINUE
H1=HVAP(T1*459.67, V1, P1)
S1=SVAP(T1*459.67, V1, 0.0)

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IF (FLA+EQ.2) PRINT 2, V1, H1, S1
2 FORMAT('TURBINE INLET SPECIFIC VOLUME (CU. FT/LB)"*,T65*1PE14.6/
+"H1=TURBINE INLET ENTHALPY (BTU/LB)"*,T65*1PE14.6/
+"S1=TURBINE INLET ENTROPY (BTU/LB/DEG. F)"*,T65*1PE14.6/
+S1=ITERATION TO DETERMINE TEMPERATURE AND SPECIFIC VOLUME AT STAGE
+S1EST")
S1EST<1

C*ERROR FOR T2EST AND V2EST USING MATHLIB KNOWING P2 AND S2EST
N=2
FA(1)=1000
FA(2)=0.1
TOL=0.1
PRINT=0
IP(1)=100
FX(1)=659.67
IF (NFLUID .EQ. 4 .OR. NFLUID .EQ. 5) FX(1)=659.67
FX(2)=1.0
TERM1=2
TERM2=S2EST
MAEX=0

254 CONTINUE
SAVT(1)=0.0
SAVT(2)=0.0
JFLAG=N
CALL RONWT(FX, N, EFTPV2, PRNT, TOL, IP, RMS, FA)
IF (JFLAG .EQ. 0) GO TO 255

C*ERROR
MAEX=MAXE=1
IF (MAXE .GT. 10) STOP "REON = MAEX"
FX(1)=SAVT(1)-DEL(1)
FX(2)=SAVT(2)
GO TO 254

255 CONTINUE
T2EST=FX(1)=459.67
V2EST=FX(2)
IF (FLA+EQ. 2) PRINT 1, T2EST, V2EST
1 FORMAT('"T2EST=IDEAL TURBINE EXHAUST TEMPERATURE (DEG. F)"*,T65*
+"V2EST=IDEAL TURBINE EXHAUST SPECIFIC VOLUME (CU. FT/LB)"*,T65*
+"J2EST=VAR(T2EST*459.67, V2EST, P2)
CALL WFFRT(GISO, M2, SOPEH, GPKW, T1, H1, H2EST, H2SATL, TCRIT)
IF (SUPER) GO TO 340

C*DETERMINING T4 ASSUME STARTING VALUE OF 100
T4=100.
MAX2=0
IF (FLA+EQ. 2) PRINT 9, WISO, H2EST, H2
9 FORMAT("WISO-WORKING FLUID FLOW RATE (LB/HR)"*,T65*1PE14.6/
+"H2EST=IDEAL TURBINE EXHAUST ENTHALPY (BTU/LB)"*,T65*1PE14.6/
+"H2ENTHALPY (BTU/LB)"*,T65*1PE14.6)

260 CONTINUE
PLOG=VRESA(T4*459.67)
P4EST=P0
IF (ABS(P4EST - P1) LT. 0.5) GO TO 280
MAX2=MAX2 + 1
IF (MAX2 .GT. 40) STOP "MA22"
T4=T4*(1.0 + 0.5*(P1-P4EST)/P1)
SUBROUTINE FREON

GO TO 260

260 CONTINUE
T5=T4
P4=P1
P5=P1

C**DETERMINE VS TO BE USED IN CALCULATING HS
VS=R*(T5+459.67)/P5
MAX3=0

300 CONTINUE
P5TV=GSTAP*(T5+459.67, VI, 0, 0)
IF(ABS(P5TV-P5) .LT. 0.5) GO TO 320
MAX3=MAX3 + 1
IF(MAX3 .GE. 4) STOP "MAX3"
VS=VS*(1.0 + (P5TV-P5)/P5)
GO TO 300

320 CONTINUE
V5=HVAP*(T5+459.67, VS, P5)
DPS=VAP5B(P5, T5+459.67)
IF(NFLUID .EQ. 6 OR, NFLUID .EQ. 9) DLS=DNLIGB(T5+459.67)
IF(NFLUID .EQ. 6 OR, NFLUID .EQ. 8) DLS=DNLIGC(T5+459.67)
VLS=1.0/DLS
DVAP=DPS/DV5P*(T5+459.67)*(V5*VLS)
$TERM/NE H3 AND 73

340 CONTINUE
P3=P1
CALL PUMP(T3, H3, T2SATL, H2SATL, P2, P3, NFLUID, SVSATL)
IF(SUPFR) GO TO 400
WT3=WTPNOHP
ESTW7104WWT3*(HI H3)/(MI.H4)WT1*(113H4)/(RIH4)
GRATUOiSO4(H1°H3)Y(W71ES)WT4)
IF(FLUNER.2) PRINT Alo V5 HS. OPOTo
OLS VLS, DMVAPo H4. 1073.
ESTW74, GM4T
400OPOT.DERIVATIVE of PRESSURE AS FUNCTION
RE14.6/HOOL5.SATURATED VAPOR DENSITY
336x289
0VL9"IATISRATED VAPOR SPECIFIC VOLUME
336x286}(CU.FT/LB)4.765,1PE14.06/
110DHVARHEAT OF VAPORIZATION (8TU/LB)".765,1PE1406/
ROWT3wATER TEMPERATURE At pREMEATER
329x258INNER EXIT (F).T65,1RE13.6/6XH(EXCLUDINO SALT)"/
105x241A0WoRRtNO FLUID CONOIT/ON/SA/0TEMPERATURES",
2 TIOOTITURGINE INLET",
3 T100T2EST-TURBINE OUTLE (IDEAL)",
4 T10"T3-PREHEATER INLET",
5 T10"T4-VAPORIZER INLET",
6 T10"T5-SUPER HEATER INLET",
7 T10"T6-TURBINE INLET",

PRINT T, MAX, Q130, QMAT, T1, T2EST, T3, T4, T5, P1, P2, #4, P5,
*V1, V2=EST, VS
7 FORMAT("10CYCLE CONDITIONS FOR ITERATION NO.,IS/NOQISO-WORKING FLU"
+ID FLOW RATE (LB/HR)".740,1PE14.6/40WAT-GEOTHERMAL FLUID FLOW RAT"
+U (LB/HR)".740,1PE14.6/6XH(EXCLUDING SALT)"
+WORKING FLUID CONDITIONS"/
2 T10"T1-TURBINE INLET",
3 T10"T2EST-TURBINE OUTLE (IDEAL)",
4 T10"T3-PREHEATER INLET",
5 T10"T4-VAPORIZER INLET",
6 T10"T5-SUPER HEATER INLET",
7 T10"T6-TURBINE INLET"

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8 T10 = "92-TURBINE EXHAUST", T40,1PE14.6/
9 T10 = "4-A VAPORIZER INLET", T40,1PE14.6/
A T10 = "5-SUPER HEATER INLET",
"SPECIFIC VOLUME",
B T10 = "1-TURBINE INLET", T40,1PE14.6/
C T10 = "2EST-TURBINE OUTLET (IDEAL)", T40,1PE14.6/
E T10 = "5-SUPER HEATER INLET", T40,1PE14.6/
PRINT 10, H1, H2, H2EST, H3, H4, H5, S1, S2EST, WT1, WT3,
W4, ESTWT4
10 FORMAT(
1 T10 = "1-TURBINE INLET", T40,1PE14.6/
2 T10 = "2-TURBINE EXHAUST", T40,1PE14.6/
3 T10 = "2EST-TURBINE OUTLET (IDEAL)", T40,1PE14.6/
4 T10 = "3-PREHEATER INLET", T40,1PE14.6/
5 T10 = "4-VAPORIZER INLET", T40,1PE14.6/
6 T10 = "5-SUPER HEATER INLET", T40,1PE14.6/
7 T10 = "6-TURBINE INLET", T40,1PE14.6/
8 T10 = "2EST-TURBINE OUTLET (IDEAL)", T40,1PE14.6/
9 "GEO-HEAT FLUID CONDITIONS"/"WATER TEMPERATURES /
A T10 = "W1-TURBINE INLET", T40,1PE14.6/
B T10 = "W3-PREHEATER INLET", T40,1PE14.6/
C T10 = "W4-PREHEATER EXIT (PRESET)", T40,1PE14.6/
D T10 = "W5-PREHEATER EXIT (ACTUAL)", T40,1PE14.6/
IF(PONEP404) GO TO 400
CALL RmNEIDELOLD DELNEW, POLO, PI, P17, WT4 ESTINT44 PINCI4P
IFLAG, IPCNT)
IF(MAX .GT. 40) STOP "MAX"
GO TO(360* 200, 4110)9 IFLAG
360 CONTINUE
PINCHP = (WT4 - (H1 = H4) - WT1 = (H3 = H4) - T4 = (H1 = H3)) / (H1 - H3)
IF(PINCHP = LE. 30) GO TO 200
C*SUPERCritical CYCLE
IF(FLAG = EQ. 4) PRINT 11; PINCHP, PI
11 FORMAT("SUPERCRITICAL"/"UPINCHP[P1=""1P2E14.6)
SUPER = TRUE,
GO TO 200
400 CONTINUE
C**KNOWING P2 AND H2 DETERMINE T2 AND V2
N=2
TERM1=p2
TERM2=h2
PA(1,1)=1000.
TOL=0.01
PRINT=0
IF(1)=0
FX(1)=2EST + 459.67 * 25 + 0
FX(2)=V2EST
460 CONTINUE
SAVTV(1)=0.0
SAVTV(2)=0.0
JFLAG=0
MAXF=0
OLDRI=0.0
CALL RnNwT(FX, N, FFYV2V2, PRNT, TOL, IP, RMS, FA)
IF(JFLAG = EQ. 0) GO TO 47v
C**ERROR
MAXF=MAXF+1
SUBROUTINE FREON

IF (MAXF .GT. 10) STOP "FREON - MAXF"
FX(1) = SAVTV(1) - BEL(1)
FX(2) = SAVTV(2) - BEL(2)
IF (FX(2) .LE. 0.0) FX(2) = SAVTV(2)
GO TO 460

470 CONTINUE
T2 = FX(1) - 459.67
V2 = FX(2)
RETURN
END
FUNCTION FUG

FUNCTION FUG(RHO,S1,S2,S3,S4,S5,GAMMA)
FUGASITY EQUATION
GAM2=GAMMA*RHO**2
F1=1.0-S2=GAM2=GAMMA**2*H0**4
F2=EXP(-GAM2)
FUG=S1*ALOG(RHO*S1)+2.*RHU=S2+1.5*RHO**4*S3+1.2*RHO**5*S4*S5/
1 GAMMA*(1.0-F1+F2)
RETURN
END
FUNCTION HEQST

FUNCTION HEQST(T, D)
COMMON /BUTANE/ A0, AZERO, A1, A2, AB, AB3, A4, BB, BZERO, BB1,
* BB2, AB3, BB4, CZERO, DZERO, EZERO, SMA, SMAB, SMC, SMD, APMAB,
* GAMMA, PHI, BUTMW, ER
TSO=T**2
DSQ=D**2
RT=RT

GAM02=GAMMA*DSQ
IF (GAM02>2.25) EXPGAM = 0.
IF (GAM02>2.45 .AND. GAM02<0.001) EXPGAM=EXP(-GAM02)
HEQST=(AZERO+RT-AZERO)*CZERO/TSO**3.39EZERO/T**1.544*SMO**2.*
* 2.415SMR*RT...4.41SMA**4.*SMD/SMW/TSO(3.*SMA**6.35GAM02-2.4104)EXPGAM)
RETURN
END
FUNCTION HIDEAL

FUNCTION HIDEAL(T)
COMM IDEAL GAS ENTHALPY
COMM COMMON FOR THE BUTANE CONSTANTS
COMMON /BUTANE/ AB, AZERO, AB1, AB2, AB3, AB4, BB, BZERO, BB1,
* BB2, AB3, AB4, CZERO, DZERO, EZERO, SMA, SMBB, SMC, SMD, ALPHAB,
* GAMMA, PHI, BUTMW, BB
HIDEAL=AB + AB1*T + AB2*T**2 + AB3*T**3 + AB4*T**4
RETURN
END
FUNCTION HTAREA

FUNCTION HTAREA(QISO, M1, M2, U, WT, FT)

C FUNCTION DETERMINES AVERAGE HEAT TRANSFER AREA

C QISO - FLUID FLOW RATE
C M1 - ENTHALPY AT INLET
C M2 - ENTHALPY AT EXHAUST
C U - OVERALL HEAT TRANSFER COEFFICIENT
C TLMD - LOG MEAN TEMPERATURE DIFFERENCE

C TLMD = (WT-FT)/ ALOG(WT/FT)
C HTAREA = QISO * (M1-M2)/ U * TLMD

C RETURN
C END
FUNCTION HTCOEF

FUNCTION HTCOEF (COND, DIA, PRANO, REYNO, EXPN)

FUNCTION DETERMINES HEAT TRANSFER COEFFICIENT
COND - THERMAL CONDUCTIVITY
DIA - DIAMETER
PRANO - PRANDTL NUMBER
REYNO - REYNOLDS NUMBER
EXPN - EXPONENT, = 0.3 WHEN FLUID IS BEING COOLED
      = 0.4 WHEN FLUID IS BEING HEATED

HTCOEF = 0.023 * COND / DIA * PRANO**EXPN * REYNO**0.8

RETURN
END
FUNCTION HTLNGH

FUNCTION HTLNGH(HTA, DIAO, TUBEN)

C FUNCTION COMPUTES THE HEAT TRANSFER LENGTH
C HTA- AVERAGE HEAT TRANSFER AREA
C DIAO- OUTSIDE DIAMETER
C TUBEN- NUMBER OF TUBES
C
C HTLNGH= HTA / 3.1416 / DIAO / TUBEN
C
RETURN
END
FUNCTION HVAP(T, V, P)
COMMON /BINARY/ SUPER, NFLUID, WT1, WT2, WT3, WT4,
*ESTWT, WM1, WM3, CWT1, CWT2, CPW1, CPW2, T1,
*T2, T2EST, T2SATL, T3, T4, T5, P1, P2,
*T3, PA, P5, H1, H2, M2EST, M2SATL, M3,
*H4, MB, S1, SCEST, S3, S4, SVSATL, D1,
*DB, D2EST, V1, V2EST, PINCHP, VVELL, DIATL, DIATL,
*VVELV, DIAV, DIAOV, WVELS, DIAS, DIAOS, WVELSC, DIASC,
*DIAOSC, PVELC, DIAC, DIAOC, FOUFLF, FOUFLF, BA, QISQ,
*MTAEX(7), HTACON, TMPRCH, WFOUR, APPCON, APPCT, APFCH,
*DELMX, PINSUB, PINSUP, PUNEP, RANGE,
LOGICAL SUPER
REAL JAY
REAL KAY

C***COMMON FOR THE FREON CONSTANTS
C, A4, B4, C4, A5, B5, C5, A6, B6, C6, KAY, ALPHA, HCA, HCB, HCC, HCD, HCF, XFR
**YFR = PC, TL, VC
JAY = V, 185053
ONE = 49
TWO = 2, ONE = 2
THREE = 2, ONE = 3
FOUR = 4, ONE = 4
EXPAM = 9, 0
PARTA = 9, 0
AV = ALPHA * V
IF (AV .GT. 230.) GO TO 220

C***COMMON FOR R12, R21, R113 AND R114 AND A6 AND ALPHA = 0.0
IF (ALPHAEQ 0.0) GO TO 200
EXPAM = EXP (AV)
PARTA = (1.0 / EXPAM) * (ALPHA * A6)

200 CONTINUE
PART1 = HCA * T + (HCB * T ** 2) / 2 + (HCC * T ** 3) / 6 + (HCD * T ** 4) / 24 + HCF / T,
1 + JAY * P
PART5 = 1 / (ONE + A3 / TWO + A4 / THREE + A5 / FOUR)
PART6 = JAYEXP (-KAY * T / TC) * (1.0 + KAY * T / TC)
PART7 = 2 / (ONE + C3 / TWO + C4 / THREE)
IF (NFLUID .EQ. 1) PART7 = PART7 * C4 / THREE
HVAP = PART1 * JAY * (PART5 * PART7) * PART6 * PART7 / XFR
RETURN
END
SUBROUTINE LOSSES

SUBROUTINE LOSSES(TKWLOS,CTA,QW1,QR1,PWH,IPUMP,RPKLOS,
* NPW,DMPKW,*FCPEP)

COMMON /FLAGS/ FLAG, NEXT, FLAG1
INTEGER FLAG

THIS ROUTINE CALCULATES THE TOTAL KILOWATT LOSS (TKWLOS)
ATTRIBUTABLE TO
1. COOLING TOWER FANS (FAVOR)
2. CONDENSATE PUMPS (CPKW)
3. COOLING WATER PUMPS (LJKW)
4. REINJECTION PUMPS (RJNW)
5. ROASTER PUMPS FOR OVEMPRESSURIZING WATER (RPKLOS)
6. DOWNSHOLE PUMPS TO PREVENT TWO PHASE FLOW (DMPKWT)

INPUT PARAMETERS
CTA = COOLING TOWER AIR FLOW RATE
QW1 = COOLING WATER FLOW RATE INTO THE TOWER
QR1 = TOTAL COOLING WATER REQUIREMENT
RC1 = REINJECTION RATE
WP = PRESSURE OF WATER LEAVING GROUND

REAL MDF,MEFF

CONSTANTS
PT = TOTAL PRESSURE
DATA PT / 100 /
EFFT = TOTAL FAN EFFICIENCY
DATA EFFT / .79 /
MPF = POWER FACTOR
DATA MDF / .90 /
MEFF = COOLING TOWER FAN MOTOR EFFICIENCY
DATA MEFF / .90 /
CPOM = CONDENSATE PUMP PRESSURE (FT. OF H2O)
DATA CPOM / 45.0 /
CPEFF = CONDENSATE PUMP EFFICIENCY
DATA CPEFF / 0.85 /
CWHM = COOLING WATER PUMP PRESSURE (FT. OF H2O)
DATA CWHM / 92.269 /
CWEFF = COOLING WATER PUMP EFFICIENCY
DATA CWEFF / 0.85 /
FACTOR = DATA FACTOR / 2.6572E6 /
RJEFF = REINJECTION PUMP EFFICIENCY
DATA RJEFF / .80 /

COOLING TOWER FANS
FAN REQUIRED HORSEPOWER
FHPCTA = FHP x PT / 6356.0 / EFFT

KW REQUIRED FOR FANS
FKW = FHPCTA / 0.7457 / (MEFF * MPF)

CONDENSATE PUMPS TO COOLING TOWER
CPKWQW1 = CPOM / CPEFF / FACTOR / MPF

COOLING WATER PUMPS
CWHQWC = CWHM / CWEFF / FACTOR / MPF
SUBROUTINE LOSSES

C REINJECTION PUMPS
C RJDH = REINJECTION PUMP PRESSURE
C CONVERT FROM PSIA TO FT$^2$ OF H2O
   RJDH = 50.$* = 27.6807 / 12.$0
   RJKW = 0.
   IF(1UPMP$* FE 0)RJKW = ORI$*RJDH/RJEFF/FACTUR$*MPF
C DOWNHOLE PUMPS
   DHPKW$* = FLOAT(NPW$*)*DHPKW$*(1$*+FRCEPW)
C
total kw loss
   TKWLOS$* = FKWR$*C*CPKW$*C*CWKW$*RJKW$*RPKLUS$*DHPKW$*
   IF(FLAG$* FE 0) WRITE(6$*,1) FKWR$*,CPKW$*,CWKW$*,RJKW$*,RPKLUS$*,DHPKW$*
RETURN
1 FORMAT("KILOWATT LOSSES", F5$*,1PE14$*,6)
1 6X,"FANS", F5$*,1PE14$*,6
2 6X,"CONDENSATE PUMPS TO COOLING TOWER", F5$*,1PE14$*,6
3 6X,"COOLING WATER PUMPS", F5$*,1PE14$*,6
4 6X,"REINJECTION PUMPS", F5$*,1PE14$*,6
5 6X,"BOOSTER PUMPS TO OVEPRESSURIZE WATER", F5$*,1PE14$*,6
6 6X,"DOWNHOLE PUMPS TO PREVENT TWO PHASE FLOW", F5$*,1PE14$*,6
END
FUNCTION OHTCOE

FUNCTION OHTCOE(D1AO, HTCI, DIA, KALUM, HTCN, FOULFI, FOULFO)

FUNCTION DETERMINES OVERALL HEAT TRANSFER COEFFICIENT

DIAn= OUTSIDE DIAMETER
HTCn= INSIDE HEAT TRANSFER COEFFICIENT
DIA= INSIDE DIAMETER
KALUm= THERMAL CONDUCTIVITY OF CARBON STEEL
HTCN= OUTSIDE HEAT TRANSFER COEFFICIENT
FOULFI=INSIDE FOULING FACTOR
FOULFO=OUTSIDE FOULING FACTOR

REAL KALUm

OHTCOE= (U1AO/HTC1/DIA + U1AO/2.0/KALUm + ALOG(D1AO/DIA) + 1.0/HTCN + FOULFI + FOULFO)**(-1.0)

RETURN
END
SUBROUTINE OPBIN
EXTERNAL BFT20P
COMMON /NAMLIST/ LL1, LL2, LL3, LL4, LL5, LL6,
LL7, LL8, LL9, LL10, LL11, LL12, NUMFTF, ITEMS,
AGFS1, AGFS2, AGFS3, AGFS4, AGFS5,
PM, CACO3, WELS, CASPC, PDRAT, NUMTP, WELSPC,
PMNTH, SMFRC, FRONR, NHC, LHC3, LHC4, LHC5, LHC6,
LHC7, LHC8, LHC9, LHC10, LHC11, LHC12,
COMMON /NAMIL/ LL1, LL2, LL3, LL4, LL5, LL6,
LHC7, LHC8, LHC9, LHC10, LHC11, LHC12,
COMMON /NAM/ LL1, LL2, LL3, LL4, LL5, LL6,
LHC7, LHC8, LHC9, LHC10, LHC11, LHC12,
COMMON /NAMLIST/ LL1, LL2, LL3, LL4, LL5, LL6,
LHC7, LHC8, LHC9, LHC10, LHC11, LHC12,
COMMON /NAMLIST/ LL1, LL2, LL3, LL4, LL5, LL6,
LHC7, LHC8, LHC9, LHC10, LHC11, LHC12,
COMMON /NAMLIST/ LL1, LL2, LL3, LL4, LL5, LL6,
LHC7, LHC8, LHC9, LHC10, LHC11, LHC12,
COMMON /NAMLIST/ LL1, LL2, LL3, LL4, LL5, LL6,
LHC7, LHC8, LHC9, LHC10, LHC11, LHC12,
COMMON /NAMLIST/ LL1, LL2, LL3, LL4, LL5, LL6,
LHC7, LHC8, LHC9, LHC10, LHC11, LHC12,
COMMON /NAMLIST/ LL1, LL2, LL3, LL4, LL5, LL6,
LHC7, LHC8, LHC9, LHC10, LHC11, LHC12,
COMMON /NAMLIST/ LL1, LL2, LL3, LL4, LL5, LL6,
LHC7, LHC8, LHC9, LHC10, LHC11, LHC12,
COMMON /NAMLIST/ LL1, LL2, LL3, LL4, LL5, LL6,
LHC7, LHC8, LHC9, LHC10, LHC11, LHC12,
COMMON /NAMLIST/ LL1, LL2, LL3, LL4, LL5, LL6,
LHC7, LHC8, LHC9, LHC10, LHC11, LHC12,
COMMON /NAMLIST/ LL1, LL2, LL3, LL4, LL5, LL6,
LHC7, LHC8, LHC9, LHC10, LHC11, LHC12,
COMMON /NAMLIST/ LL1, LL2, LL3, LL4, LL5, LL6,
LHC7, LHC8, LHC9, LHC10, LHC11, LHC12,
COMMON /NAMLIST/ LL1, LL2, LL3, LL4, LL5, LL6,
LHC7, LHC8, LHC9, LHC10, LHC11, LHC12,
COMMON /NAMLIST/ LL1, LL2, LL3, LL4, LL5, LL6,
LHC7, LHC8, LHC9, LHC10, LHC11, LHC12,
COMMON /NAMLIST/ LL1, LL2, LL3, LL4, LL5, LL6,
LHC7, LHC8, LHC9, LHC10, LHC11, LHC12,
COMMON /NAMLIST/ LL1, LL2, LL3, LL4, LL5, LL6,
LHC7, LHC8, LHC9, LHC10, LHC11, LHC12,
COMMON /NAMLIST/ LL1, LL2, LL3, LL4, LL5, LL6,
LHC7, LHC8, LHC9, LHC10, LHC11, LHC12,
SUBROUTINE OPBIN

TOL=0.0
PRNT=0
IP(1)=10
IF ( P1 .LT. 150.) FX(1)=0.3*P1*500.
IF ( P1 .GT. 150.) FX(1)=0.3*P1*575.
FX(2)=P1
TERM=1
TERM=2
CALL RNW((FX*N+HFT20)*PRNT,TOL,IP,RMS,PA)
T1= FX(1) = 459.67
D1=FX(2)
H1=EST(1.459.66.01)/PHI+HIDEAL(T1+459.67)
S2EST=S1
N=2
FX(1)=1000.
TOL=0.0
PRNT=0
IP(1)=10
FX(1)=459.67
FX(2)=0.1
TERM=P2
TERM=2
CALL RNW((FX*N+HFT20)*PRNT,TOL,IP,RMS,PA)
T2EST=FX(1) = 459.67
D2EST=FX(2)
H2EST=EST(T2EST+459.67)/PHI+HIDEAL(T2EST+459.67)
CALL WFFNT(01+H2,SUPEP,AH+1+H2EST,H2SATL,TCRIT)
T=TPR+FALID(T1+D1+D2+D3+D4+D5+FLUID)
H=EST(T1+459.67,TL10)/PHI+HIDEAL(T1+459.67)
PINCHP=15
OM=0.3*(M1-H1)/(WT1-T4-PINCHP)/1.01
WT=WT1-459.67*(H1-H2SATL-5)/2198*(M2-H2SATL)*026417/OM
*1.01*(WT)-TZER0*(TZERO+459.67)*ALOG((WT1+459.67)/(TZERO+459.67))
IF (UEFFM .LT. UEFFM) GO TO en:
UEFFM=UEFF
SP1=1
ST1=T1
PI=P1+200
IF (PI .GT. 520.0) GO TO 100
GO TO 10
20 CONTINUE
GO TO 10
20 CONTINUE
TI=ST1
PINCHP=PINCHP
SUPEP=F
GO TO 200
100 CONTINUE
T1=WT1-200
PINCHP=15
IF (WT1 .GE. 400.) GO TO 140
S1=11
N5=0
O1=.23
11 CONTINUE
N5=N5
S1=EST(EST(T1+459.67.01)/PHI+SIDEAK
SUBROUTINE OPBIN

S1EST=FST(T+459.67*01)/HII+SIUEAL(T+459.67)
IF(AHS+S1EST-S1) LT 0.01 GO TO 120
IF(Ns GT 10) STOP "NS MAX"
D1=D1+(S1EST-S1)/S1
GO TO 110

120 CONTINUE
P1=EFFUS(T+459.67*01+0.1)
NOFP=1
IF(P1 LT 500) GO TO 200

140 NOFP=500
200 CONTINUE
RETURN
END
FUNCTION PEQST

FUNCTION PEQST(T, D, P)

C***COMMON FOR THE BUTANE CONSTANTS

COMMON /BUTANE/ AB, AZERO, AB1, AB2, AB3, AB4, BB, BZERO, BB1,
       BB2, AB3, AB4, CZERO, DZERO, EZERO, SMA, SHBB, SHC, SHD, ALPHAB,
       GAMMA, PMI, BITMW, BR

RT=AB*T
TSO=T**2
DSO=D**2
GAM2D=GAMMA*D**2
IF(GAM2D.GT.225.) EXPGAM=.01
IF(GAM2D.LT.0.001) EXPGAM=1.
IF(GAM2D.LE.2.5*AND. GAM2D.GE.0.001) EXPGAM=EXP(-GAM2D)
PEQST=RT*DSO*(AZERO*R1-AZERO-CZERO/TSO+DZERO/T**3-EZERO/T**4)
1+D**2*(SMAB*RT-SMA-SMD/1.)*ALPHAB*D**3*(SMA*SMD/T)
2.0+DSO/TSQ*(1.0+GAM2D)*EXPGAM = P
RETURN
END
SUBROUTINE PONE

SUBROUTINE PONE(DOLOLD, DELNEW, POLD, P1, PCRT, WT4, ESTWT4,
* PINCHP, IFLAG, IRCNT)

C***ROUTINE DETERMINES IF P1 IS CONVERGING AND IF IT ISN'T, THE PINCHP
C***IS INCREASED).

DELNEW=ESTWT4 - WT4
POLD=P1

IF (DELOLD .EQ. 0.0) GO TO 200
 IF (ABS(DELNEW) .LT. 18.) UC TO 300
 IF (SIGN(1*DELOLD) .NE. SIGN(1*DELNEW)) GO TO 240
 IF (ABS(DELOLD) .LT. ABS(DELNEW)) GO TO 240

C***STILL CONVERGING PROPERLY

200 CONTINUE

DELOLD=DELNEW
P1=P1*(1.0 + (WT4-ESTWT4)/WT4)
IF (P1 .LE. 0.0*PCRT) GO TO 280

240 CONTINUE

C***ESTWT4 IS MOVING AWAY FROM WT4 OR THE 2 DELTAS NO LONGER HAVE THE
C***SAME SIGN

IFCNT=0
DELOLD=0.0
PINCHP=PINCHP + 10.0
IF (PINCHP .LE. 30.0) GO TO 280
IFLAG=1
RETURN

280 CONTINUE
IFLAG=2
RETURN

300 CONTINUE
IFLAG=3
RETURN
END
FUNCTION PRNDTL

FUNCTION PRNDTL(T)

C FUNCTION TO DETERMINE PRANDTL NUMBER AT TEMPERATURE T
C COEFFICIENTS WERE DETERMINED BY MARTHA
C
C PRNDTL = 1.19612 + 10.47801*T + 2.86576*T**2 + .25099*T**3
C
RETURN
END
FUNCTION PTWO

FUNCTION PTWO(P, T, RHOL)

C***FUNCTION TO DETERMINE SATURATION PRESSURE

C***RHOL IS USED BY MAIN PROGRAM TO DETERMINE SVSATL AND H2SATL

C***COMMON FOR THE BUTANE CONSTANTS

COMMON /FLAGS/ FLAG, NEXT, FLAG1

COMMON /BUTANE/ AB, AZER0, AB1, AB2, AB3, AB4, BB, BZERO, BB1,
* BB2, AB3, AB4, CZERO, DZERO, EZERO, SMA, SMB, SMC, SMO, ALPHAB,
* GAMMA, PHI, BUTMG, BR

200 CONTINUE

C***CALCULATE CONSTANTS USED BY SUBROUTINE RHOITR

S1=AB * T
S2=BZERO*S1-AZERO*CZERO/T**2-DZERO/T**3-EZERO/T**4
S3=SMB*S1 - SMA - SMO/T
S4=ALPHAB*(SMA + SMO/T)
S5=SMC/T**2

C***DETERMINE RHO IT IS USED BY subroutine RHOITR

BEGINP=0.7
DRHO=0.01
IFLAG=-1
CALL RHOUTR(RHOL,IFLAG,T,*S1,S2,S3,S4,S5,GAMMA,BEGRHO,DRHO,PP)
IF(IFLAG.EQ.1)STORRPTWO=RHOL FAILED TO CONVERGE WHILE DETERMINING 
P2

C***DETERMINE RMOV

BEGINP=0.001
DRHO=0.0001
IFLAG=-1
CALL RMOVTR(RHOL,IFLAG,T,*S1,S2,S3,S4,S5,GAMMA,BEGRHO,DRHO,PP)
IF(IFLAG.EQ.1)STORRPTWO=MOV FAILED TO CONVERGE WHILE DETERMINING 
P2

C***DETERMINE FUGASITIES

FV=FUG(RHOL,T,*S1,S2,S3,S4,S5,GAMMA)
FL=FUG(RHOL,T,*S1,S2,S3,S4,S5,GAMMA)
FRATIO=FL/FV
IF(ABS(1.0+FRATIO).LT.0.001) GO TO 300
PPE*(0.9+FRATIO/14.0)
GO TO 300

300 CONTINUE

IF(IFLAG.EQ.2) PRINT 4, P, T, RHOL, RHOL, FL, FV, FRATIO
4 FORMAT("DETERMINATION OF SATURATED DENSITIES AT TURBINE EXHAUST P 
RESURE"/"FINAL VALUES"/ "P= TURBINE EXHAUST PRESSURE (PSIA)"/ T601PE14.6/ 
"T= TURBINE TEMPERATURE OF SATURATED WORKING FLUID (DEG. R)"/ T60,
+1PE14.6/ "RHOL=SATURATED LIQUID DENSITY (LB MOLD/CU. FT)"/ T601PE14.6/ 
"RHOL=SATURATED VAPOR DENSITY (LB MOLD/CU. FT)"/ T601PE14.6/ 
"F=LIQUID FUGACITY"/ T601PE14.6/ 
"F=VAPOR FUGACITY"/ T601PE14.6/ 
"FRATIO=RATIO OF LIQUID TO VAPOR FUGACITY"/ T601PE14.6)

C PTWOP=P 
RETURN 
END
SUBROUTINE PUMP

SUBROUTINE PUMP (T3, H3, T2SATL, H2SATL, P2, P3, NFLUID, SVSATL)

C
PUMP WORK IN COMPRESSING THE WORKING FLUID
SUBROUTINE TO CALCULATE WORKING FLUID TEMPERATURE AND ENTHALPY AT
PREHEATER INLET

OUTPUT VALUES
T3 = WORKING FLUID TEMPERATURE AT PREHEATER INLET
H3 = WORKING FLUID ENTHALPY AT PREHEATER INLET

INPUT VALUES
T2SATL = SATURATED LIQUID TEMPERATURE AT T2
H2SATL = ENTHALPY OF THE LIQUID AT SATURATED LIQUID AT T2
P2 = ROOSTER PUMP INLET PRESSURE
P3 = HEAT EXchanger INLET PRESSURE
NFLUID = WORKING FLUID INDEX
SVSATL = SPECIFIC VOLUME OF THE SATURATED LIQUID AT T2

VALUES ACQUIRED VIA FLUID FUNCTION
WM = MOLECULAR WEIGHT
CPL = SPECIFIC HEAT OF THE LIQUID AT T2

CONSTANTS

PWCON = 0.185053
R = 1.986
ETA = 0.90

SET CONSTANTS

PWCON = 0.185053
R = 1.986
ETA = 0.90

ACQUIRE VALUES FROM FLUID FUNCTION

WM = FLUID(13, NFLUID, 0.0, "PUMP AT WM CALC")
CPL = FLUID(18, NFLUID, T2SATL, "PUMP AT CLP CALC")

CALCULATE DELTA H AND THEREBY H3

DELTH = SVSATL * (P3 - P2) * PWCON / ETA
H3 = H2SATL + DELTH

CALCULATE SPECIFIC HEAT RATIO (SHRAT) AND THEREBY T3

SHRAT = CPL / (CPL - R / WM)
T3 = T2SATL + DELTH * WM * (SHRAT-1.0) / SHRAT / R

RETURN
END
SUBROUTINE QAREA

ROUTINE CALCULATES TOTAL FLOW AREA OF HEAT EXCHANGER TUBES

OUTPUT VALUES
UFLAR = AREA OF STREAM CROSS SECTION
PERIM = WETTED PERIMETER
EQDIA = EQUIVALENT DIAMETER
FLAREA = TOTAL FLOW AREA

INPUT VALUES
DIAM = OUTSIDE DIAMETER
TUBEN = NUMBER OF TUBES

S = 1.75 * DIAM
0.86603 = SIN(1.0472 RADIAN) OR 60 DEGREES
R = S * 0.86603
UFLAR = 0.5 * S * R = (3.1416 * DIAM**2) / 8 * R
PERIM = 3.0 * (S * DIAM) + (3.0 * 3.1416 * DIAM) / 6 * R
EQDIA = 4.0 * UFLAR / PERIM
FLAREA = UFLAR * TUBEN
RETURN
END
FUNCTION OCC

FUNCTION OCC(CKW, TCI, TCEXIT)

ROUTINE CALCULATES THE COOLING WATER REQUIREMENTS (OCC) FOR EITHER
A TURBINE OIL COOLER OR A HYDROGEN COOLER

INPUT PARAMETERS
CKW= KILOWATT EQUIVALENT OF HEAT REMOVED
TCI= COOLING WATER INLET TEMPERATURE
TCEXIT= COOLER EXIT COOLING WATER TEMPERATURE

CP= SPECIFIC HEAT OF WATER AT TCI
T = (TCI + TCEXIT) / 2 + 0 / 100 + 0
CP = 9.01648E-1 + 2.167E-2*T -1.5327E-2*T**2 + 3.7245E-3*T**3

OCC = CKW * 3414.43 / (CP * (TCEXIT - TCI))

RETURN
END
FUNCTION REYNO

FUNCTION REYNO(WRHO, WVEL, DIA, WVISC)

C
C FUNCTION TO DETERMINE REYNOLDS NUMBER

C WRHO= DENSITY
C WVEL= VELOCITY
C DIA= DIAMETER
C WVISC= VISCOSITY

C REYNO= WRHO * WVEL * DIA / WVISC

C RETURN
END
SUBROUTINE RHOITR

SUBROUTINE RHOITR(RHO,IFLAG,Y,P,S1,S2,S3,S4,S5,GAMMA,BEGRHO,DRHO,* 
*PP)

COMMON /FLAGS/ FLAG * NEXT * FLAG1

INTEGER FLAG, IFLAG

C** ROUTINE TO DETERMINE DENSITY OF EITHER VAPOR OR LIQUID

C** WHEN DETERMINING RHOL (IFLAG=1), THE VALUE OF P IS APPROACHED FROM

C** VALUES GREATER THAN ITSELF. WHEN THE SIGN OF PEST-P BECOMES

C** NEGATIVE, THE VALUE OF RHO IS INCREASED BY THE VALUE OF THE LAST

C** DELTA RHO AND DRHO IS REDUCED BY 10.

C** THE OPPOSITE HAPPENS WHEN DETERMINING RHOV (IFLAG=1) BECAUSE P IS

C** BEING APPROACHED BY VALUES LESS THAN ITSELF. SO, WHEN PEST-P BECOMES

C** POSITIVE, RHO IS DECREASED BY DRHO WHICH IS SUBSEQUENTLY DECREASED

MAX = 0

SFLAG = IFLAG

RHO = BEGINRHO + DRHO

200 CONTINUE

MAX = MAX + 1

IF (MAX GT 2000) GO TO 240

RHO = RHO + SFLAG * DRHO

GAM2 = GAMMA * RHO**2

S6 = EXP(-GAM2)

S7 = 1.0 + GAM2

PEST = S1 * RHO * S2 * RHO**2 * S3 * RHO**3 * S4 * RHO**6 * S5 * S6 * S7 * RHO**3

PEST = P

IF (RHO LT 0.3 * AND * IFLAG EQ -1) GO TO 220

IF (ABS(PP) LT .0) GO TO 260

IF (SIGN(1.0, PP) *NE. SIGN(1.0, SFLAG)) GO TO 200

C** MISSED - GO BACK ONE ITERATION AND DECREASE DECREMENT

RHO = RHO - SFLAG * DRHO

DRHO = DRHO / 10.0

GO TO 200

220 CONTINUE

C** RHO IS BELOW THE RANGE FOR RHOL (.54 ... 3), ADJUST THE TEMPERATURE AND

C** START OVER

IFLAG = 1

RETURN

260 CONTINUE

C** DID NOT CONVERGE

IFLAG = 1

RETURN

260 CONTINUE

IFLAG = 1

RETURN

END
FUNCTION SEQST(T,D)

 COMMON /BUTANE/ AB, AZERO, AB1, AB2, AB3, AB4, BR, DZERO, RH1,
     AB2, AB3, AB4, CZERO, DZERO, EZERO, SMA, SM18, SMC, SMD, ALPHAR,
     GAMMA, PHI, BITMW, BR

 TS0=T**2
 TCUER=**3
 DSGB=2
 GAM2=GAMMA/D2
 IF(GAM2.GT.225.) EXPGAM=1.
 IF(GAM2.LT.0.001) EXPGAM=1.
 IF(GAM2.LE.25.*AND.GAM2.GE.0.001) EXPGAM=EXP(-GAM2)

 SEOST=ALOG(D**T)=D*(AZERO-RH2*CZERO/TCUER-3.*DZERO/
     T**4 + 4.*EZERO/T**5) = VSG/2*(SMB+SM18/SM0/TSQ + ALPHAR*SMD* 
     2)**5/E*/TSQ2. SM0/GAMMA/TCUER*(1-1.5*GAM2)*EXPGRAM)
 RETURN
 END
SUBROUTINE SETCON

SUBROUTINE SETCON(NFLUID)
C
**ROUTINE SUPPLIES THE PROPER CONSTANTS TO EITHER COMMON FREON OR
**BUTANE DEPENDING ON THE TYPE OF FLUID
C
COMMON /FRECON/ FREN(40)
COMMON /BUTANE/ BUTN(24)
DIMENSION ALLFRG(40), ALLBTN(24)

C**AL
DATA(ALLFRG(1:1),1:1) = [data values]
C**AR
DATA(ALLFRG(2:1),1:1) = [data values]
C**CL
DATA(ALLFRG(3:1),1:1) = [data values]
C**DL
DATA(ALLFRG(4:1),1:1) = [data values]
C**FL
DATA(ALLFRG(5:1),1:1) = [data values]
C**A
DATA(ALLFRG(6:1),1:1) = [data values]
C**R
DATA(ALLFRG(7:1),1:1) = [data values]
C**C
DATA(ALLFRG(8:1),1:1) = [data values]
C**F
DATA(ALLFRG(9:1),1:1) = [data values]
C**1
DATA(ALLFRG(10:1),1:1) = [data values]
C**R
DATA(ALLFRG(11:1),1:1) = [data values]
C**S
DATA(ALLFRG(12:1),1:1) = [data values]
C**C
DATA(ALLFRG(13:1),1:1) = [data values]
C**2
DATA(ALLFRG(14:1),1:1) = [data values]
C**C
DATA(ALLFRG(15:1),1:1) = [data values]
C**3
DATA(ALLFRG(16:1),1:1) = [data values]
C**A
DATA(ALLFRG(17:1),1:1) = [data values]
SUBROUTINE SETCON

C* C*1
DATA (ALFNG(18,1),I=1,7)/-9.472103E4, .00, 0.0, -1.672612E5,
 0.0, 0.0, 0.0 /
C* C*C
DATA (ALFNG(22,1),I=1,7)/ 0.0, 0.0, 0.0, 0.0, 0.0, 0.0, 0.0, 0.0, 0.0, 9.3798E1 /
C* C*D
DATA (ALFNG(23,1),I=1,7)/ -2.358930E5, 0.0, 0.0, -3.724044E5,
 0.0, 0.0, 0.0 /
C* C*E
DATA (ALFNG(24,1),I=1,7)/ 5.395458E9, 0.0, 6.26324E10, -2.69487E5 /
C* C*F
DATA (ALFNG(25,1),I=1,7)/ -1.84053E-4, 0.0, -1.01531E-5, -3.18142E-4 /
C* C*A6
DATA (ALFNG(26,1),I=1,7)/4.057504E8, 0.0, 0.0, 1.363387E8, 0.0, 0.0, 0.0 /
C* C*R6
DATA (ALFNG(27,1),I=1,7)/4.057504E8, 0.0, 0.0, 1.363387E8, 0.0, 0.0 /
C* C*C6
DATA (ALFNG(28,1),I=1,7)/ 0.0, 0.0, 0.0, 0.0, 0.0, 0.0, 0.0 /
C* C*Y
DATA (ALFNG(29,1),I=1,7)/4.50, 5.475, 0.0, 4.2, 0.0, 3.0, 6.4E19 /
C* C*ALPHA
DATA (ALFNG(30,1),I=1,7)/ 0.0, 0.0, 0.0, 0.0, 548.2, 0.0, 0.0 /
 2.10641E2 /
C* C*HCA
DATA (ALFNG(31,1),I=1,7)/-9.472103E4, 0.0, 0.0, -1.672612E5, 0.0, 0.0, 0.0 /
C* C*HCB
DATA (ALFNG(32,1),I=1,7)/-2.358930E5, 0.0, 0.0, -3.724044E5, 0.0, 0.0, 0.0 /
 0.0, 0.0, 0.0 /
C* C*HCC
DATA (ALFNG(33,1),I=1,7)/-2.123734E7, -2.413896E7, 0.0, 0.0, -1.672612E5, 0.0, 0.0 /
 0.0, 0.0, 2.9421E10 /
C* C*HCD
DATA (ALFNG(34,1),I=1,7)/-3.99018E11, 0.72363E11, 0.0, 0.0, 0.0 /
 0.0, 0.0, 0.0 /
C* C*HCF
DATA (ALFNG(35,1),I=1,7)/-1.1055 /
C* C*X
DATA (ALFNG(36,1),I=1,7)/0.0, 0.0, 39.55555122, 76.24996, 62.4009, 0.0, 0.0, 0.0 /
 25.19, 25.3398211, 195.933 /
C* C*Y
DATA (ALFNG(37,1),I=1,7)/0.0, 0.0, 0.0, 0.0, 0.0, 0.0, 0.0, 0.0, 0.0, 0.0 /
 2.10641E2 /
SUBROUTINE SETCON

* -0.045335  -0.04052  -0.1151  1.151756  -1.40
C**PC
DATA (ALFMG(38,I),I=1,7) /039.5  596.9  /50.0  721.906  495.26
* 473.1  1638.0
C**TC (DEGREES RANKIN)
DATA (ALFMG(39,I),I=1,7) / 048.07  693.30  A12.9  664.50  877.0
* 753.95  219.0
C**VC
DATA (ALFMG(40,I),I=1,7) / 0.028927  0.02870  0.030675  0.030525
* 0.027910  0.02753  0.02683
C**UTANES
C
C***AR
DATA (ALL8TN(1.1),I=1,2) /776.11  -739.17
C***ZERO
DATA (ALL8TN(2.1),I=1,2) /7.7294E4  3.52447E4
C***AR
DATA (ALL8TN(3.1),I=1,2) /0.06472  0.7239
C***AB2
DATA (ALL8TN(4.1),I=1,2) /3.07E-4  -0.601E-4
C***AB3
DATA (ALL8TN(5.1),I=1,2) /0.0271  0.601E-4
C***AR4
DATA (ALL8TN(6.1),I=1,2) /0.0  3.93E-7
C***RR
DATA (ALL8TN(7.1),I=1,2) /0.86401  0.9008
C***ZERO
DATA (ALL8TN(8.1),I=1,2) /1.8789  1.56588
C***AR1
DATA (ALL8TN(9.1),I=1,2) /0.079E-4  10.06E-4
C***AB2
DATA (ALL8TN(10.1),I=1,2) /-2.07E-7  -3.006E-7
C***AR3
DATA (ALL8TN(11.1),I=1,2) /0.885E-10  0.846E-10
C***AR4
DATA (ALL8TN(12.1),I=1,2) /-0.241E-13  0.0
C***ZERO
DATA (ALL8TN(13.1),I=1,2) /1.014E3  1.37436E10
C***DZERO
DATA (ALL8TN(14.1),I=1,2) /8.5317E6  3.33159E11
C***EZERO
DATA (ALL8TN(15.1),I=1,2) /8.40860E13  2.30902E12
C***SMA
DATA (ALL8TN(16.1),I=1,2) /4.79907E4  1.1818E4
C***SMBB
DATA (ALL8TN(17.1),I=1,2) /8.58663  9.14066
C***SMD0
DATA (ALL8TN(18.1),I=1,2) /4.06763E10  7.00044E10
C***SMD
DATA (ALL8TN(19.1),I=1,2) /2.16863E7  5.64238E7
C***ALPHAB
DATA (ALL8TN(20.1),I=1,2) /4.23987  4.00985
C***GAMMA
DATA (ALL8TN(21.1),I=1,2) /7.11486  7.54122
C***PHI
DATA (ALL8TN(22.1),I=1,2) /313.8731773  313.8731773
C***RUTMNW
SUBROUTINE SETCON

DATA (ALLHITN(23,I),I=1,2)/58.121, 58.121 /
C***RB
DATA (ALLHITN(24,I),I=1,2)/ 10.7335, 10.7335 /
IF(INFLUID GT J) GO TO 3un
C***FLUID IS ONE OF THE BUTANES
I=INFLUID-1
DO 220 J=1,24
BEON(J)=ALLHITN(J+I)
220 CONTINUE
CALL BUTANE
RETURN
3un CONTINUE
C***FLUID IS ONE OF THE FREONS
I=INFLUID-J
DO 320 J=1,40
FREN(J)=ALLFRG(J+I)
320 CONTINUE
CALL FREON
RETURN
END
FUNCTION SIDEAL(T)

COMMON / BUTANE / AB1, AB2, AB3, AB4, BB, BZERO, BB1,
- BB2, BB3, BB4, CZERO, DZERO, EZERO, SMA, SMB, SMC, SMD, ALPHAB,
- GAMMA, PHI, BUTNOW, RA
SIDEAL = BB + BB1*T + BB2*T^2 + BB3*T^3 + BB4*T^4
RETURN
END
SUBROUTINE SUBHEX

* QTSO, T1, T2, T3, T4, P4, M1, M2, M3, M4, M5, FOULFI, FOULFO;
* WVEL, DIAl, DIAOL, WVELV, DIAV, DIAOL, WVELS, DIAS, DIAOS,
* N.FluidHTAEFP,1, QNA, WPDS;
* COMMON/FLAGS/ FLAG, NExt, FLAG;
* COMMON/HXOUT/HXRSLT(9,3)
* COMMON/BUTANE/AR, AZERO, AR1, AR2, AR4, BR, AZERO, BR;
* BR2, BB3, BB4, BB5, BB6, BB7, BB8, BB9;
* BB10, BB11, BB12, BB13, BB14, BB15, BB16, BB17, BB18, BB19,
* COMMON/FMFCON/AL, A1, A2, A3, A4, A5, A6, A7, A8, A9, A10,
* A11, A12, A13, A14, A15, A16, A17, A18, A19, A20,
* COMMON/BOFCON/BOF1, BOF2, BOF3, BOF4, BOF5, BOF6, BOF7,
* BOF8, BOF9, BOF10, BOF11, BOF12, BOF13, BOF14, BOF15,
* DIMENSION HTAE(3)

C DETERMINES THE CHARACTERISTICS OF THE SUB-CRITICAL CYCLE HEAT EXCHANGERS
C INPUT PARAMETERS
C QWAT- GEOTHERMAL FLUID FLOW RATE
C WT1- WATER TEMPERATURE AT INLET TO SUPER HEATER
C WT2- WATER TEMPERATURE AT INLET TO PREHEATER
C WT3- WATER TEMPERATURE AT EXIT OF PREHEATER
C WP1- WATER PRESSURE AT INLET TO SUPER HEATER
C WP2- WATER PRESSURE AT OUTLET OF SUPER HEATER
C WP3- WATER PRESSURE AT INLET TO PREHEATER
C WP4- WATER PRESSURE AT OUTLET OF PREHEATER
C Q15- WORKING FLUID FLOW RATE
C T1- WORKING FLUID TEMPERATURE AT TURBINE INLET
C T3- WORKING FLUID TEMPERATURE AT PREHEATER INLET
C T4- WORKING FLUID TEMPERATURE AT VAPORIZER INLET
C T5- WORKING FLUID TEMPERATURE AT SUPER HEATER INLET
C P1- WORKING FLUID PRESSURE AT TURBINE INLET
C P4- WORKING FLUID PRESSURE AT VAPORIZER INLET
C H1- WORKING FLUID ENTHALPY AT TURBINE INLET
C H3- WORKING FLUID ENTHALPY AT PREHEATER INLET
C H4- WORKING FLUID ENTHALPY AT VAPORIZER INLET
C H5- WORKING FLUID ENTHALPY AT SUPER HEATER INLET
C FOULFI-INSIDE FOULING FACTOR
C FOULFO-OUTSIDE FOULING FACTOR
C WVEL- WATER VELOCITY IN THE LIQUID HEAT EXCHANGER (PREHEATER)
C DIAI- INSIDE TUBE DIAMETER IN PREHEATER
C DIAOL- OUTSIDE TUBE DIAMETER IN PREHEATER
C WVELV- WATER VELOCITY IN VAPORIZER
C DIAV- INSIDE TUBE DIAMETER IN VAPORIZER
C DIAOL- OUTSIDE TUBE DIAMETER IN VAPORIZER
C WVELS- WATER VELOCITY IN SUPER HEATER
C DIAE- INSIDE TUBE DIAMETER IN SUPER HEATER
C DIAOS- OUTSIDE TUBE DIAMETER IN SUPER HEATER
C N.FluidINDEX OF WORKING FLUID

C OUTPUT PARAMETERS
C WT2- WATER TEMPERATURE AT EXIT OF SUPER HEATER

C REAL KALUM, MARTPA

C CONSTANTS
SUBROUTINE SUBHEX

C THERMAL CONDUCTIVITY OF CARBON STEEL
DATA KALUM/30.0/
C GRAVITY IN FT/HM*2
DATA GDAV/4.17E8/
VS(WPDS,T) = (4.30361 + 214.277/T - 2532.8/T**2)*(1.**0.276*WPDS)
C ****************************************LIQUID HEAT-EXCHANGER (HEATHEATER) ****************************************

WT34=AVGTMP(WT3+WT4)
T34= AVGTMP(T3+T4)

C PROPERTIES OF WATER AT WT34

WCOND= FLUID(2, 1, WT34, "LIQ HT EX AT WCOND ")
WRHO= FLUID(3, 1, WT34, "LIQ HT EX AT WRHO ")
WREYN= FLUID(4, 0.07566*WPDS, 1, 0)
IF(WPDS, EQ, 0.0) WVIS = FLUID(4, 1*WT34, "LIQ HT EX AT WVIS ")
IF(WPDS, GT, 0.0) WVIS = VS(WPDS, WT34)
WREYNO= REYN(WRHO, WVELL+ DIAL, WVIS)
TMP= WT4/100.0
WCPL = FLUID(1, 1, WT34, "LIQ HT EX AT WCPL ")
WPANO = WCPL = WVIS / WCOND

C INSIDE HEAT TRANSFER COEFFICIENT

HTCI= WTCOEF(WCOND, DIAL, WPANO, WREYN, 0.3)
C NUMBER OF TURES IN HEATEXHEATER

TUREN=TURES(QNA, WRHO+WVELL+DIAL)
C TOTAL FLOW AREA

CALL GAREA(DIAOL, TUBEN, EQDIA, FLAREA)
C WORKING FLUID PROPERTIES AT T34

PEQL= FLUID(1, 1, NFLUID, T34, "LIQ HT EX AT PCPL ")
PCONDL= FLUID(2, NFLUID, T34, "LIQ HT EX AT PCONDL ")
PRHOL= FLUID(3, NFLUID, T34, "LIQ HT EX AT PRHOL ")
PVISL= FLUID(4, NFLUID, T34, "LIQ HT EX AT PVISL ")
IF(PVL, LT, 0.0) PVISL=0.1
PVELL= VELOCY(QISQ, FLAREA, PRHOL)
REYNOL= REYNOL(PRHOL, PVELL, EQDIA, PVISL)
PRANOL= PCPL = PVISL / PCONDL
C OUTSIDE HEAT TRANSFER COEFFICIENT

HTCO= WTCOEF(PCONDL, EQDIA, PRANOL, REYNOL, 0.4)
C OVERALL HEAT TRANSFER COEFFICIENT

U=HTCOE(DIAOL, HTCI, DIAL, KALUM, HTTO, FOULFI, FOULFO)
C AVERAGE HEAT TRANSFER AREA

WT = WT4 - T3
SUBROUTINE SUBHEX

FT = WT3 - T4
HTA = WTAMEA(QISO, H4, H3, U, WT, FT)
HTAEX(1) = HTA

HEAT TRANSFER LENGTH

HTL = HTLENGTH(HTA, DIAOL, TUREN)

HXRLT(1,3) = DIAL
HXRLT(2,3) = DIAOL
HXRLT(3,3) = TUREN
HXRLT(4,3) = ERDIA
HXRLT(5,3) = HTC1
HXRLT(6,3) = HTC0
HXRLT(7,3) = U
HXRLT(8,3) = HTA
HXRLT(9,3) = HTL

*****************************************************************************
VAPOR HEAT EXCHANGER*****************************************************************************

ITERATE TO DETERMINE CPEST AT (WT2 + WT3)/2.0
FINISHED WHEN WT2 HAS NOT CHANGED MORE THEN 0.5 DEGREES

CPEST = FLUID(1, 1, T3, "VAP MT EX AT CPEST1 ")
TOLD = QISO * (H5 - H4) / CPEST / Q * WT3

CONTINUE

T = AVGTEMP(TOLD, T3)
CPEST = FLUID(1, 1, T, "VAP HT EX AT CPEST2 ")
TNEW = QISO * (H5 - H4) / CPEST / Q * WT3

IF(ABS(TOLD-TNEW) <LT. 0.3) GO TO 300
TOLD = TNEW
GO TO 200

CONTINUE

WT2 = TNEW
WT23 = AVGTEMP(WT2, WT3)
T45 = AVGTEMP(T4, T5)

PROPERTIES OF WATER AT WT23

WCOND = FLUID(2, 1, WT23, "VAP MT EX AT WCOND ")
WHRO = FLUID(3, 1, WT23, "VAP MT EX AT WHRO ")
WHRO = WHRO + (0.007564*WPDS)*1.0
IF(WPDS<EQ(0.1)) WVISC = FLUID(4,1,WT23,"VAP MT EX AT WVISC ")
IF(WPDS<GT(0.0)) WVISC = VS(WPDS*WT23)
WCPL = FLUID(1, 1, WT 3, "VAP MT EX AT WCPL ")
WRPANO = WCPL * WVISC / WCOND
WREYN0 = REYN0(WRHO, WVELV, DIAV, WVISC)

INSIDE HEAT TRANSFER COEFFICIENT

HTCI = HTCUEF(WCOND, DIAV, WRPANO, WREYN0, 0.3)

NUMBER OF TUBES IN THE VAPORIZER

TUBEN = TUBES(QUA, WHRO, WVELV, DIAL)

HEAT TRANSFER IN VAPORIZER
SUBROUTINE SUBHEX

C
MTVAP = 0.50 = (H5-H4)

C
THICKNESS OF WALL
WALLTH = (UIAOV-DIAV) / 2.0

C
PROPERTIES OF LIQUID AND VAPOR WORKING FLUID AT T45

C
LIQUID PROPERTIES
PCPL = FLUID(1,NFLUID,145, "VAP HT EX AT PCPL ")
PCOND = FLUID(2,NFLUID,145, "VAP HT EX AT PCOND ")
PRHOL = FLUID(3, NFLUID, 145, "VAP HT EX AT PRHOL ")
PVISL = FLUID(4, NFLUID, 145, "VAP HT EX AT PVISL ")
IF(PVISL <LT. 0.0) PVISL = U*1

C
VAPOR PROPERTIES
PRMOV = FLUID(7, NFLUID, 145; "VAP HT EX AT PRMOV ")
PVISV = FLUID(8, NFLUID, 145, "VAP HT EX AT PVISV ")
HVAP = FLUID(9, NFLUID, 145, "VAP HT EX AT HVAP ")
PCRIT = FLUID(10, NFLUID, 145, "VAP HT EX AT PCRIT ")
SIGMA = FLUID(12, NFLUID, 145, "VAP HT EX AT SIGMA ")

C
DELTP = PCM1T - P4

C
TOTAL FLOW AREA
CALL QAREA(DIAOL, TUBEN, CGDIA* FLAREA)

C
PVEL = VELOCITY(QISO, FLAREA, PRHOL)
REYNOL = REYNOL(PRHOL*PVEL*EGDIA, PVISL)
PRANOL = PCPL * PVISL / PCOND

C
BOILING HEAT TRANSFER = ASUMIE MACROCONVETIVE HEAT TRANSFER ONLY

C
MARTINELI PARAMETER

C
MARTPA = (PRHOL/PRMOV)**5 * (PVISV/PVISL)**1

C
MULTIPLICATION FACTOR
F = 1.7698 + 1.17931**MARTPA = 0.0074865*MARTPA**2 + 2.85855E-5 * MARTPA**3

C
MACROCONVETIVE HEAT TRANSFER COEFFICIENT
FHTCOEF = 0.023*REYNOL**0.6*PRANOL**0.4*PCOND/EGDIA

C
OVERALL HEAT TRANSFER COEFFICIENT
U = 0.7HCOE(DIAOV, MTC1, DIAV, KALUM, FHTCOEF, FOU/LI, FOU/L0)

C
WT = W72 - T5
FT = W73 - T4

C
AVERAGE HEAT TRANSFER AREA
SUBROUTINE SUBHEX

C MTA = WTAMEA(QISO, HS, H4, U, WT, FT)
C MTAEX(2) = MTA
C
C HEAT TRANSFER LENGTH
C
C MTL = WTLMGH(MTA, DIAOV, LUBEN)
C
45N CONTINUE
C
HXSRSLT(1,2) = DIAV
HXSRSLT(2,2) = DIAOV
HXSRSLT(3,2) = TUREN
HXSRSLT(4,2) = EQDIA
HXSRSLT(5,2) = HTCi
HXSRSLT(6,2) = FTHTCO
HXSRSLT(7,2) = U
HXSRSLT(8,2) = MTA
HXSRSLT(9,2) = MTL
C
***************SUPER HEATER**********************
C
T15 = AVGTEMP(T1, T5)
WT12 = AVGTEMP(WT1, WT2)
C
DETERMINE PROPERTIES OF WATER AT WT12
C
WCOND = FLUID(2, 1, WT12, "SUPER HEAT AT WCOND")
WRHO = FLUID(3, 1, WT12, "SUPER HEAT AT WRHO")
WRHO = WRHO * (WPRD + 1) / WPRD
IF(WPRD .EQ. 0) WVISC = FLUID(4,1,WT12,"SUPER HEAT AT WVISC")
IF(WPRD .LT. 0.) WVISC = VS(WPRD, WT12)
WCPL = FLUID(1, 1, WT12, "SUPER HEAT AT WCPL")
PRANO = WCPL / WVISC / WCOND
WREYNO = REYNO(WRHO, WVELS, DIA, WVISC)
C
INSIDE HEAT TRANSFER COEFFICIENT
C
HTCIC = HTCICF(WCOND, DIA, PRANO, WREYNO, 0, 3)
C
NUMBER OF TUBES IN SUPERHEATER
C
TUBEN = TUBES(GNA, WRHO, WVELS, DIA)
C
WORKING FLUID PROPERTIES (VAPORS) AT T15
C
PCPVW = FLUID(5, NFLUID, 1, 15, "SUPER HEAT AT PCPV")
PCONDV = FLUID(6, NFLUID, 1, 15, "SUPER HEAT AT PCONDV")
IF(NFLUID .GT. 2) GO TO 500
NITA = 0
DR = 1.0 / (8.0 * (T15 - 459.6) / P1)
4AN CONTINUE
NITA = NITA + 1
PESTV = PESTV(T15, 459.67, WK, 0, 0)
IF(ABS(PESTV - P1) .LT. 0.5) GO TO 490
IF(NITA .LT. 40) STOP "SUBHEX--DB FAILED TO CONVERGE"
SUBROUTINE SUBHEX

DB = Da*(1.0 + (P1-PESTV)/P1)
GO TO 430
490 PRHOV = DB * BULT
GO TO 520
500 CONTINUE
VH = Ra*(T15 + 459.67)/P1
MAXR = U
510 CONTINUE
MAXR = MAXR + 1
PESTV = EDSTAP(T15+459.67+VH*0.0)
IF (ABS(PESTV-P1) <LT. 0.5) GO TO 515
IF (MAXR <LT. 10) STOP "SUBHEX FAILED TO CONVERGE"
VH = VP*(1.0 + (PESTV-P1)/P1)
GO TO 510
515 CONTINUE
PRHOV = 1.0/V8
520 CONTINUE
PVISV = FLUID(A, TFLUID, 115, "SUPER HEAT AT PVISV")
PVEL = VELOCY(QISO, FLAREA, PRHOV)
REYN = REYN(PRHOV, PVEL, EQDIA, PVISV)
PRANOV = PCPV = PVISV / PCONDV

C C C
C OUTSIDE HEAT TRANSFER COEFFICIENT
C
HTCO = WTCOF(PCONDV, EQDIA, PRANOV, REYN, 0.4)

C C C
C OVERALL HEAT TRANSFER COEFFICIENT
C
U = OHTOE(DIAOS, HTCI, DIAS, KALUM, HTC, FOULF, FOULFO)

C C
C HEAT TRANSFER AREA
WT = WT2 - TS
FT = WT1 - T1

C C
C HTA = WT AREA(QISO, H1, HS, U, WT, FT)
HTAEX(1) = HTA

C C
C HEAT TRANSFER LENGTH
C
HTL = HTLVH(HTA, DIAOS, TUREN)

C C
C HXRSLT(1+1)=DIAS
C HXRSLT(2+1)=DIAS
C HXRSLT(3+1)=TUBEN
C HXRSLT(4+1)=EQDIA
C HXRSLT(5+1)=HTCI
C HXRSLT(6+1)=HTCO
C HXRSLT(7+1)=U
C HXRSLT(8+1)=HTA
C HXRSLT(9+1)=HTL
RETURN
END
SUBROUTINE SUPHEX

SUBROUTINE SUPHEX(WT1,W4,QISO,T1,T3,H3,WVELSC,DIASC,DIASCON,
Foulf1,Foulf0,Fluidf,Fl1,HTAEX,QNA,WPDS,PINSUP)
COMM /FLAGS/ FLAG, NEXT, FLAG1
INTEGER FLAG, FLAG1
COMM /HXLUT/ HXRLT(9,3)
COMM /FXCON/AL,BC,CL,DL,EL,A,BFR,C,DFR,E,F,SR,A2,B2,C2,A3,B3,
* C3,A4,R4,A5,B5,C5,A6,B6,K,Y,ALPHA,MC,A,HCB,HCC,HCD,HCF,XFR,YFR,PC,
* TE+VC
COMM /BUTANE/AR,ZERO,AR1,AR2,AR3,AR4,MR,ZERO,RR1,RR2,RR3,RR4,
* ZERO*ZERO*ZERO*SM,SMD,SMC,SM0,APLAR,AMMA,PHI,BUTW4,RR
DIMENSION HTC(25),TUBEN(25),HTC0(2),0K(25),HTA(25),HTL(25)
DIMENSION HTAEX(3)

DETERMINES THE CHARACTERISTICS OF THE SUPER-CRITICAL CYCLE HEAT EXCHANGER

INPUT PARAMETERS

WT1- GEOTHERMAL FLUID FLOW RATE
WT4- WATER TEMPERATURE AT HEAT EXCHANGER INLET
WT4- WATER TEMPERATURE AT HEAT EXCHANGER OUTLET
QISO- WORKING FLUID FLOW RATE
T1- WORKING FLUID TEMPERATURE AT INLET TO TURBINE
T3- WORKING FLUID TEMPERATURE AT HEAT EXCHANGER INLET
P1- WORKING FLUID PRESSURE AT TURBINE INLET
H1- WORKING FLUID ENTHALPY AT TURBINE INLET
H3- WORKING FLUID ENTHALPY AT HEAT EXCHANGER INLET
WVELSC-WATER VELOCITY IN HEAT EXCHANGER
DIASC- INSIDE TUBE DIAMETER IN HEAT EXCHANGER
DIASCON-OUSTIDE TUBE DIAMETER IN HEAT EXCHANGER
F0U,FI-INSIDE FOULING FACTOR
F0U,FO-OUTSIDE FOULING FACTOR
Fluidf-WORKING FLUID INDEX

REAL KALUM

CONSTANTS

THERMAL CONDUCTIVITY OF CARRON STEEL
DATA KALUM/(30,
VS(WPDS) = (-30361 + 214.7277 T - 2552.82/T**2)*(1.0*276*WPDS)

ITER = 1
IF(ITER.EQ.1) GO TO 200
100 WT4 = WT4 + 0.0
GO TO 120
110 WT4 = WT4 + 5.0
120 ITER = ITER + 1
IF(ITER.EQ.1) STOP"SUPHEX = SUPERCRITICAL HEAT EXCHANGER FAILED TO CONVERGE"
T = AVGWTMP(WT1,WT4)
CPES1 = FLUID(1")/T,"PECALC WT4")
G = QISO*(H1-H3)/CPES1/(WT1-WT4)
QNA=G/(1.0*WPDS/100)
200 HTAEX(1) = 0

DETERMINE DELT FOR TWENTY FIVE SERIES HEAT EXCHANGERS
SUBROUTINE SUPHEX

C
DELT = (T1 - T3) / 25 * 0
FTMIN = 10 * 0
C
DO 100 N = 1, 25
T3 = N + DELT * (I - 1)
T1 = T3 + DELT
C
IF(NFLUID .GT. 3) GO TO 260
C
CALCULATE ENTHALPIES FROM MUTHANE
IF(I .EQ. 1) HM3 = H3
IF(I .GT. 1) HM3 = HM1
IF(I .EQ. 29) GO TO 400
C
NIT = 0
IF(TT1 .GT. 10 * FLUID(I) * FLUID(TT1 + "CR mobile") ) GO TO 215
DO1 = FLUID(3 + FLUIDTT1 * "SUPHEX AT DD1 CALC ") / FLUID(13 + FLUID * TTT1 * "SUPHEX AT DD1 CALC ")
GO TO 240
215 CONTINUE
DO1 = 1.0 / (RT + (TT1 + 459.67) / P1)
220 CONTINUE
NIT = NIT + 1
PESTV = EOSTP(TT1 + 459.67, DD1 * 0 * 0)
IF(ABS(PESTV - P1) * LT. 0.05) GO TO 240
IF(NIT1 .GT. 40) STOP "SUPHEX -- DD1 FAILED TO CONVERGE"
DO1 = NDI * (1.0 + (P1 - PESTV) / P1)
GO TO 220
240 CONTINUE
HM1 = EOSTP(TT1 + 459.67, DD1) + HIDEAL(TT1 + 459.67)
GO TO 400
C
C
260 CONTINUE
C
CALCULATE ENTHALPIES FROM FREON
IF(I .EQ. 1) HM3 = R3
IF(I .GT. 1) HM3 = HM1
C
MAX1 = 0
VV1 = DO1 = (TT1 + 459.67) / P1
280 CONTINUE
MAX1 = MAX1 + 1
PESTV = EOSTP(TT1 + 459.67, VV1 * 0 * 0)
IF(ABS(PESTV - P1) * LT. 0.05) GO TO 300
IF(MAX1 .GT. 40) STOP "SUPHEX -- VV1 FAILED TO CONVERGE"
VV1 = VV1 * (1.0 + (PESTV - P1) / P1)
GO TO 280
300 CONTINUE
HM1 = HVAC(TT1 + 459.67, VV1 + P1)
C
400 CONTINUE
IF(I .EQ. 25) HM1 = H1
C
WATER TEMPERATURE CALCULATION
C
IF(I .EQ. 1) WTT4 = WT4
IF(I .GT. 1) WTT4 = WTT1
SUBROUTINE SUPHEX

IF(1.E0.25) GO TO 500

C

WT1 = 0130*(MH+H3)/Q/FLUID(1+1*WT14,"SUPHEX AT WT1 CALC")

500 IF(1.E0.25) WT1 = WT1

WT14 = AVGTEMP(WT11,WT14)

T13 = AVGTEMP(TT1,T13)

C

PROPERTIES OF WATER AT WT14 DEGREES

WCOND = FLUID(2,1,WT14,"SUPHEX AT WCOND CALC")

WRHO = FLUID(3,1,WT14,"SUPHEX AT WRH CALC")

IF(WPOSE.EQ.0) WVIS = FLUID(4+1,WT14,"SUPHEX AT WVIS CALC")

IF(WPOSE.EQ.0) WVIS = VS(WPOS+WT14)

MCPL = FLUID(1+1,WT14,"SUPHEX AT MCPL CALC")

PRAN0 = WPL/WVIS/WCOND

WREYNO = REYNO(WRHO,WVELS+DIASC+WVIS)

C

INSIDE HEAT TRANSFER COEFFICIENT

HTCI(I) = HTCOEF(WCOND+DIASC+PRAN0+WREYNO,0.0+3)

NUMBER OF TUBES IN HEAT EXCHANGER

TUBEN(I)=TUBES(QNA,WRHO,WVELS+DIASC)

C

TOTAL FLOW AREA

CALL QAREA(DIAOSC,TUBEN(I),EQDIA,FLAREA)

C

VAPOR WORKING FLUID PROPERTIES AT T13 DEGREES

PCPV = FLUID(5,NFLUID,113,"SUPHEX AT PCPV CALC")

PCONOV = FLUID(6,NFLUID,113,"SUPHEX AT PCONOV")

PRHOV = Q10*FLUID(13,NFLUID,113,"SUPHEX AT PRHOV CALC")

PVISV = FLUID(13,NFLUID,113,"SUPHEX AT PVISV CALC")

PVEL = VELOC(Q10,FLAREA,PRHOV)

WREYNO = REYNO(PRHOV,PVEL,EQDIA,PVISV)

PRANOV = PCPV/PVISV/PCONOV

C

OUTSIDE HEAT TRANSFER COEFFICIENT

HTCO(I) = HTCOEF(PCONOV,EQDIA,PRANOV,WREYNO,0.0+4)

C

OVERALL HEAT TRANSFER COEFFICIENT

U(I) = OHTCOE(DIAOSC,HTCI(I)+DIASC,KAUH,HTCO(I),FoulF,0)

C

HEAT TRANSFER AREA

WT = WT14-T13

FT = WT11-T11

IF(FTMN>U+FT) FTMIN = F

C

CHECK IF SUPERCRITICAL PINCH POINT IS LESS THAN MINIMUM

ALLOWABLE PINCH POINT DEFINED IN THE INPUT DATA
SUBROUTINE SUPHEX

C CHECK IF SUPERCritical PINCH POINT is GREATER THAN maximum
C ALLOWABLE PINCH Point
IF(FTMIN.GT.PINSUR) GO TO 100
IF(FT.GT.0.1T) GO TO 55
HTA(I) = HTAREA(QISO+HM1+MH3+U(I)*WT,WT)
GO TO 100
55 HTA(I) = HTAREA(QISO+HM1+MH3+U(I)*FT,WT)
600 HTAEX(I) = HTAEX(I) + HTA(I)

C HEAT TRANSFER LENGTH
HTL(I) = HTLNLH(MTA(I), TIAOSC, TUBEN(I))
IF(FLAG.EQ.2 .AND. [EQ.1]) WRITE(6,2090) ITER
2090 FORMAT(1H1,"ITERATION NO."I4,"FOR SUPERCritical HEAT EXCHANGER")
IF(FLAG.EQ.2)
WRITE(6,3000) I,ITER,WTT4,T3,WTT1,FTMIN(I),MTA(I),HTC(T)
3000 FORMAT(1H1,"ITER="I5,"TWT4="1E14.6,"T3="1E14.6,"MTA="1E14.6,"HTC="1E14.6,"U(I)="1E14.6,"MTL="1E14.6,"HTL="1E14.6)

C CONTINUE
1000 IF(I.EQ.25 .AND. FTMIN.GT.2R4) GO TO 110
CONTINUE

HTC(T) = 0.
TUBET = 0.
HTC(T) = 0.
UT = 0.
MTAT = 0.
MTLT = 0.
DO 2000 I = 1,25
HTC(T) = HTPC(1) + HTPC(I)
TUBET = TUBET(I) + TUBEN(I)
HTC(T) = HTPC(T) + HTC(T)
UT = UT + U(I)
HTA(I) = HTA(I) + HTA(I)
MTLT = MTL(T) + HTPC(T)
2000 CONTINUE

HTC(T) = HTPC(T)/25.0
TUBET = TUBET/25.0
HTC(T) = HTC(T)/5.0
UT = UT/25.0
MXRSLT(1+1) = DIAOSC
MXRSLT(2+1) = DIAOSC
MXRSLT(3+1) = TUBET
MXRSLT(4+1) = EQDIA
MXRSLT(5+1) = HTC(T)
MXRSLT(6+1) = HTC(T)
MXRSLT(7+1) = UT
MXRSLT(8+1) = HT(T)
MXRSLT(9+1) = HTLT

C RETURN
END
FUNCTION SVAP(T, V, S)
COMMON /BINARY/ SUPER, NFLUID, WT1, WT2, WT3, WT4,
       ESTWT4, WP1, WP3, CWT1, CWT2, CWP1, CWP2, T1,
       T2, T2SATL, T3, T4, T5, T6, P1, P2,
       P3, PA, P4, H1, H2, MEST, M2SATL, H3,
       M4, H5, S1, SEFST, S3, SL, SVSATL, D1,
       D2, D2EST, V1, VEEST, PINCHP, WVELL, DIAL, DIAOL,
       WVELV, DIAV, DIAOV, WVELS, DIAS, DIAOS, WVELSC, DIASC,
       DIAOSC, PYELC, DIAC, DIAOC, FOULFI, FOULFO, BAR, GIS0,
       HTAEX(3), HTACON, TPRCH, WTFOUR, APPCON, APPCT, APPH X,
       DELPMX, PINSUB, PINSUP, PUNEP, RANGE
LOGICAL /SUPER /
REAL JAY, REAL KAY

C** COMMON FOR THE FREON CONSTANTS
   *C3, A4, B4, C4, A5, B5, C5, A6, B6, C6, KAY, ALPHA, HCA, HCB, HCC, HCD, HCF, XFR
   YFR(4, 1, V, VC)
   JAY = V(1, 85, 53)
   ONE = V(4, 6)
   TWO = 2, 0, ONE = 2
   THREE = 3, 0, ONE = 3
   FOUR = 4, 0, ONE = 4
   EXPAV = 0, 0
   PART = 0, 0
   AV = ALPHA + V
   IF(AV GT 230.) GO TO 220

C*** FOR R12, R21, R113 AND R114, B6 AND ALPHA = 0.0
IF(B6 EQ 0.0) GO TO 220
EXPAV = EXP(AV)
PART = 1, U + EXPAV * (B6/ALPHA)

220 CONTINUE
PART1 = HCA - ALOB(T) - HCB* T + (HCC* T**2)/2. + (HDD* T**3)/3. - HCF
       /(2*T**2) + JAY*ALO1(ONE)
PART5 = B2/ONE + R3/TWO + B5/THREE + B5/FOUR
PART6 = JAY*KAY*EXP(-KAY* T/1C)/TC
PART7 = C2/ONE + C3/TWO + C3/FOUR
IF(NFLUID EQ 1) PART7 = PART7 + C4/THREE
SVAP = PART1 + JAY*(PART5 + PAR14) + PART6*PART7 + YFR*S
RETURN
END
SUBROUTINE TOWER

SUBROUTINE TOWER(CTA, QWI, QWI, TOWI, WHAT, DBAT, RI, TCI, + APCT)
COMMON /FLAGS/ FLAG, NXT, FLAG1
INTEGER FLAG, FLAG1

THIS ROUTINE RETURNS TO THE CALLING PROGRAM THE FOLLOWING
1. CTA, COOLING TOWER AIR FLOW RATE
2. QLA, EVAPORATION RATE

INPUT PARAMETERS

QWI- COOLING WATER FLOW RATE
TOWI- TEMPERATURE OF QWI
WHAT- WET BULB AIR TEMPERATURE
DBAT- DRY BULB AIR TEMPERATURE
RI- BAROMETRIC PRESSURE (IN. HG)
RT- HUMIDITY RATIO FROM PSYCHROMETRIC CHART (WHAT AND DBAT)
TCI- COOLING WATER EXIT TEMPERATURE

REAL MA

CONSTANTS
CPA- SPECIFIC HEAT OF AIR
DATA CPA / 0.2404 /

C ENTH- FUNCTION TO DETERMINE ENTHALPY OF MIXTURE CORRESPONDING
TO SPECIFIC HUMIDITY (W) AND TEMPERATURE (T)
ENTH(T,W) = CPA * T * W / (1062.0 + 0.44 * T)

C HUMID- FUNCTION TO DETERMINE SPECIFIC HUMIDITY CORRESPONDING
TO ACTUAL VAPOR PRESSURE (AVP)
HUMID(AVP) = AVP / (1.61 * (R - AVP))

C PDI- AIR INLET SATURATED VAPOR PRESSURE AT DBAT
PDI = PSL(DBAT)*2.04177

C WI- SPECIFIC HUMIDITY AT INLET
WI = HUMID(PDI)

C HMI- ENTHALPY OF MIXTURE AT INLET
HMI = ENTH(DBAT, WI)

C TLA- TEMPERATURE OF SATURATED AIR AT EXIT OF TOWER
TLA = DBAT * 29.0
PVO = PSL(TLA)*2.04177
W0 = HUMID(PVO)
HMO = ENTH(TLA, W0)

MA- LBS OF AIR ENTERING TOWER PER LB OF COOLING WATER ENTERING
X- LBS OF WATER EVAPORATED PER LB OF COOLING WATER ENTERING

SOLVE SIMULTANEOUSLY

X = MA * (W0 - WI)

(TOWI - 32.0) + MA * HMI = (1.0 - X) * (TCI - 32.0) * MA * HMO

MA = (TPI - TOWI) / ((HMI - HMO) * (TCI - 32.0) * (W0 - WI))
SUBROUTINE TOWER

C
C RMOAI = (H-PRVI)/(DRAT*59*1.3223)
C
C CTA= COOLING TOWER AIR FLOW RATE
C CTA= QWI * MA / (60.0 * RHOAI)
C
C QLA= EVAPORATION RATE
C QLA= QWI * X

C
C HL= HEAT LOAD (BTU/HR)
C CP1= SPECIFIC HEAT OF WATER AT TEMPERATURE 1
C CP2= SPECIFIC HEAT OF WATER AT TEMPERATURE 2
C WTAV1= AVERAGE OF EXIT COOLING WATER TEMPERATURE AND INLET COOLING
C WATER TEMPERATURE
C WTAV2= AVERAGE OF EXIT COOLING WATER TEMPERATURE AND TEMPERATURE
C OF SATURATED AIR AT EXIT OF TOWER
C
C CALCULATION OF CPI AND CP2
C WTAV1= (TQWI + TCI) / 2.0
C WTAV2 = (TIA + TCI) / 2.0
C
C T = WTAV1 / 100.0
C CP1= 9.916E-1 * 2.167E-1 * T - 1.5327E-2 * T**2 + 3.7245E-3
C
C T = WTAV2 / 100.0
C CP2= 9.916E-1 * 2.167E-1 * T - 1.5327E-2 * T**2 + 3.7245E-3
C
C IF(TCI.WEIAT.LT.8.0) STOP "TOWER -- COOLING TOWER APPROACH IS INSUF
CICIENT"
C
C CALCULATION OF HEAT LOAD (BTU/HR)
C HL = QWI * CP1 * (QWI - TCI) - QLA * CP2 * (TIA - TCI)
C
C IF(FLAG.NE.0) WRITE(6,1) APPCT, CTA, QLA, WBAT, DRAT, R, RI, HL
1 FORMAT(""SUBROUTINE TOWER -- COOLING TOWER")
C
C RETURN
END
FUNCTION TFOUR(P, RHOL, RN0, NFLucD)

C***FUNCTION TO DETERMINE T4 (SATURATION TEMPERATURE)
COMMONTFLAGS/ FLG  + N1NT  + FLG1
INTEGEO FLG  + FLG1
COMMONTBUTANE/ AD, AZERO, AB, AB, AB, AB, 0, RZERO, BB, 0, AB, AB,
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FUNCTION TFOUR

IF(FLAG .EQ. 2) PRINT 5, P, T, RHOL, RHOV, FL, FV, FRATIO
5 FORMAT("DETERMINATION OF SATURATED DENSITIES AT TURBINE INLET PRESSURE"
* "FINAL VALUES",
* "P = TURBINE INLET PRESSURE (PSIA)"
* T = TEMPERATURE OF SATURATED WORKING FLUID (DEG. R)
* RHOL = SATURATED LIQUID DENSITY (LB MOLE/STANDARD CU. FT)
* RHOV = SATURATED VAPOR DENSITY (LB MOLE/STANDARD CU. FT)
* FL = LIQUID FUGACITY
* FV = VAPOR FUGACITY
* FRATIO = RATIO OF VAPOR TO LIQUID FUGACITY)
TFOUR = 459.67
RETURN
END
FUNCTION TUBES

FUNCTION TUBES(Q, WRHO, WVEL, DIA)

FUNCTION DETERMINE NUMBER OF TUBES IN THE HEAT EXCHANGER

Q = WATER FLOW RATE
WRHO = DENSITY
WVEL = VELOCITY
DIA = DIAMETER

TUBES = 4.0 * Q / WRHO / WVEL / 3.1416 / DIA**2
X = AINT(TUBES)
IF (AMON(TUBES+X) .NE. 0.0) TUBES = X + 1.0

RETURN
END
FUNCTION VELOCITY

FUNCTION VELOCITY(QISO, FLAREA, PRMOL)

C Computes velocity
C QISO - fluid flow rate
C FLAREA - flow area
C PRMOL - fluid density
C
C VELOCITY = QISO / FLAREA / PRMOL
C
RETURN
END
FUNCTION VPRESA(T)
COMMON /BINARY/ SUPER, NPFLUID, WT1, WT2, WT3, WT4,
*ESTWT4, WP3, CMT1, CMT2, CwP1, CwP2, P1, P2,
*T3, T2EST, T2SATL, T4, T5, T6, T2, T5,
*P3, P4, PB, PH, H2, H2EST, M2ATL, M3,
*H4, S1, S2EST, S3, S4, SVSATL, D4,
*D2, D2EST, V1, V2EST, PINCHP, WVELL, DIAL, DIAOL,
*WVELV, DIAV, DIAOVC, WVELS, DIA, DIAOS, WVELSC, DIAASC,
*DIAOVS, PVFLC, DIA, DIAOCC, FOULFI, FOULFO, BAR, QISO,
*HTAEX(3), HTACON, T*PPCH, WFTOUR, APPCON, APPCT, APPH4,
*DLPDH, PINSUB, PINSUP, PINCM, RANGE
LOGICAL SUPER
REAL KAY

COMMON FOR THE FREON CONSTANTS
**C3, A4 + B4, C4, A5 + B5 + C5, A6 + B6 + C6, KAY, ALPHA, HCA, MCB, HCC, MCD, MCF, XFR
**YFR, RAF + TC + VC

IF(NFLID NE 10) AA = E * T**C
IF(NFLID NE 10) GO TO 200
IF(E * FQ * N 0) GO TO 200
AA = ((F - T) / T) * ALOG10(F - 1)

200 CONTINUE
VPRESA = A + B F R / T + C * ALOG10(T) * D F R * T + AA
RETURN
END
FUNCTION VPRESB

COMMON /BINARY/ SUPER, NrLUID, WT1, WT2, WT3, WT4, .ESTWT4, .Ww1, WP3, TcEST, T2SATL, T3, T4, T5, P1, P2, P3, P4, P5, H1, H2, M2, M2EST, M2SATL, M3, .ESTWT4, .Ww1, WP3, TcEST, T2SATL, T3, T4, T5, P1, P2, P3, P4, P5, H1, H2, M2, M2EST, M2SATL, M3, .ESTWT4, .Ww1, WP3, TcEST, T2SATL, T3, T4, T5, P1, P2, P3, P4, P5, H1, H2, M2, M2EST, M2SATL, M3

COMMON /FRECON/ ALRL, CLmLELtAt8FRoERPtSEItA2, 82C2A3, 83ocoE3tA4.54, C4tAstEiboCS, A6ochtC60KATtALPHA, HCAtHCRIHCCoHCDoMCF0

C*** COMMON FOR THE FREON CONSTANTS
COMMON /FRECON/ ALRL, CLmLELtAt8FRoERPtSEItA2, 82C2A3, 83ocoE3tA4.54, C4tAstEiboCS, A6ochtC60KATtALPHA, HCAtHCRIHCCoHCDoMCF0

C*** FOR R12* P21 AND R113 F AND E ARE = 0.0
IF (F *EQ. 0.0) GO TO 220
PART1 = F *ALO810 (F*T)
220 IF (NFLH/n.E0,10) 80 TO 30u
VPPES8sP*((-ALOG(In.0)* (SrP.PART1))/T**e*(CmE)/T*DFR*ALOG(1°.0))
GO TO 999
300
VPPES8sP=P* ALOG(10.* )* (BFR*PART1)/T**2*(C-E)/T*DFR*ALOG(10.* )
GO TO 999
999 RETURN
END
SUBROUTINE WFFRT

SUBROUTINE WFFRT(QISO, H2, SUPER, GRKW, T1, H1, H2EST, H2SATL, 
* TCRIT)
COMMON/WFFRT/IREFF, REFF, OEFF

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INPUT VALUES

SUPER- LOGICAL VARIABLE = TRUE IF SUPER-CRITICAL CYCLE
GRKW- GROSS KILOWATT OUTPUT
WT1- WATER TEMPERATURE AT INLET OF SUPER HEATER
P1- WORKING FLUID PRESSURE AT TURBINE INLET
T1- WORKING FLUID TEMPERATURE AT TURBINE INLET
H1- WORKING FLUID ENTHALPY AT TURBINE INLET
P2- WORKING FLUID PRESSURE AT TURBINE EXHAUST
T2EST- WORKING FLUID TEMPERATURE AT TURBINE EXHAUST (ESTIMATED)
H2EST- WORKING FLUID ENTHALPY AT TURBINE EXHAUST (ESTIMATED)
S2EST- WORKING FLUID ENTROPY AT TURBINE EXHAUST (ESTIMATED)
H2SATL-ENTHALPY OF WORKING FLUID AS A SATURATED LIQUID AT P2
TCRT- CRITICAL TEMPERATURE OF WORKING FLUID

REAL IREF, IEFF
LOGICAL SUPER

ESTIMATED INTERNAL EFFICIENCY OF THE TURBINE
DATA IEFF / 0.9 /

IF(.NOT. SUPER) GO TO 300
SUPER-CRITICAL CYCLE
IF(T1 .LT. TCRIT) STOP "T1 .LT. TCRIT"
300 CONTINUE

QIDEAL= IDEAL WORKING FLUID FLOW RATE

QIDEAL=3414.43 / (H1-H2EST) = GRKW

IEFF= IDEAL RANKINE CYCLE EFFICIENCY

IEFF= (H1-H2EST) / (H1-H2SATL)

MECHANICAL LOSSES

PCMCL0 = 4.0 / SORT(GRKW/1000.0)
IF(GRKW .LT. 25000.0) PCMCL0=PCMCL0 * 1.05

BREAK EFFICIENCY

BEFF= REFF / (1.0 * (PCMCL0 / 1000.0))

OVERALL EFFICIENCY

OEFF = REFF * 0.985

WORKING FLUID FLOW RATE
SUBROUTINE WFFRT

C DICO = DIDEAL / OEFF
C HK = M1-3414.43 * GRKW / QISO
C REFF = (M1-M2) / (M1-M2SATL)
C RETURN
END