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THE OPTIMIZATION OF ALTERNATIVE ENERGY CYCLES USING PROGRAM GEOTHM

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ABSTRACT

GEOTHM is a thermodynamic process program which will calculate a wide variety of thermodynamic cycles using a variety of working fluids. GEOTHM has a unique optimizing ability which permits a thermodynamic cycle to be optimized for minimum cost, maximum efficiency or any other user-specified parameter. GEOTHM can optimize a complicated cycle with many parts in a single step optimization. The program has been used to optimize cycles with over 20 optimizable parameters.

The optimization process is quite different for most alternative energy cycles because the energy source is often diffuse or it has a low temperature. Three examples of optimized power cycles are presented here in order to illustrate the broad capabilities of the GEOTHM code.

INTRODUCTION

Lawrence Berkeley Laboratory has been developing the GEOTHM computer program since December 1973 [1]. Program GEOTHM can be applied to a wide variety of alternative energy cycles. Until recently the GEOTHM program has been applied only to Geothermal energy cycles[2,3,4]. (The program name comes from this application.) This paper will demonstrate the use of GEOTHM in three types of power cycles. These are: 1) a Geothermal power plant which uses a binary or bi-fluid cycles, 2) a simple ocean thermal power cycle, and 3) a power cycle which is a combined gas turbine and a Rankine bottoming cycle.

GEOTHM is an extremely versatile thermodynamic cycle simulator. Its primary features are:

1. The thermodynamic processes are modularized into fundamental building blocks. The blocks can be arranged into any type of thermodynamic system.

2. The calculation of thermodynamic properties and transport properties is separated from the thermodynamic process elements.

3. The program is fast due to efficient programming in all of its iterative convergence routines(i.e., Newton's Method).

4. The thermodynamic cycle generator in GEOTHM can be used as a function generator which is steered by a mathematical optimizer. The optimization routine can design and optimize a thermodynamic cycle with respect to any user specified criterion.

5. The program is interactive. A preprocessor corrects many errors which the user makes in inputing data into GEOTHM. The program, which can be run from a remote terminal, is user oriented[5].

GEOTHM is a large computer program with over 100 FORTRAN subroutines which contain over 10,000 FORTRAN statements. Since GEOTHM is a very large program, its use has been limited to the CDC-7600 computer system.

Before proceeding to the sample cycle calculations, it is useful to review the primary features of the program. GEOTHM has subroutines which perform various thermodynamic processes. The processes that GEOTHM can model include: turbines, pumps, fans, compressors, isenthalpic expanders, flash tanks, direct contact condensers, surface heat exchangers, surface condensers, pipe, geothermal wells, and fossil fuel burners.

Fluid properties are calculated using several equations of state. The program includes equations of state for pure water and sodium chloride brines with a concentration of up to 300,000 ppm[6]. The Starling BWR equation of state is used with non-chlorinated hydrocarbons[7]. Two forms of the Martin equation of state can be used for the Freons and ammonia[8,9]. Air, noncondensible gasses and products of combustion are represented by a modified ideal gas equation of state. Thermal transport properties currently coded were obtained from the National Bureau of Standards[10].

GEOTHM can be operated in two modes, the passive design mode and the dynamic design mode. When the program operates in the passive design mode, the program designs the power plant using user specified state parameters and thermodynamic processes. When the optimizer steers the state parameters (now called optimizable parameters), the power plant is designed and optimized using the dynamic design mode. Optimization is a one step process in multidimensional space. Optimization of power cycles with up to 55 optimizable parameters is possible because the program is extremely fast. Any reasonable objective function can be optimized using the program. We have optimized geothermal power cycles for: 1) minimum plant capital cost, 2) minimum bus bar energy cost, 3) maximum energy yield per unit well flow, and 4) maximum cycle thermodynamic efficiency.

CYCLE OPTIMIZATION

The optimization process for most alternative energy cycles usually proceeds along different lines from conventional fossil fuel or nuclear cycles. Most alternative energy cycles are characterized by low energy density and or low thermodynamic availability. The former characterizes virtually all solar cycles; while the latter is characterized by the low temperature waste heat cycles. There are large sources of energy which have either low energy density or low temperature. The economic viability of these cycles is questionable. Therefore, the economic optimization of such cycles requires careful consideration of all cycle parameters.

A typical fossil fuel power cycle has relatively few parameters which must be optimized simultaneously. The temperature and pressures in the cycle are, in general, limited by technology. The highest temperature and pressure are therefore, nonoptimizable. In fossil fuel plants the regeneration and reheat processes are an important part of the optimization process; a great deal of effort is expended in optimizing these parameters. Until recently, relatively little effort has been expended in doing a multiparameter optimization process that is needed to create a cycle which is the most economical. The sample cycles which are given in this report are optimized in at least seven dimensional space. They all require careful optimization of the heat exchangers and the heat rejection system.

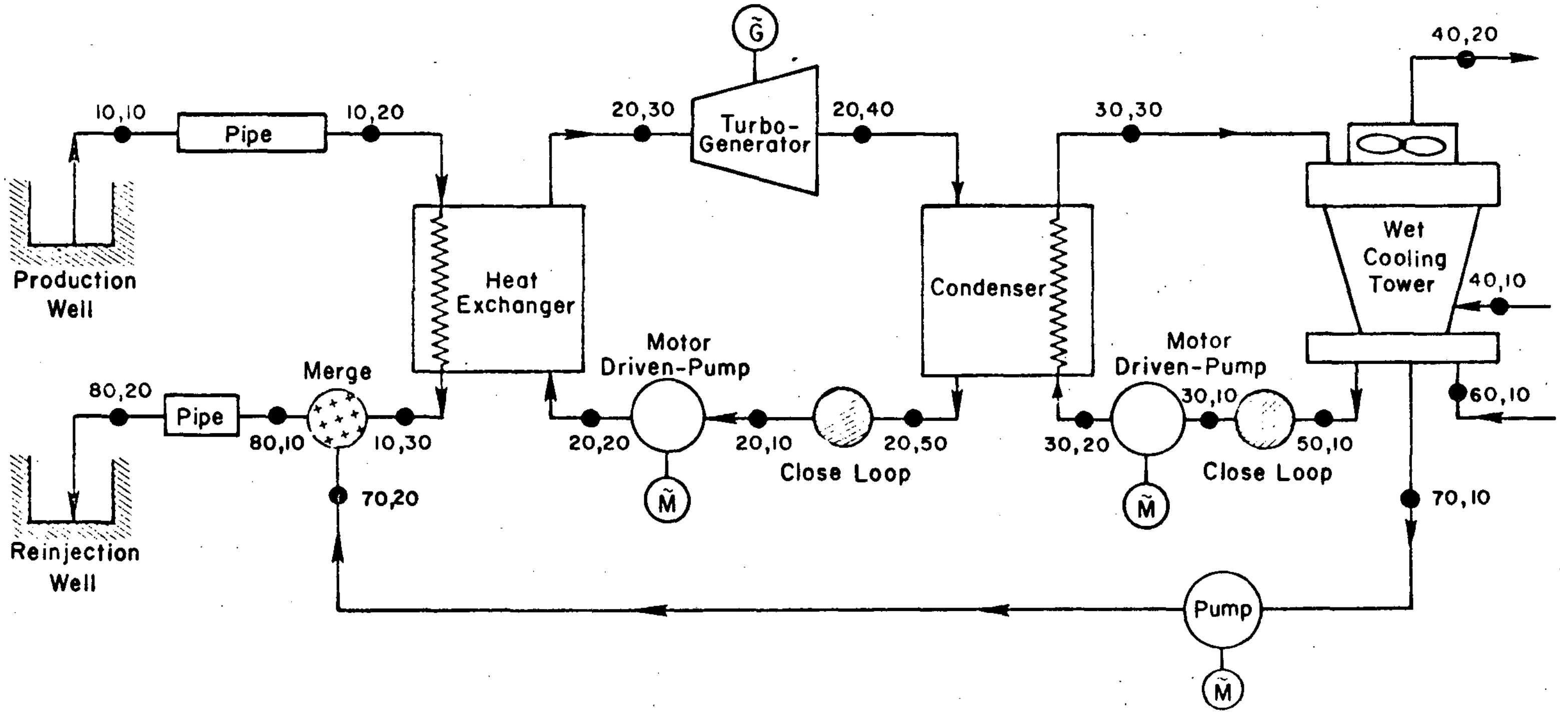
The three example cycles given in this report all generate 50 MW of electric power at the bus bar. The mass flows of the *fuel stream* (the term *fuel* here is defined in a very broad way) vary from hundreds of metric tons per second in the ocean thermal cycle to a kew kilograms per second for the gas turbine cycle. In each case the cost of *fuel* has a different impact on the optimization of the cycle.

The first sample cycle, a geothermal binary cycle, shows clearly the trade off between thermodynamic efficiency and economic viability. This cycle illustrates that a power cycle can be optimized to maximize thermodynamic performance or to minimize energy cost. The second sample cycle, an ocean thermal cycle, is even more extreme than the first sample cycle. The thermodynamic availability is extremely limited by the narrow temperature difference between surface waters and bottom waters. In this cycle, the friction loss in piping (particularly the water piping) becomes an optimizable parameter. The third cycle, which is a fossil fuel gas turbine cycle which tops an organic working fluid cucle, illustrates the interplay between two different thermodynamic cycles which have different characteristics.

OPTIMIZATION OF A 50 MW GEOTHERMAL BINARY POWER PLANT

The cycle shown in Figure 1 has been optimized for minimum bus bar energy cost. This cycle represent one of the cycles which is proposed for a geothermal power plant to be built in the Imperial Valley of California[11]. The cost coefficients for various plant components were fitted to cost quotations obtained by a major engineering firm[12]. The heat transfer coefficients assumed for the brine heat exchanger and the condenser came from the same source. The brine leaving the well is assumed to be pure water. The secondary working fluid is isobutane. (See Reference 12 for other details.)

The cycle shown in Figure 1 has six major optimizable parameters which are: 1) the turbine inlet temperature, 2) the turbine inlet pressure



SIMPLE BINARY CYCLE

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Figure 1. A Simple Binary Geothermal Cycle.

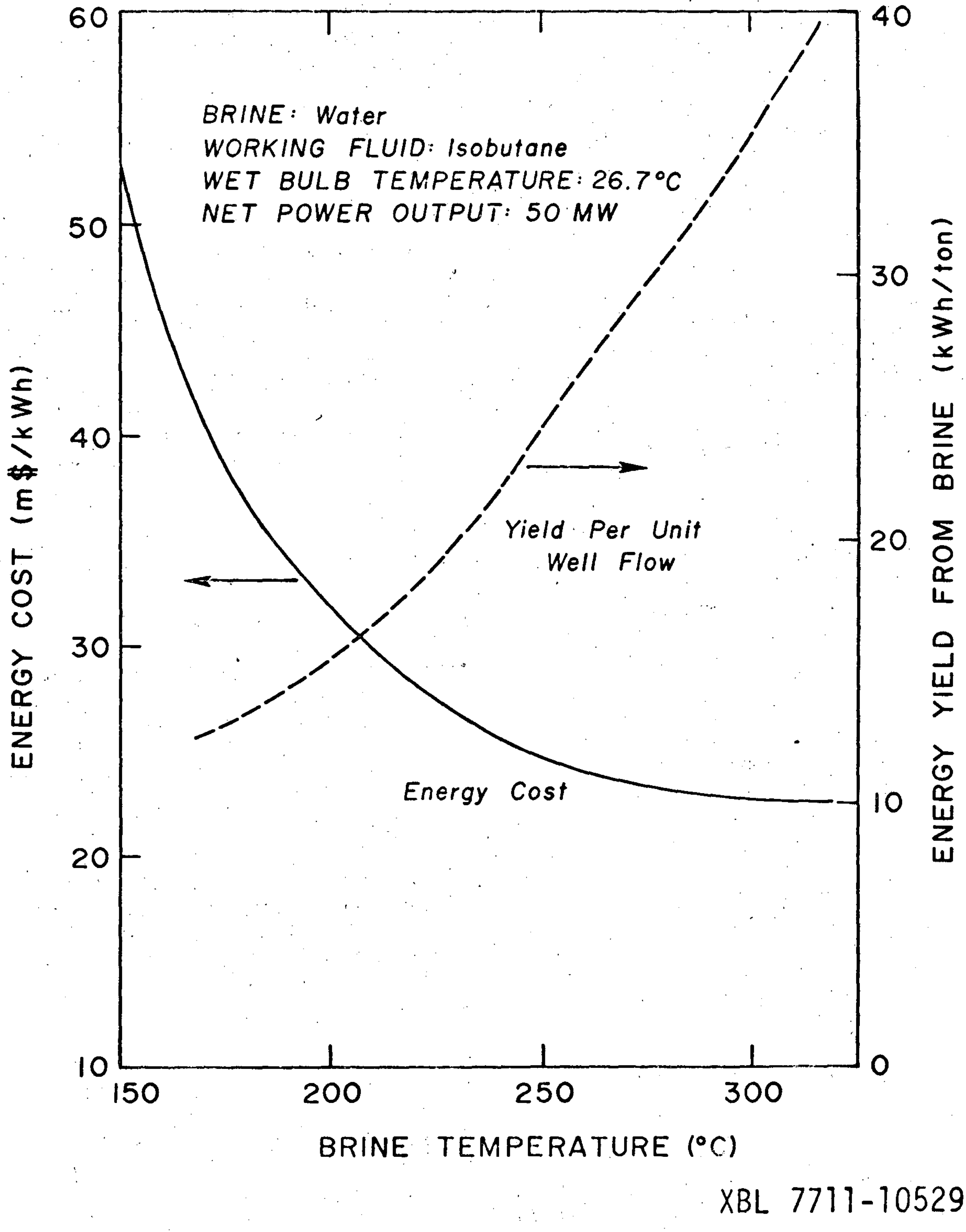

Figure 2. Electrical Energy Cost and Energy Yield from the Brine as a Function of Brine Inlet Temperature in the Cycle Shown in Figure 1.

3) the condensing pressure, 4) the pinch point temperature difference for the brine heat exchanger, 5) the pinch point temperature difference in the condenser, and 6) the exit temperature of the water leaving the cooling tower. A typical passive mode cycle design of the cycle shown in Figure 1 takes about 75 milliseconds of 7600 computer time. Dynamic mode optimization takes around 20 seconds using the previously given optimizable parameters.

Figure 2 shows the optimum bus bar energy cost and the energy yield per unit well flow for the cost optimized plant as a function of the inlet temperature of the geothermal water entering the plant. Table 1 makes a comparison of various plant parameters for 50 MW net electric power plants which have been optimized for minimum bus bar energy cost and maximum yield per unit well flow. This comparison of parameters is made at geothermal resource temperatures of 175°C and 250°C. From Table 1, one can see that the optimizable parameters change considerably when a plant is optimized for a maximum theoretical yield per unit fuel flow (flow from the geothermal wells) instead of minimum bus bar energy cost. One would not build a plant which maximized the yield per unit well flow simply because it makes no economic sense.

Figure 3 shows the three dimensional surface for maximum resource utilization efficiency (yield per unit well flow over maximum yield per unit flow if the geothermal fluid were expanded isentropically from the resource temperature to the wet bulb temperature) versus the temperature and pressure of the inlet of the turbine. (All other optimizable parameters have been set at the optimum value.) This surface was generated for a 200°C resource. Figure 4 is a three dimensional plot of energy cost at the bus bar versus resource temperature and resource utilization efficiency. This surface shows that minimum cost energy is produced at a resource utilization efficiency of around 40 percent[12]. This is 60 to 70 percent of the maximum resource utilization efficiency possible for the cycle.)

Optimization of geothermal power cycles can be extended beyond plant design once the plant has been designed, the optimizer can be used to maximize electric power output at other conditions besides the plant design conditions. As an example, the GEOTHM optimizer can be used to maximize the net power output from the power cycle shown in Figure 1 when the air wet bulb temperature is reduced.

OPTIMIZATION OF A SIMPLE OCEAN THERMAL POWER CYCLE - A 50 MW PLANT

The power cycle shown in Figure 5 represents a simple ocean thermal power plant cycle. This cycle is in many ways similar to the cycle shown in Figure 1. Hot water (lukewarm ocean water) is drawn in at one end. It is used to evaporate an organic working fluid. In this example isobutane is used. (This is probably not the best working fluid to use; ammonia is probably better.) Cold water is used to cool the heat exchanger condenser. The cycle in Figure 5 was programmed to illustrate the capabilities of the GEOTHM program. It is not necessarily the best cycle to use for an ocean thermal power plant.

Table 1. A comparison of cycle parameters for 50 Mw power plants which have been optimized for minimum bus bar energy cost and maximum yield per unit well flow (reference 12).

PARAMETERS	MINIMUM COST OF ENERGY OPTIMIZATION		MAXIMUM YIELD PER UNIT WELL FLOW OPTIMIZATION	
	Brine Temperature		Brine Temperature	
	175°C	250°C	175°C	250°C
General Parameters				
Bus Bar Energy Cost (m$/kWh)	38.23	24.47	∞	∞
Plant Yield (kWh/ton)*	13.044	24.438	19.109	39.499
Cycle Efficiency (%)	11.30	12.40	12.88	15.88
Brine Mass Flow (kgs^{-1})	1065.	568.	727.	351.
Turbine Gross Power (MW)	67.77	66.64	65.78	72.82
Plant Capital Cost ($/kW)	849.	568.	∞	∞
Optimizable Parameters				
Turbine Inlet Pressure (bar)	43.56	61.89	62.99	129.54
Turbine Inlet Temperature (°C)	154.04	214.70	164.50	237.72
Condenser Pressure (bar)	6.26	7.28	4.29	4.19
Brine Heat Exchanger Pinch Point - ΔT (°C)	7.29	23.40	0.	0.
Condenser Pinch Point - ΔT (C°)	4.29	5.46	0.	0.
Cooling Tower Water Temperature (°C)	33.91	35.06	26.67	26.67

*kWh per metric ton of geothermal brine processed.

Net Electric Power	50.0 MW	Brine Heat Exchanger Average U Factor	1514 $Wm^{-2}K^{-1}$
Air Dry Bulb Temperature	48.89 °C (120 °F)	Condenser Average U Factor	~500 $Wm^{-2}K^{-1}$
Air Wet Bulb Temperature	26.67 °C (80 °F)		

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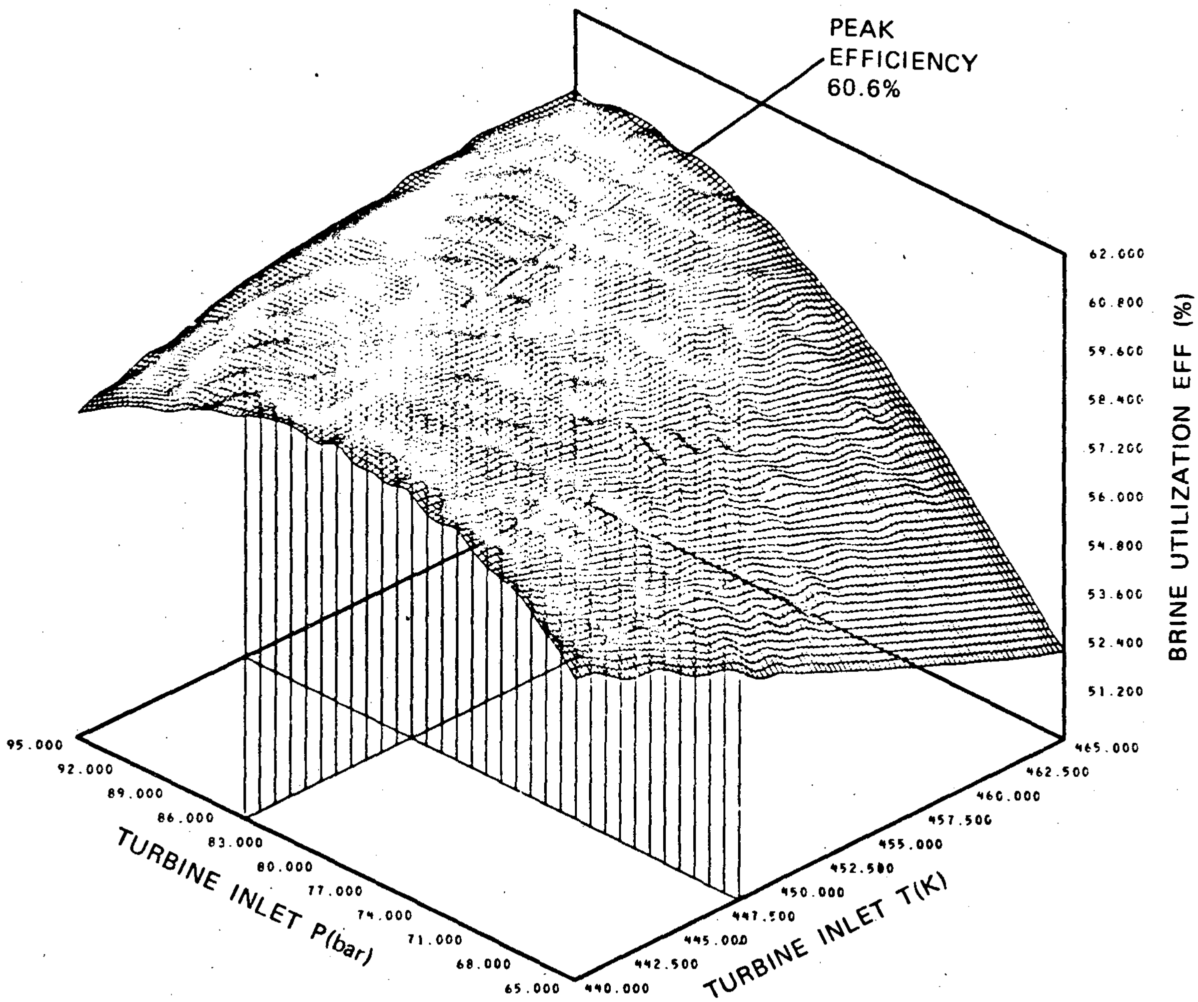

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Figure 3. Brine Utilization Efficiency as a Function of Turbine Inlet Pressure and Turbine Inlet Temperature.

Resource Temperature	200 °C
Condensing Pressure	4.26 bar
Heat Exchanger Pinch Point Temperature Difference	0
Condenser Pinch Point Temperature Difference	0
Cooling Tower Water Temperature	26.67 °C

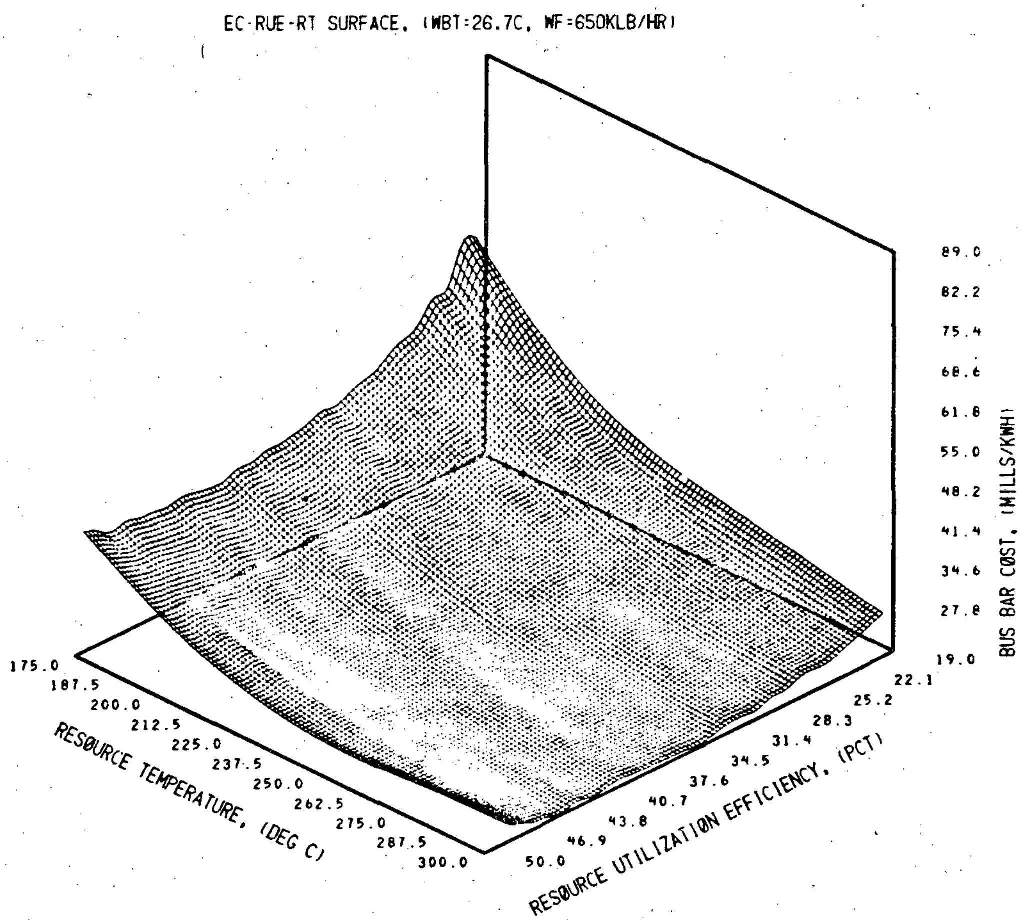

Figure 4. Bus Bar Electrical Energy Cost as a Function of Resource Temperature and Resource Utilization Efficiency (50 MWe Isobutane Geothermal Binary Cycle, Reference 12).

Air Drying Bulb Temperature	48.89 °C
Air Wet Bulb Temperature	26.67 °C
Net Electric Power	50.0 MW

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Since we lack expertise in ocean thermal systems, we have made a number of simple assumptions, which are given as follows:

1. The warm water inlet pipe is 500 m long extending horizontally away from the plant. The cold water inlet pipe which is 1000 m long extends to a depth of 1000 m below the surface of the water. The outlet pipes for both the warm and cold water systems are 200 m long.

2. Inlet water temperatures of 20 and 25°C are used. An inlet cold water temperature of 5°C is assumed.

3. Turbine and pump efficiencies of 85 percent are assumed.

4. Two heat exchanger U factors are assumed in this study. The heat exchangers are the major plant items, the heat transfer per dollar per degree C has an important affect on the plant optimization.

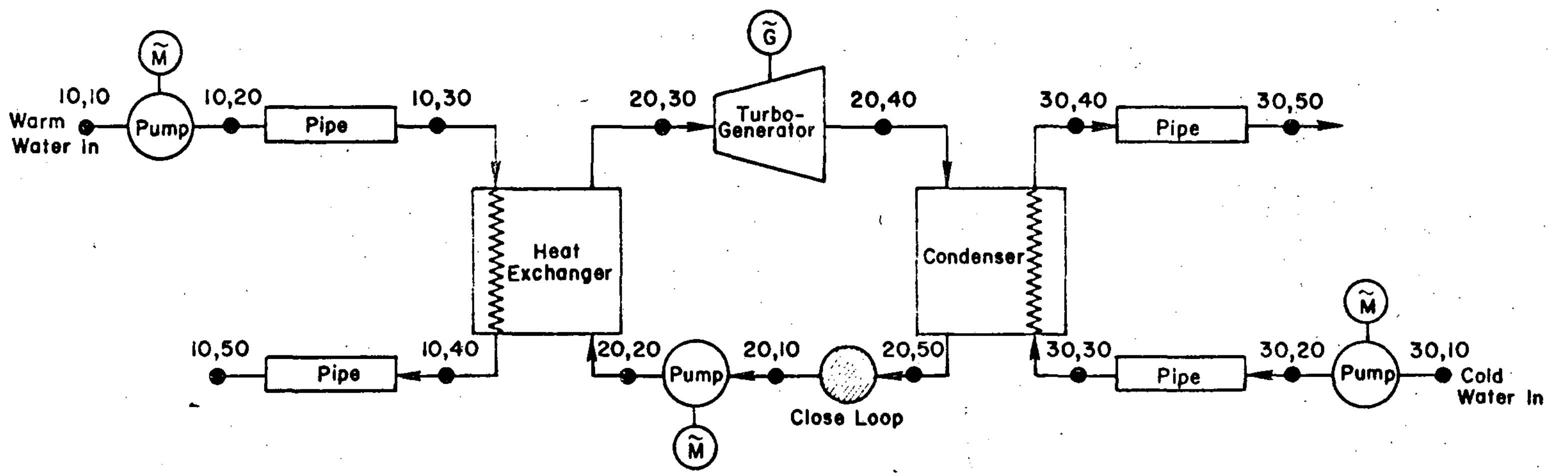
5. The total cost of the facility which includes the wessel that supports the plant is assumed to be 2.5 times the cost of the major plant conversion components. A direct plant cost factor of 1.70 is assumed. The annual maintenance cost is assumed to be 10 percent of the plant capital cost. The annual plant cost (this includes, taxes, insurance, interest on the capital expenditure and profit) is 25 percent of the plant capital cost.

6. The plant is assumed to be operating 85 percent of the time.

The pressure drop in the sea water transport pipes becomes an important optimizable parameter in an ocean thermal cycles. The cycle shown in Figure 5 was optimized with six optimizable parameters. They were: 1) warm sea water piping pressure drop; 2) turbine inlet temperature; 3) condenser temperature; 4) hot water to isobutane heat exchanger pinch point temperature difference; 5) cold water to isobutane heat exchanger pinch point temperature difference; and 6) cold sea water piping pressure drop. It is important to point out that there are other optimizable parameters such as the pressure drops in the heat exchanger which are not included. The optimized cycle shown in Figure 5 is only a *first cut* at ocean thermal cycle optimization.

Table 2 shows the effect of the inlet temperature on the optimization of a ocean thermal cycle. The assumed heat exchanger U factors were 1514 $Wm^{-2}K^{-1}$. The assumed heat exchanger cost was \$18.3 m^{-2}. Table 3 shows the effect of the U factor on the cost of energy from an optimized power cycle. A factor of two increase in the heat transferred per dollar has an effect on the cost of energy and the optimizable parameters of the plant.

In Table 2 it is interesting to compare the bus bar energy cost, plant yield, and cycle efficiency with the corresponding columns on the lefthand side of Table 1. The plant yield per unit mass of water processed is equivalent in the two systems. While the resource warm water in



SIMPLE OCEAN THERMAL CYCLE

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Figure 5. A Simple Ocean Thermal Cycle.

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Table 2. The Effect of Inlet Sea Water Temperature on the Optimization of an Ocean Thermal Cycle for Minimum Cost Energy.

PARAMETERS	INLET WATER TEMPERATURE (°C)	
	20	25
General Parameters		
Bus Bar Energy Los (m$/kWh)	226.7	105.1
Plant Yield (kWh/ton)*	0.0212	0.0428
Cycle Efficiency (%)	0.987	1.81
Warm Sea Water Flow (kgs^{-1})	653874.	324157.
Gross Turbine Power (MW)	84.39	69.85
Heat Exchanger Area (m^2)	2669006.	1046979.
Plant Capital Cost ($/kW)	4822.	2235.
Optimizable Parameters		
Turbine Inlet Temperature (°C)	15.75	19.99
Condenser Temperature (°C)	8.61	9.70
Hot Water Heat Exchanger Pinch Point - ΔT (°C)	1.34	2.11
Condenser Pinch Point - ΔT (°C)	1.51	2.40
Hot Water Pipe Pressure Drop (bar)	0.011	0.041
Cold Water Pipe Pressure Drop (bar)	0.017	0.017

* kWh per metric ton of warm sea water processed

Plant Net Electric Power	50 MW
Cold Water Inlet Temperature	5 °C
Heat Exchanger Cost	\$18.3 m^{-2}
Heat Exchanger U Factor	1514 $Wm^{-2}K^{-1}$
Condenser U Factor	1514 $Wm^{-2}K^{-1}$

Table 3. The Effect of Heat Exchanger U Factor on Cost Optimized Ocean Thermal Cycle Parameters and Energy Cost at the Bus Bar.

PARAMETERS	Case 1	Case 2
Heat Exchanger Parameters		
U Factor ($Wm^{-2}K^{-1}$)	1514.	3028.
Cost per Unit Area ($\$m^{-2}$)	18.3	18.3
Normalized Heat Transfer Cost ($W\$^{-1}K^{-1}$)	82.73	165.46
General Parameters		
Bus Bar Energy Cost (m$/kWh)	105.1	66.5
Plant Yield (kWh/ton)*	0.0428	0.0479
Cycle Efficiency (%)	1.810	1.856
Warm Sea Water Flow (kgs^{-1})	324157.	289714.
Gross Turbine Power (MW)	69.85	68.12
Heat Exchanger Area (m^2)	1046979.	532724.
Plant Capital Cost ($/kW)	2235.	1414.
Optimizable Parameters		
Turbine Inlet Temperature (°C)	19.99	19.99
Condenser Temperature (°C)	9.70	9.70
Hot Water Heat Exchanger Pinch Point ΔT (°C)	2.11	1.94
Condenser Pinch Point ΔT (°C)	2.40	2.15
Hot Water Pipe Pressure Drop (bar)	0.041	0.038
Cold Water Pipe Pressure Drop (bar)	0.017	0.016

* kWh per metric ton of warm sea water produced

Net Electric Power	50 MW
Cold Water Inlet Temperature	5 °C
Warm Water Inlet Temperature	25 °C

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the ocean thermal cycle costs nothing, the cost of processing the water, which is reflected in the cost of energy, is not negligible. If ocean thermal cycles are to ever be economically viable, it is clear that mulitiparameter optimization of the thermodynamic cycles will be important.

COMBINED GAS-TURBINE AND ORGANIC WORKING FLUID - 50 MW PLANT

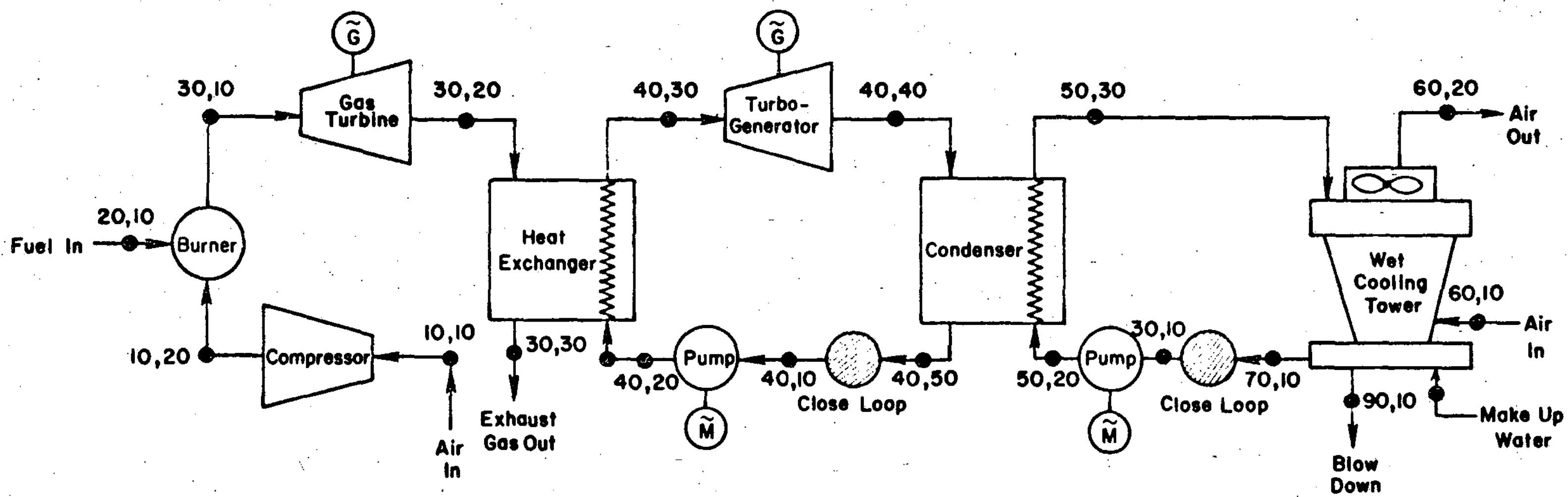
The cycle shown in Figure 6 consists of a gas turbine cycle with an isobutane bottoming cycle. This kind of cycle, which has a high thermodynamic efficiency, could run off of small sources of gaseous or liquid fuel generated by processing agricultural or human waste (for example, sewer gas). The plant represented by this kind of cycle would have a low capital cost. The plant could be quite compact and thus suitable for use on low heating value gasses or liquid and gaseous products resulting from waste processing.

The cycle shown in Figure 6 has seven major optimizable parameters. They are: 1) the compressor exit pressure; 2) the heat exchanger pinch point temperature difference; 3) the isobutane turbine inlet pressure; 4) the isobutane turbine inlet temperature; 5) the condensing pressure in the isobutane cycle; 6) the condenser pich point temperature difference, and 7) the exit temperature of the water from the cooling tower.

Table 4 compares the parameters of a gas turbine system with and without an isobutane bottoming cycle, Two fuel costs are used in both cases. The fuel is expensive (equivalent to $13 and $78 per barrel of oil). The Table shows that the bottoming cycle is potentially worthwhile. The optimizer increases the capital cost of the plant in order to save expensive fuel. It should be noted at GEOTHM can calculate gas turbine cycles with a regenerative preheater and with cycles which use other working fluids in the bottoming loop. The optimizer will optimize the cycles for minimum cost while looking at all of the optimizable parameters simultaneously.

CONCLUSIONS

The report demonstrates how the LBL GEOTHM computer program can be used to design and optimize various types of thermodynamic power cycles. The cycles shown here only illustrate the potential of the program. The optimizer can be used to maximize the power output from power plant cycles already designed and built. It can be extended to other types of cycles such as refrigeration cycles and power cycles which derive heat from other sources. Multiparameter optimization techniques can be used to make some kinds of alternative energy sources economically viable.



GAS TURBINE WITH BOTTOMING CYCLE

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Figure 6. A Gas Turbine Cycle with a Light Hydrocarbon Rankin Bottoming Cycle.

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Table 4. The Effects of Fuel Lost on Cycle Parameters on a Cost Optimized Gas Turbine Cycle With and Without a Bottoming Cycle.

PARAMETERS	FUEL COST ($ per kg)			
	Without Bottoming Cycle		With Bottoming Cycle	
	0.10	0.60	0.10	0.60
General Plant Parameters				
Bus Bar Energy Cost (m$/kWh)	41.1	165.6	39.6	152.3
Plant Yield (kWh per ton)*	3798.	4083.	4182.	4522.
Cycle Efficiency (%)	34.195	36.747	37.641	40.702
Fuel Mass Flow (kg, s^{-1})	3.657	3.402	3.321	3.071
Gross Turbine Power (MW)	108.26	133.25	101.85	124.48
Plant Capital Cost ($/kW)	380.	478.	404.	503.
Optimizable Parameters				
Compressor Exit Pressure (bar)	10.164	17.208	9.159	15.849
Isobutane Turbine Inlet Pressure (bar)	DNA	DNA	128.245	149.829
Isobutane Turbine Inlet Pressure (°C)	DNA	DNA	383.61	326.88
Condenser Pressure (bar)	DNA	DNA	6.198	6.500
Heat Exchanger Pinch Point ΔT (°C)	DNA	DNA	6.66	1.78
Condenser Pinch Point ΔT (°C)	DNA	DNA	2.58	1.81
Cooling Tower Water Temperature (°C)	DNA	DNA	33.20	31.29

* kWh per metric Ton of Fuel Burned; DNA = Does Not Apply.

Net Electric Power	50 MW	Heat Exchanger U Factor	100 $Wm^{-2}K^{-1}$
Gas Turbine Inlet Temperature	1250 °C	Condenser U Factor	~400 $Wm^{-2}K^{-1}$
Air Dry Bulb Temperature	48.89 °C		
Air Wet Bulb Temperature	26.67 °C		

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