Presented at the Second United Nations
Symposium on the Development and
Use of Geothermal Resources,
San Francisco, May 19 - 29, 1975

LBL-3238
CONF-750525--11

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May 1975

Prepared for the U. S. Energy Research and
Development Administration under Contract W-7405-ENG-48

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LBL-3238

CALCULATION OF GEOTHERMAL POWER PLANT CYCLES USING PROGRAM GEOTHM*

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ABSTRACT

Program GEOTHM, a thermodynamic process program under development at the Lawrence Berkeley Laboratory, can be used to calculate the thermodynamic characteristics of geothermal power plant cycles. The computer program has been used to model some of the major geothermal power plant cycles. These include: (1) flashed steam cycles, (2) two-phase expander cycles, and (3) bi-fluid or binary cycles. The program will design the power plant thermodynamically (for example, it calculates pump and turbine ratings and heat exchanger area), and it can be used to model off-design operation of the plant. The paper includes three examples of program GEOTHM's capabilities: (1) A study of the electrical energy yield per unit well flow in bi-fluid cycles using ammonia, R-22, isobutane, and R-113 as a function of turbine inlet temperature and pressure is presented; (2) The change in power plant yield in bi-fluid cycles using ammonia and isobutane as the brine heat exchanger U factor changes is shown; and (3) The effect of non-condensible gases on the optimum performance of flashed steam and two-phase expander power plants is calculated.

*Work performed under the auspices of the U. S. Energy Research and Development Administration.

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Program GEOTHM has been under development since December of 1973. This paper will report the development of the program since October of 1974. Reference 1 describes the program and its development before October 1974. This paper discusses recent developments in fluid property calculations and describes some problems which have been solved using GEOTHM. The program has been designed so that power plant cost optimization can be ultimately achieved.

Fluid Property Calculations

The routines for calculating working fluid properties have evolved continuously since the inception of program GEOTHM. During the early stages of program development, the selection of the appropriate equation of state to represent a particular class of working fluids was dictated by practical considerations.[1] Figure 1 depicts the phase diagram for a typical working fluid and the various regions where different state equations are most applicable. The Martin equation of state[2] was used to represent the refrigerants because coefficients had already been developed by Downing[3] and Milora[4]. The Starling-BWR equation[5,6] is used for the light hydrocarbons, and the ideal gas equation of state is used for near room temperature low density air gases.

The Martin equation is well-behaved in the superheated gas and supercritical fluid regime (Region 3). However, this equation is not reliable for calculating fluid properties in the dense fluid regime at reduced temperatures of 0.9 and below. Region 1 has been modeled by an incompressible fluid state equation. This equation is valid for describing the

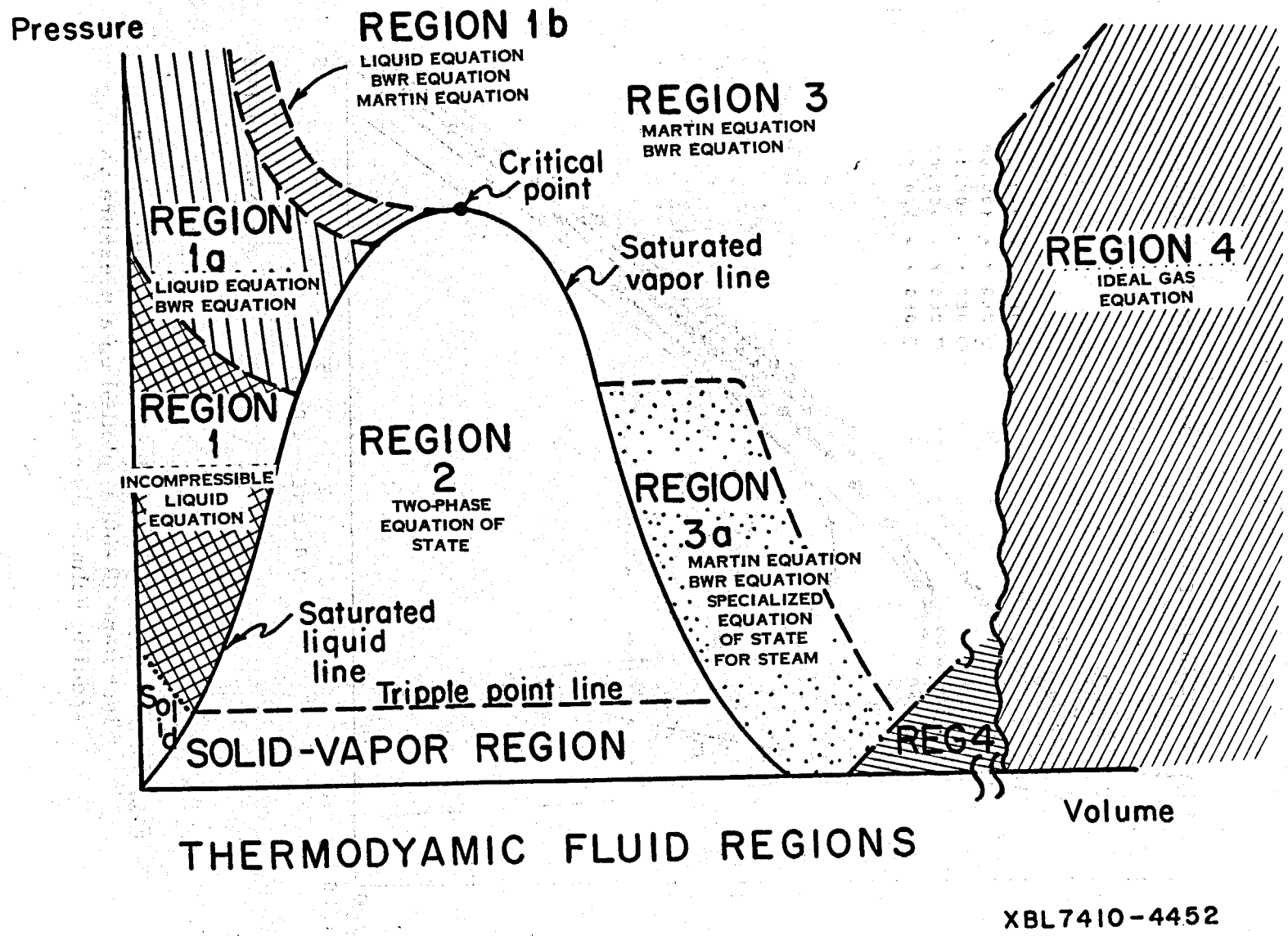
subcooled liquid regime, but it leads to inaccuracies of increasing magnitude at higher pressures in Regions 1a and 1b. The inaccuracies associated with the assumption of incompressibility lead to inconsistent fluid property calculations at the boundary between Regions 1b and 3. Any thermodynamic process passing through this boundary experiences a stepwise discontinuity in property calculations when stepping from the region governed by the incompressible fluid equation to the region described by the Martin or Starling equation in Region 3. This discontinuity, first encountered during development of the off-design heat exchanger routine, resulted in the breakdown of a convergence routine.

This line of discontinuity was eliminated by a routine which was designed to compute the weighted average of the solutions to both the incompressible fluid equation and the Region 3 equation for any state point in the transition zone 1b. Although this merge routine provides consistent fluid calculations in the Region 1b, it continues to yield inaccurate results in the dense fluid region near the critical temperature where the assumption of incompressibility is invalid.

The errors associated with the liquid equation can be reduced by extending the merge region down into the liquid regime 1a. The extension of the merge region is limited by the degree to which the Region 3 state equation is valid in the liquid regime. Tests indicated that the Martin equation was only valid down to reduced temperatures of 0.9. However, the Starling-BWR state equation[6], produces accurate thermodynamic data

for the entire liquid regime down to the triple point. Since the Starling-BWR equation successfully predicts fluid properties along the liquid-vapor dome and in all regions outside the dome, the incompressible fluid equation and the merge routine are not required for calculations involving the light hydrocarbon fluids. (The Starling equation for isobutane is plotted on a P-V diagram in Fig. 2.) Due to the computational ease and accuracy afforded by the Starling-BWR equation, it becomes highly desirable to fit all of the potential working fluids to this equation. Starling[7] believes that his equation may be applied to the representation of many other fluids and fluid mixtures including water, ammonia, the refrigerants, and the air gases.

Program GEOTHM currently employs the Starling-BWR equation, two forms of the Martin equation, and the ideal gas equation to generate thermodynamic properties for a wide variety of working fluids. The light hydrocarbon fluids including isobutane and propane are calculated by the Starling equation; ammonia and refrigerant properties are calculated using the Martin equation. Both state equations will generate a liquid-vapor dome for their respective working fluids. Once the dome is specified, a "search" routine locates the particular region of Fig. 1 where the state point to be calculated resides. This routine then directs the appropriate equation-of-state routine to solve for the thermodynamic properties at this point. For example, in the case of the Starling fluids, any fluid point lying outside the two-phase dome is computed directly by the Starling-



THERMODYAMIC FLUID REGIONS

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Fig. 1. Regions where various thermodynamic state equations are applicable.

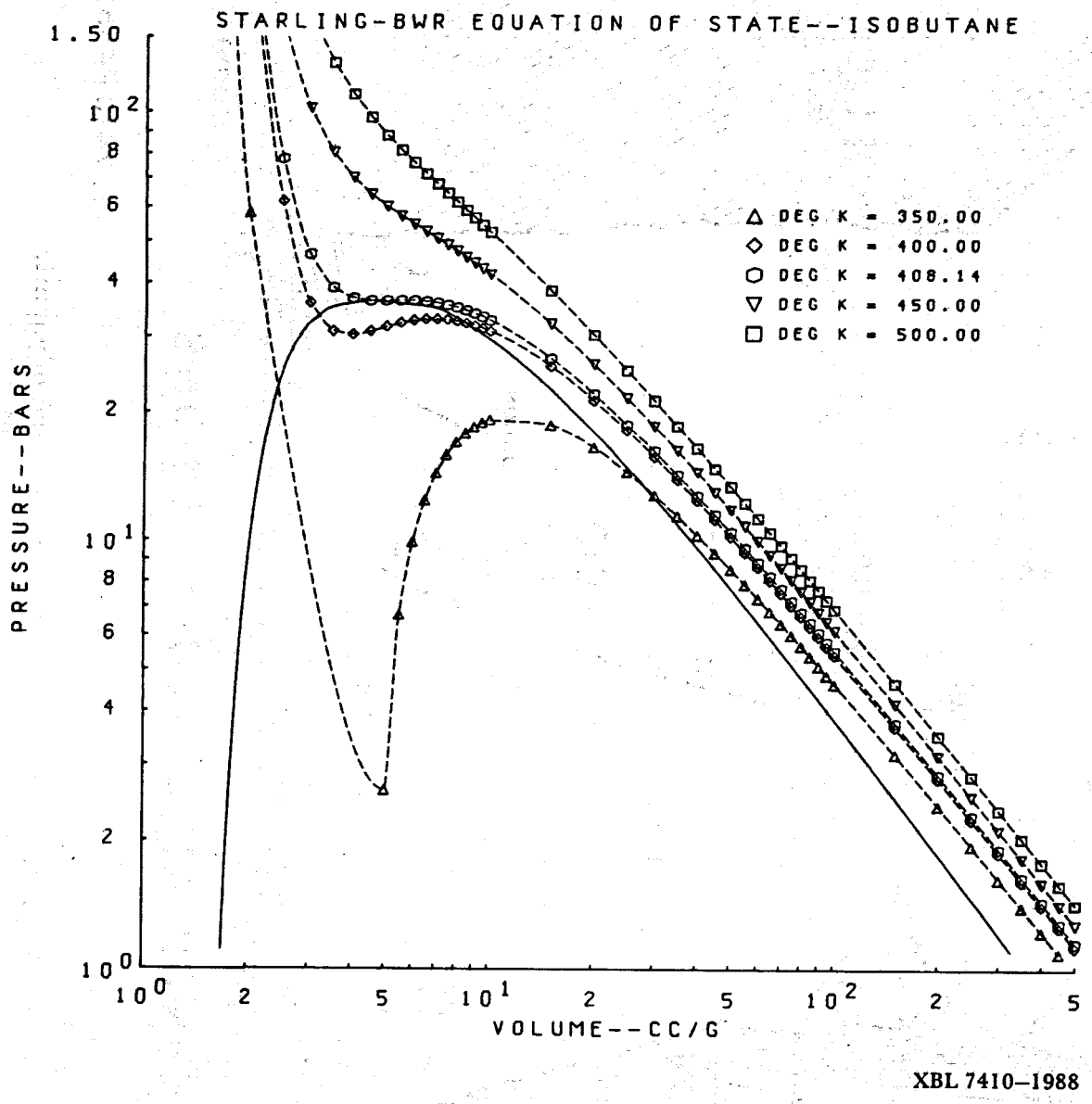


Fig. 2. Starling BWR equation of state isotherms for isobutane (R-600a) plotted on a pressure vs specific volume diagram.

BWR equation, and any point within the two-phase dome is calculated by a linear interpolation of the fluid quality.

In order to avoid the inaccuracies associated with Martin fluid calculations in the merge Region 1b, an attempt is now being made to curve fit Starling coefficients for the refrigerants, ammonia, and water using multi-property non-linear regression analysis.[8] Cycles which include fluid mixtures will also be programmed into GEOTHM using the versatile Starling equation. The procedure for curve fitting Starling coefficients to these mixtures is a nontrivial task. Ultimately, all fluid property calculations in GEOTHM, except inside the two-phase dome, will be handled by the Starling-BWR equation. GEOTHM will continue to use the ideal gas equation to represent the air gasses and CO_2 at or near room temperature.

Thermodynamic Cycle and Process Calculations

An essential feature of program GEOTHM is the ability to synthesize a ranging complexity of possible thermodynamic cycles from combinations of elemental thermodynamic processes. The LBL geothermal program has thus far utilized GEOTHM to study simple flash, bi-fluid (binary), and two-phase turbine (total flow) cycles. More complicated multi-stage flash-binary combined cycles have also been examined. An example of a combined cycle plant is detailed in the next section.

The routines which perform the various thermodynamic process calculations are numerous in the repertoire of GEOTHM. Each routine is responsible for a thermodynamic process. These processes include

isenthalpic expansion, phase separation, contact condensation, desuperheating, multiple stream mixing, and the effects of heat and friction losses in pipes. These routines are used to calculate the thermodynamics of components such as turbines, pumps, flash tanks, heat exchangers, cooling towers, and so forth. The program now runs with a dry-type cooling tower; a wet-type tower routine is currently being written.

The program models four types of counter flow heat exchangers. Two of these heat exchangers, including a regenerator, operate in the design mode. If the initial conditions specify two or three of the end-point temperatures, the mass flow in one or both streams, the average U factor, and the pinch point ΔT, the routine will compute the unknown end-point conditions, the mass flow, and the heat exchanger area which satisfies the pinch point constraint. The remaining two heat exchangers are run under off-design conditions. An example of off-design operation is presented in a later section of this paper.

The Design of Power Plants with Complicated Thermodynamic Cycles

In the previous section, we showed that program GEOTHM calculates simple thermodynamic processes and that these processes can be combined to calculate simple thermodynamic cycles. Program GEOTHM can be used on cycles which are very complicated. GEOTHM not only calculates the thermodynamics of the plant, but it also sizes the components so that the plant cost can be estimated. This section illustrates the calculation of a complicated thermodynamic power cycle.

Table 1. Cycle parameters for the power cycle shown in Fig. 3.

a. Power plant parameters

WELL FLOW RATE	164.88	KG/SEC
INLET WATER TEMPERATURE	473.16	DEG K
INLET WATER ENTHALPY	851.66	J/G
COOLING FLUID INLET TEMPERATURE	298.00	DEG K
AMBIENT TEMPERATURE	293.00	DEG K

POWER PLANT ENERGY BALANCE

AVAILABLE HEAT INPUT	126.752	MW
NET HEAT INPUT	106.219	MW
HEAT REJECTED	94.129	MW
HEAT LEAKS	1.752	MW
NET HEAT OUTPUT	95.881	MW
AVAILABLE HEAT OUTPUT	116.414	MW
TURBINE POWER	13.552	MW
PUMP POWER	1.423	MW
FAN POWER	1.792	MW
VACUUM PUMP POWER	0.000	MW
NET MECHANICAL POWER	10.337	MW
GENERATOR POWER	13.281	MW
ELECTRIC MOTOR POWER IN	3.281	MW
NET ELECTRICAL POWER	10.000	MW

POWER PLANT EFFICIENCIES AND YIELD FROM THE WATER

NET CYCLE EFFICIENCY	9.414	PCT
AVAILABLE HEAT EFFICIENCY	7.889	PCT
CARNOT EFFICIENCY	38.076	PCT
YIELD PER UNIT WELL FLOW	16.847	KWH/TONN

b. Heat exchanger parameters

HEAT EXCHANGER NUMBER	1	2	3
INITIAL MINIMUM DELTA T(K)	5.000	10.000	5.000
FINAL MINIMUM DELTA T(K)	5.000	10.000	5.000
U FACTOR(W/M**2 K)	545.000	567.700	545.000
LOG MEAN DELTA T(K)	12.640	26.717	8.638
AREA(M**2)	5659.515	3870.912	11332.280
HEAT TRANSFERRED(MW)	38.987	58.710	53.352

The plant cycle is illustrated in Fig. 3. Program GEOTHM divides the plant up into 13 fluid streams. The cycle has many components. These include: two 2-phase turbines, two flash tanks, one condensing heat exchanger, two air cooled condensers, one R-22 turbine, one R-22 pump, one vacuum pump and some condensate pumps. The results of the calculation are shown in Tables 1a and 1b. Table 1a presents basic power plant parameters and Table 1b shows the basic parameters of the heat exchangers.

The cycle shown in Fig. 3 is relatively insensitive to the effects of non-condensibles. This cycle is compared with a simple two-phase turbine cycle shown in Fig. 4. This cycle, which is similar to the cycle proposed by the Lawrence Livermore Laboratory[9], has a final condensation temperature of 324°K (the same as the complicated cycle second stage two-phase turbine exit temperature). The condenser is an air cooled condenser so that a fair comparison can be made between the two cycles. In both cycles, the two phase turbines are assumed to be 70% efficient.

Table 2. Power output per unit well flow vs percent non-condensibles for two thermodynamic cycles.

Percent non-condensibles	Yield per unit well flow (kWh per metric ton) Simple LLL cycle (see Fig. 4)	Complicated cycle (see Fig. 3)
0	18.844	16.847
0.5	18.555	16.736
1.0	18.267	16.736
3.0	17.112	16.736
5.0	15.959	16.736
7.0	14.866	16.736

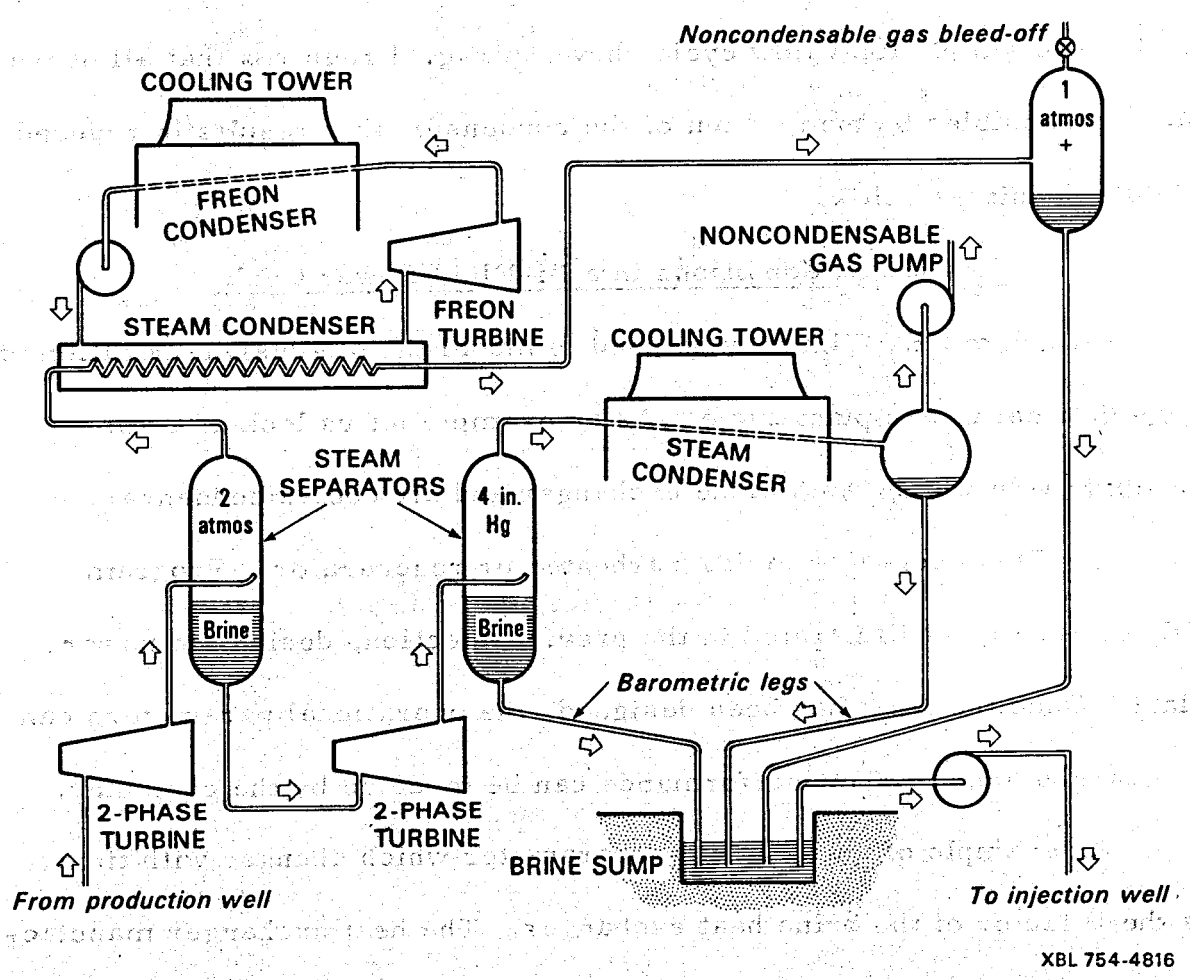


Fig. 3. Mixed power cycle with two-phase turbines and a bi-fluid loop which is insensitive to non-condensible gases.

Table 2 shows that non-condensibles can affect the performance of two-phase turbine cycles. Most of the non-condensibles are removed at a pressure above 1 atm in the cycle shown in Fig. 3. The remaining non-condensibles amount to about 0.2 percent of the original mass flow of the well. The simple total flow cycle shown in Fig. 4 requires that all of the non-condensibles be pumped out of the condensor; this results in reduced yield per unit well flow.

Off-Design Conditions in a Bi-Fluid Power Cycle

Program GEOTHM may be used to model the off-design operation of a geothermal power plant cycle. As an example let us look at a simple bi-fluid cycle which has a brine exchanger and air cooled condenser (see Fig. 5). This cycle has no desuperheater or regenerator. Program GEOTHM can, as illustrated in the previous section, design the power plant. Once the plant has been designed, the operational parameters can be changed and the plant performance can be modeled by the computer.

An example of an operational parameter which changes with time is the U factor of the brine heat exchanger. The heat exchanger manufacturer might suggest a design U factor of 568 W m^{-2} k^{-1} (100 Btu h^{-1} ft^{-2} F^{-1}). Experimental measurement tells us that the U factor for a clean heat exchanger may start out as high as 3000 W m^{-2} k^{-1} (530 Btu h^{-1} ft^{-2} F^{-1}). As the heat exchanger fouls up with salts from the brine, the heat exchanger U factor drops until it is so low the heat exchanger must be cleaned.

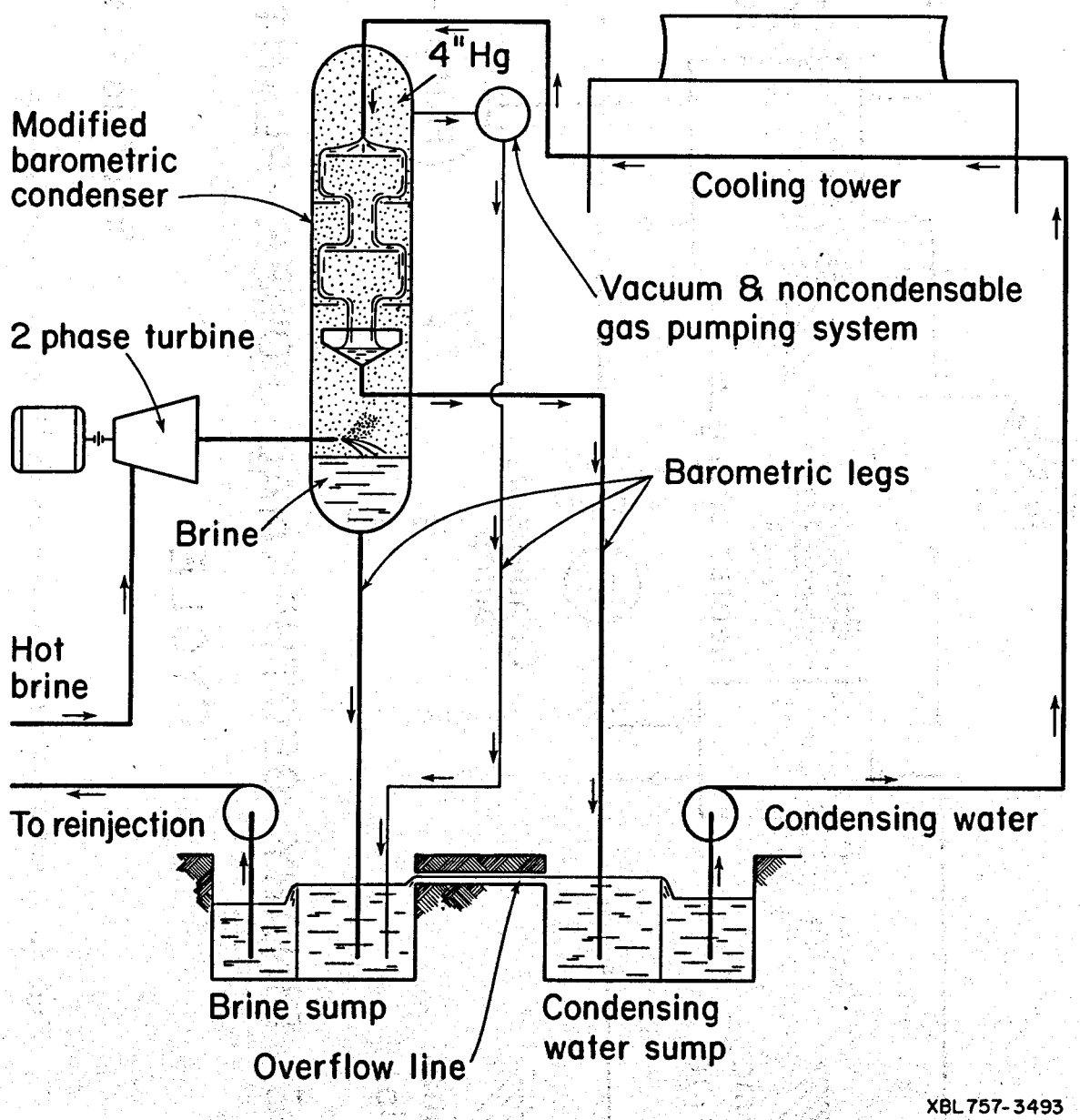
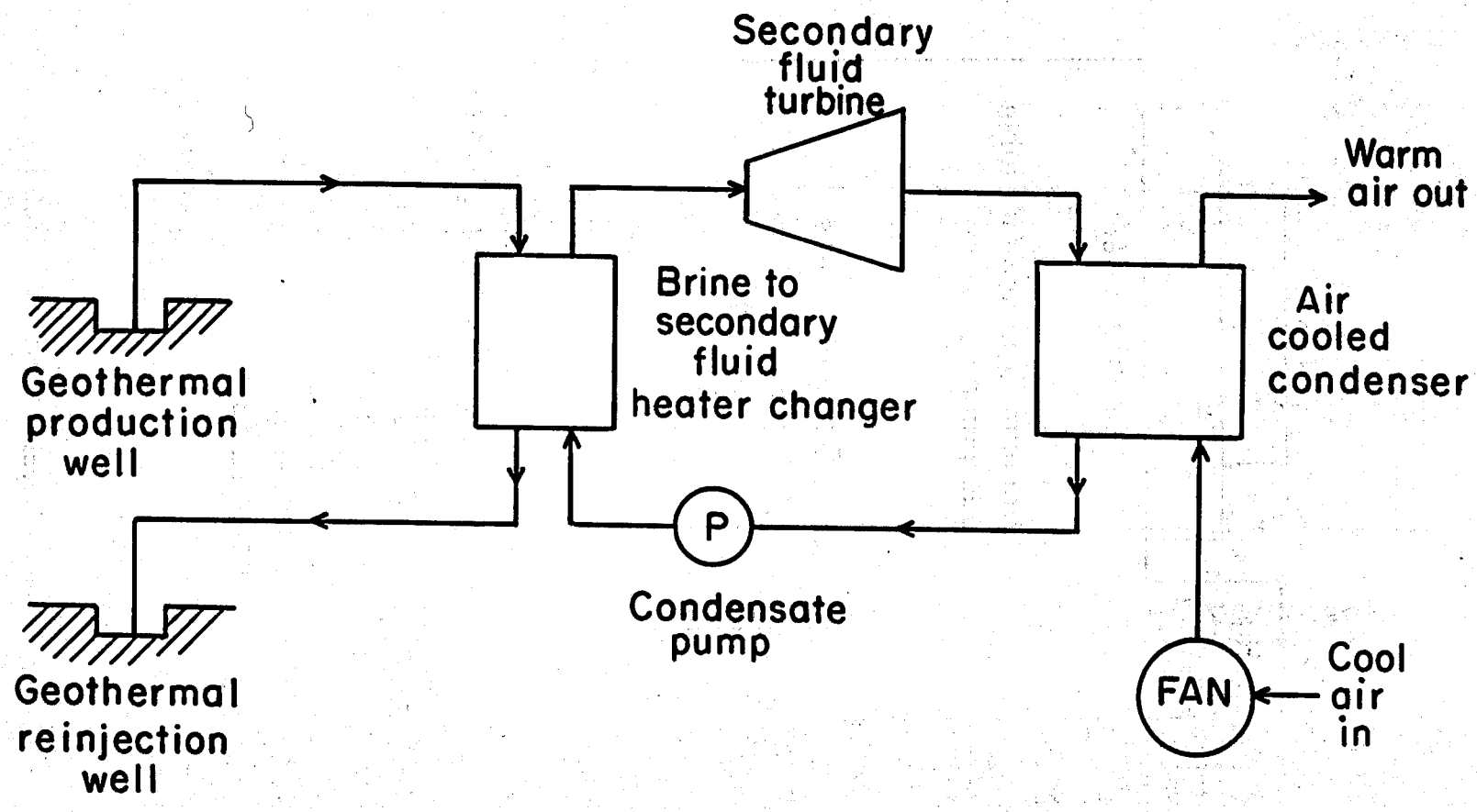


Fig. 4. Two-phase turbine power cycle (total flow cycle).



SIMPLE BINARY GEOTHERMAL POWER PLANT CYCLE

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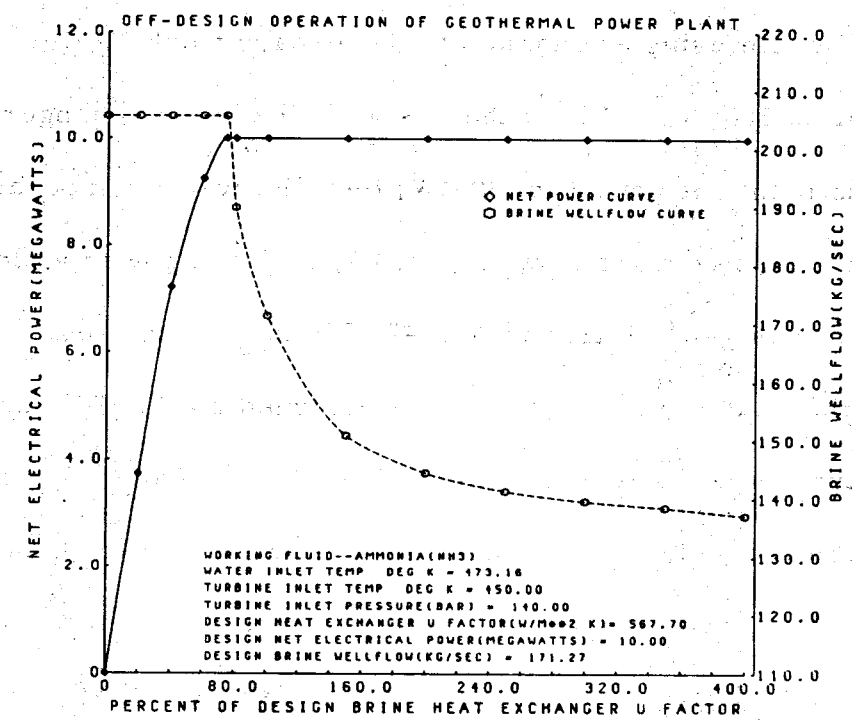
Fig. 5. A simple bi-fluid cycle with an air-cooled condenser.

We illustrate the effect of heat exchanger fouling by looking at the simple bi-fluid cycle using ammonia as a secondary working fluid. The design U factor is 568 W m^{-2} k^{-1}; the design brine heat exchanger area is 6440 m^2; the plant net power is 10 MW; the inlet water temperature is 200°C; the design water mass flow is 163.4 kg s^{-1} and power yield per unit well flow at design condition is 17.342 kWh per metric ton.

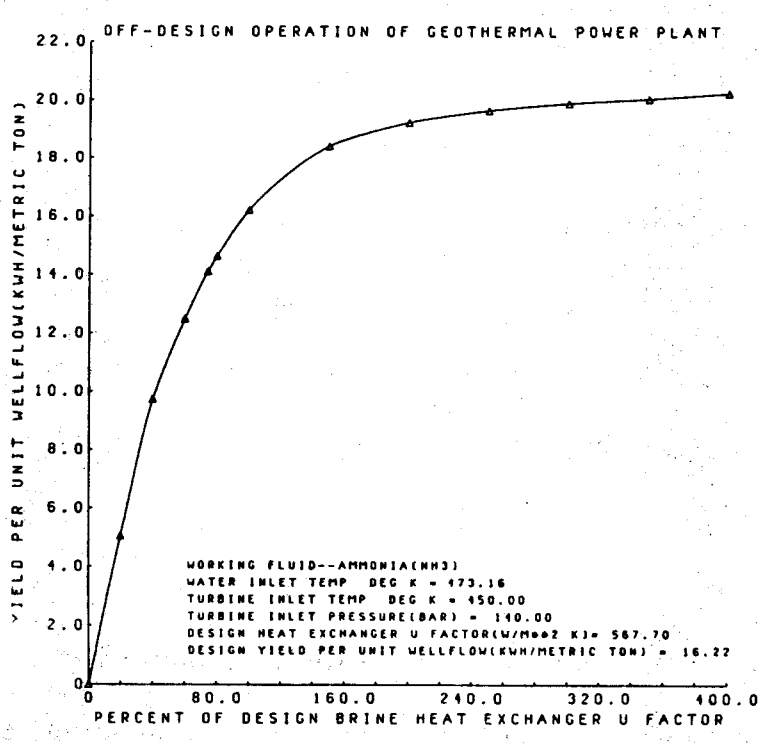
The geothermal power plant cycle is designed so that the turbine inlet temperature is maintained by regulating the flow from the wells. When the well flow reaches 120 percent of the design well flow, no additional fluid can be extracted from the wells so the turbine inlet temperature is allowed to drop. The result is a power plant which will maintain its net power rating until the well flow reaches 120 percent of design flow. The U factor of the brine heat exchanger started out at 400 percent of the design U factor (1870 W m^{-2} k^{-1}). The U factor of the brine heat exchanger dropped until it became zero.

The results of the fouling simulation calculations on the ammonia bi-fluid plant are shown in Figs. 6a and 6b. Figure 6a plots net plant power output and well mass flow as a function of the brine heat design U factor. Figure 6b plots the power plant yield per unit well flow (given in kWh per metric ton of 200 °C well water) vs the brine heat exchanger U factor. The results of the study show that there is considerable latitude in U factor over which the plant can be operated. U factors of 1870 W m^{-2} k^{-1} (the 400 percent case) don't increase yield very much. On the other

Fig. 6, Off-design operation of a simple bi-fluid cycle using ammonia as a working fluid.



a. Net plant power and well mass flow vs brine heat exchanger U factor.



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b. Plant yield per unit well flow vs brine heat exchanger U factor.

hand cutting the U factor to 50 percent of design value (233 W m^{-2} k^{-1}) cuts power production to 87 percent of the design output of the plant. Similar studies were made using isobutane as a working fluid; the results were similar.

Comparison of Various Working Fluids in Bi-Fluid Cycles

Program GEOTHM has been used to study the performance of various working fluids in various bi-fluid cycles. The performance of simple bi-fluid cycles (without a regenerator or desuperheater), such as shown in Fig. 5., has been calculated using a number of secondary working fluids.[10] A comparison of four working fluids is shown here. These fluids are (1) ammonia (R-717), (2) isobutane (R-600a), (3) Refrigerant 22 (R-22 is also known by chemical company trade names such as Freon-22), and (4) refrigerant 113 (R-113 is also known by various trade names). The four working fluids are used in cycles which have well water temperatures of 170°C, 200°C and 230°C.

The properties of the four working fluids are shown in Table 3.[11] For comparison, the properties of water (R-718) are also included in the table. The four fluids represent a spectrum of fluid properties. All of the fluids are reasonably stable at temperatures up to 230°C (R-22 is a possible exception because it probably should not be used at temperatures above 200°C).[12] A low molecular weight fluid (ammonia) and a high molecular weight fluid (R-113) are presented. Retrograde liquid vapor dome fluids (R-113 and isobutane) and non-retrograde fluids (ammonia is highly

Table 3. Properties of five working fluids used in bi-fluid geothermal power plant cycles.

Properties	Working Fluids: Water	Ammonia	Isobutane	Refrigerant 22	Refrigerant 113
ASHRAE designation	R-718	R-717	R-600a	R-22	R-113
Chemical formula	H_2O	NH_3	$CH(CH_3)_3$	$CHCl F_2$	$CCl_2F-CCl F_2$
Molecular weight	18.0	17.0	58.1	86.5	187.4
Critical temperature (K)	647.3	405.4	408.1	369.2	487.3
Critical pressure (bar)	221.1	112.7	36.5	49.8	34.1
Critical volume ($cm^3 g^{-1}$)	3.139	4.243	4.518	1.905	1.735
Critical compressibility	0.232	0.242	0.282	0.267	0.274
Normal boiling temperature (^oK)	373.2	239.8	261.4	233.2	320.7
Heat of vaporization ($J g^{-1}$) at normal boiling point	2259.5	1371.1	364.6	233.4	147.0
Type of liquid vapor dome	non-retrograde	non-retrograde	retrograde	non-retrograde	retrograde
Toxicity (Underwriters Lab. classification)	Group 6+ (non-toxic)	Group 2 (toxic)	Group 5b (non-toxic)	Group 5a (non-toxic)	between Groups 4 & 5 (M-toxic)
Flammability	non-flammable	M-flammable	V-flammable	non-flammable	non-flammable

non-retrograde; R-22 is slightly non-retrograde) are shown. (A retrograde fluid expands dryer along an isentrope; a non-retrograde fluid expands wetter along an isentrope.)

The parameter study presents the electrical power yield per unit well flow (in kilowatt hours of electricity per metric ton of well water passed through the power plant) as a function of the secondary working fluid and the well water inlet temperature. For each well water inlet temperature and working fluid, the turbine inlet temperature and pressure was varied. The pump inlet conditions corresponded to the saturated liquid condition for the working fluid at 311°K (38°C). The air-cooled condenser air inlet temperature was set at 298°K (25°C). The pinch point temperature drops were set at 10°C for the brine heat exchanger and 5°C for the air-cooled condenser. The well water was assumed to have the thermodynamic properties of pure water.

Table 4 presents the electric power yield per unit well flow for the four working fluids and the three water inlet temperatures. For each of the twelve cases the yield is given for two turbine inlet conditions. Table 4 shows the obvious; the electric power yield per unit well flow rises markedly with water inlet temperature. Since the best cases were chosen for each working fluid, it is not surprising that the power yield is comparable between the four working fluids. Therefore, one might be hard pressed to eliminate any of the four working fluids on the basis of power yield per unit well flow.

Table 4. Electric power yield per unit well flow as a function of working fluid and well water inlet temperature. (Each case has two turbine inlet conditions.)

Working fluid	Well water inlet temperature (°K)	Turbine inlet pressure (bar)	Turbine inlet temperature (°K)	Yield per unit well flow Wh kg^{-1} or kWh ton^{-1}
R-717 (ammonia)	443.16 (170°C)	120	390 (117°C)	7.8
			420 (147°C)	10.3
	473.16 (200°C)	120	420 (147°C)	17.3
			450 (177°C)	17.2
	503.16 (230°C)	140	450 (177°C)	25.8
			480 (207°C)	26.6
R-600a (isobutane)	443.16 (170°C)	40	390 (117°C)	7.7
			420 (147°C)	11.2
	473.16 (200°C)	40	420 (147°C)	19.0
			450 (177°C)	17.7
	503.16 (230°C)	60	450 (177°C)	25.3
			480 (207°C)	26.0
R-22	443.16 (170°C)	60	390 (117°C)	9.6
			420 (147°C)	11.2
	473.16 (200°C)	60	420 (147°C)	15.6
			450 (177°C)	17.2
	503.16 (230°C)	80	450 (177°C)	22.8
			480 (207°C)*	24.4
R-113	443.16 (170°C)	20	390 (117°C)	7.3
			420 (147°C)	11.7
	473.16 (200°C)	20	420 (147°C)	14.9
			450 (177°C)	19.7
	503.16 (230°C)	40	450 (177°C)	24.4
			480 (207°C)	27.6

*R-22 may be too unstable for use at this temperature.

It is clear from Table 4 that more investigation is needed in order to clearly show a definite preference for one working fluid over another. Plots of yield per unit well flow vs turbine inlet temperature and pressure, such as those shown in Fig. 7a through Fig. 7d for 200°C inlet water,

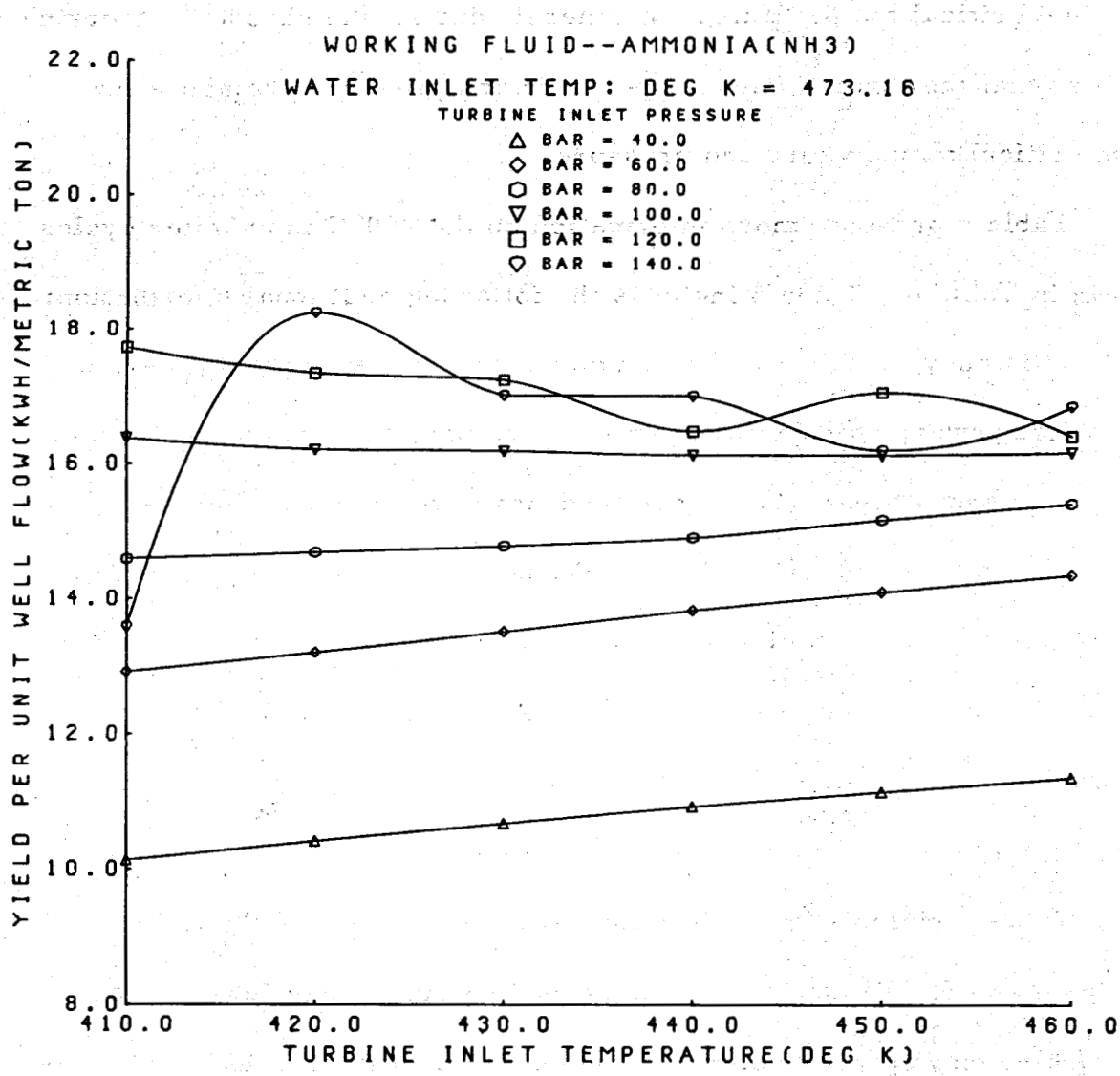
will yield more information on the behavior of the fluids. For example, cycles which have turbine inlet temperatures below the fluid critical temperature should have turbine inlet pressures which are subcritical. Supercritical cycles are best when the turbine inlet temperature is above the fluid critical temperature. In general, one should choose the working fluid so that the turbine inlet temperature and pressure are above the fluid critical temperature and pressure.

Table 5 presents more information on the 200°C inlet water cycles shown in Table 4. Table 5 includes the following additional information; cycle efficiency, turbine power, pump power, condenser fan power, net electrical power, brine reinjection temperature, area of the brine and condenser heat exchangers (the condenser area is bare tube area), and a raw capital cost factor. (The actual cost of the complete power plant complex including the wells is almost double the raw capital cost factor, which is a basic component cost.) A close look at Table 5 shows the two freon plants (R-22 and R-113) require more heat exchanger area and equipment, hence their capital cost is higher.

Table 6 yields even more information on the cycles shown in Tables 4 and 5. Table 6 shows the secondary fluid mass flow rate and the turbine exit specific volume. The exit area of the turbine, which for all practical purposes determines the physical size of the turbine, is a function of the product of secondary fluid mass flow and turbine exit specific volume. Similar arguments can be applied to plant piping.

Fig. 7. Power plant yield per unit well flow for a simple bi-fluid cycle vs turbine inlet temperature and pressure.

Inlet water temperature 200°C
Inlet air temperature 25°C
Condensing temperature 38°C



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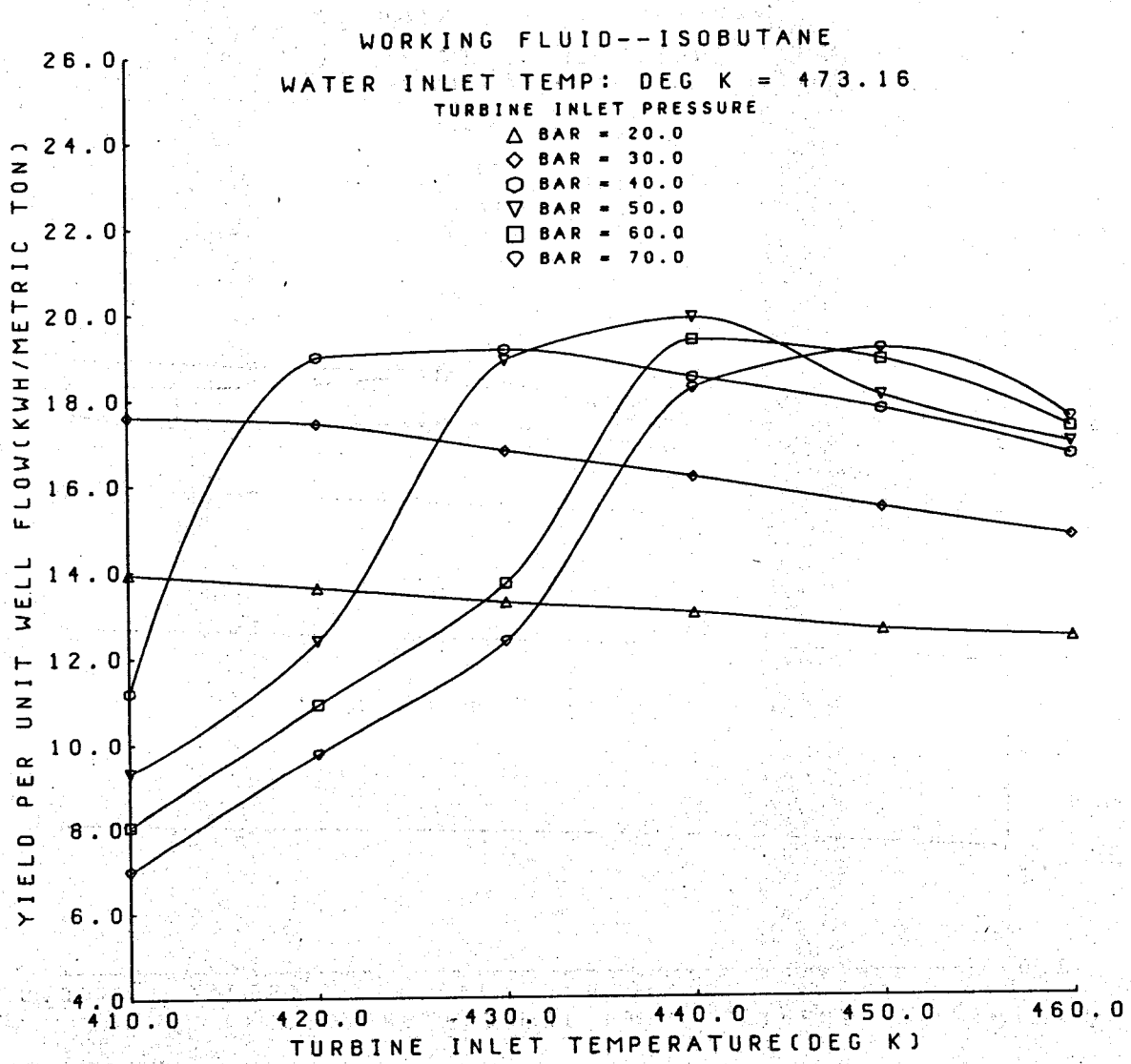
7a. An ammonia (NH_3) cycle.



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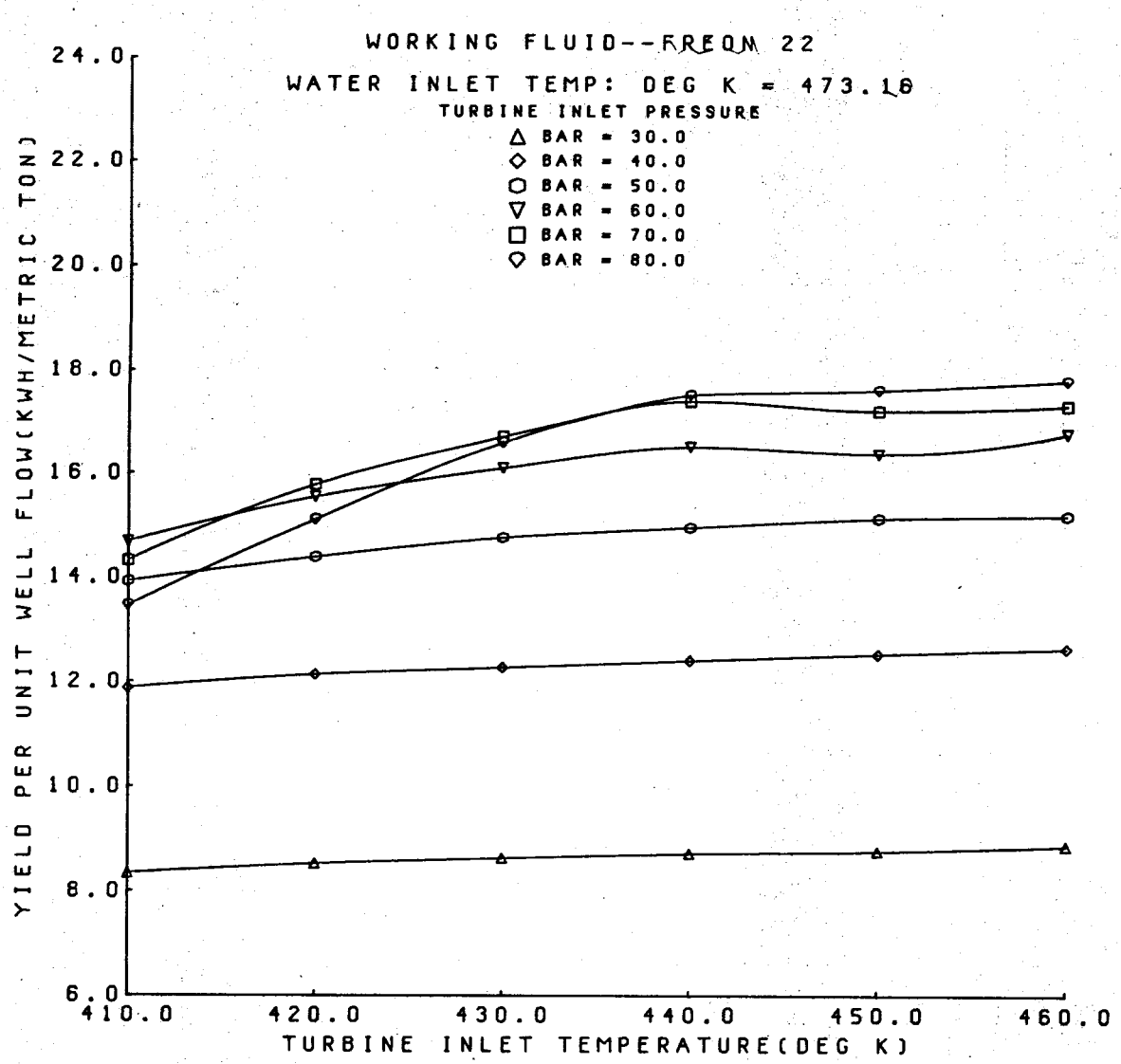
7b. An isobutane ($CH(CH_3)_3$) cycle.



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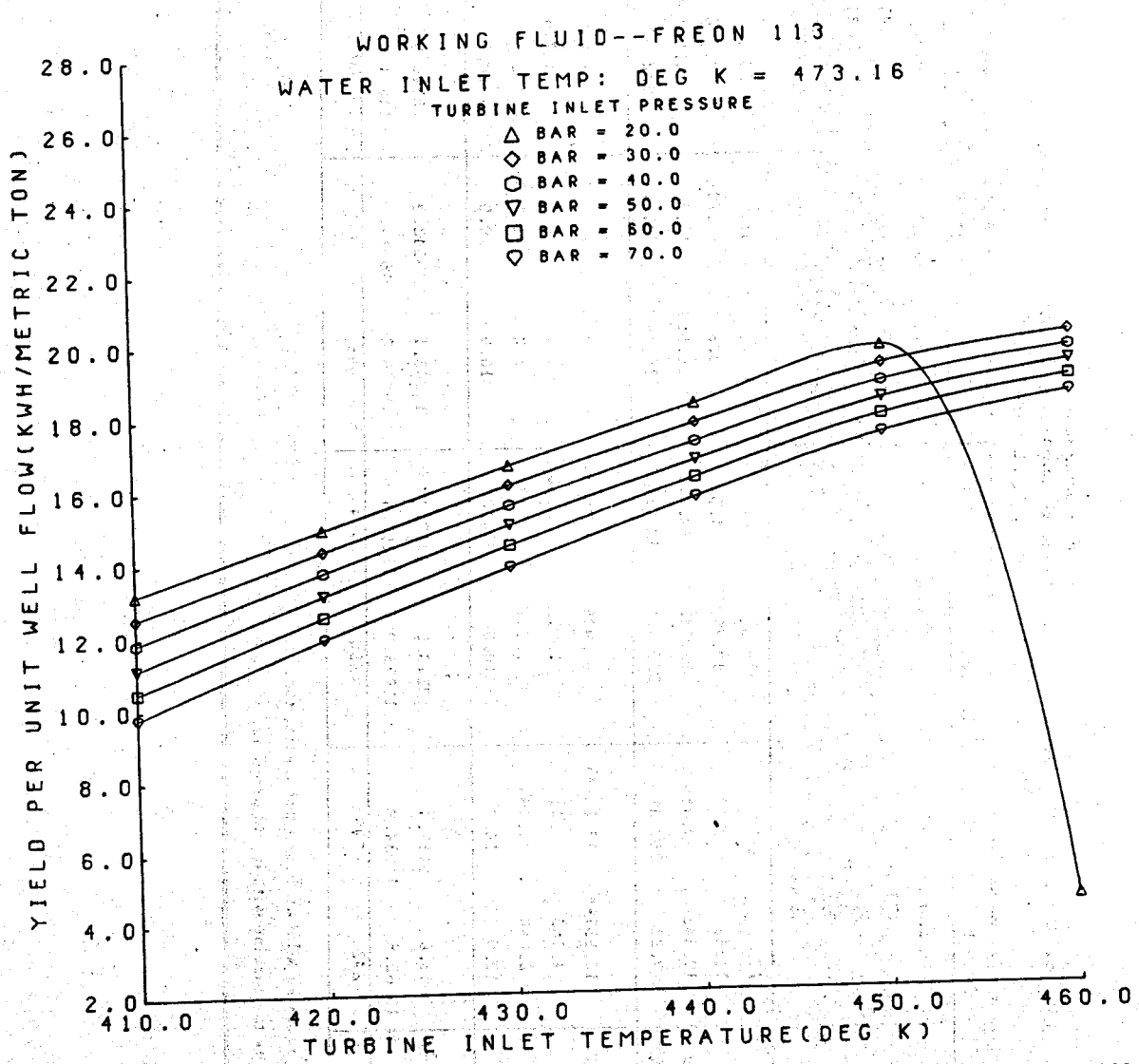
7c. A refrigerant 22 ($CH\,Cl\,F_2$) cycle.



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7d. A refrigerant 113 ($C\,Cl_2F$ - $C\,Cl\,F_2$) cycle.

Table 5. Detailed cycle parameters for eight cycles using four secondary working fluids. Well temperature 200°C, Ambient air temperature 25°C.

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8
Working fluid in loop	ammonia		isobutane		Refrigerant 22		Refrigerant 113	
Well water temperature (°K)	473.16		473.16		473.16		473.16	
Pump inlet temperature (°K)	311.00		311.00		311.00		311.00	
Pump inlet pressure (bar)	14.75		5.10		14.85		0.76	
Turbine inlet pressure (bar)	120.0		40.0		60.0		20.0	
Turbine inlet temperature (°K)	420.	450.	420.	450.	420.	450.	420.	450.
Cycle efficiency (%)	11.52	12.81	10.79	11.51	8.88	9.81	8.348	11.053
Yield per unit well flow (Wh kg^{-1})*	17.342	17.158	18.982	17.718	15.550	16.374	14.931	19.739
Water reinjection temperature (°K)	347.21	362.19	325.48	344.33	326.14	333.20	322.90	323.14
Power								
Turbine power output (MW)	13.857**	13.289**	13.905	12.948	15.346	14.213	14.772**	13.234**
Pump power input (MW)	1.563	1.168	1.823	1.352	2.707	2.111	1.773	1.121
Fan power input (MW)	2.040	1.794	1.731	1.283	2.231	1.740	2.614	1.889
Net electrical power (MW)	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0
Heat exchanger area								
Brine heat exchanger area (m^2)	7613.6	6439.4	10348.6	9417.4	10003.1	9042.9	8596.9	12027.4
Condenser area (m^2)	15862.8	14034.9	15775.0	12256.4	19956.2	16044.7	21871.6	16099.5
Raw capital cost factor ($/kW)	842	799	850	794	995	892	996	821

* Watt hours per kilogram is the same as kilowatt hours per metric ton.

** Wet turbine, working fluid leaves the turbine with liquid droplets.

Table 6. Turbine exit parameters for the 10 MW net power cycles shown in Tables 4 and 6.

Case	Working fluid	Turbine inlet pressure (bar)	Turbine inlet temperature (°K)	Gross turbine power (MW)	Working fluid mass flow (kg s^{-1})	Turbine exit specific volume (cm^3 g^{-1})	Approximate turbine exit area (m^2)
1	Ammonia (R-717)	120	420	13.857	83.7	66.78*	0.188
2			450	13.289	62.6	81.01*	0.170
3	Isobutane (R-600a)	40	420	13.905	217.6	86.34	0.628
4			450	12.948	161.4	101.42	0.537
5	R-22	60	420	15.346	510.3	18.35	0.311
6			450	14.213	397.9	21.45	0.284
7	R-113	20	420	14.772	1060.6	117.04*	4.13
8			450	13.234	610.7	148.63*	3.02

* Two phase fluid from wet turbine.
A water turbine which exits at 311°K would have an area of about 0.5 m^2 per megawatt of power rating.

While it is too early to select the best bi-fluid geothermal power plant working fluid, one can make some general statements about desirable working fluid properties. For example, (1) one should select a fluid which can be used in a supercritical cycle. (2) Low molecular weight fluids result in less mass flow in the circuit hence less pump work, smaller piping, and smaller turbines. (3) Non-retrograde fluids require less desuperheating. (4) Expansion of the fluid wet has advantages in the condenser, if the turbine can tolerate it. The effect of fluid toxicity, flammability, cost and compatibility with lubricants will all enter into the selection of a secondary working fluid for bi-fluid cycles. It is very likely that the selection of the best working fluid will not come until after the plant cycle is cost optimized for minimum cost power.

Cost Optimization - The Next Developmental Phase

Program GEOTHM can be used to do power plant parameter studies of various kinds, as was illustrated in the previous section. One may also study the effect of various plant parameters on power plant capital cost and the cost of power to be generated by that plant. While cost parameter studies may yield useful information, one must eventually optimize the power plant cycle to produce minimum cost power. Since there are many parameters to juggle, the computer is well suited for the task.

Let us look at the simple bi-fluid cycle shown in Fig. 5. This cycle is extremely simple, yet there are six major parameters which must be considered while optimizing the cost of that plant. They are: (1) the turbine inlet temperature, (2) the turbine inlet pressure, (3) the feed pump

inlet temperature, (4) the feed pump inlet pressure (the feed pump must be fed with liquid), (5) the pinch point temperature drop in the brine heat exchanger, and (6) the pinch point temperature drop in the condenser heat exchanger. The initial well bottom enthalpy (or temperature) and the air temperature are given. A six-dimensional parameter study is difficult to make even with a computer. As a result, cost optimization should proceed as a single process.

Since the cost equations are highly non-linear, one must calculate the optimum plant parameters using an iterative process. One starts out with a first guess power plant. One calculates the electrical energy cost from the plant, then the first and second derivatives of that cost with respect to each of the optimizable parameters. Using iterative technique similar to Newton's method one can, with luck, converge on an optimum power plant design in a relatively short time. Given reliable cost data and a good thermodynamic model of the power plant and the geothermal field, the computer will prove invaluable for developing power plant cycles which result in economic electrical energy.

Summary

This paper has demonstrated a number of applications for which program GEOTHM can be used. The application of the program is by no means limited to the cases shown, nor is the program limited to just geothermal power cycles. It is expected that program GEOTHM can be applied to a host of thermodynamic systems which use a variety of working

fluids to generate mechanical power, refrigeration, or transfer thermal energy from one place to another. Program GEOTHM is growing; its development is not complete. This report represents program GEOTHM as a photograph represents a growing child.

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