

THE HAWAII GEOTHERMAL PROJECT

REGENERATIVE VAPOR CYCLE
WITH ISOBUTANE AS WORKING FLUID

TECHNICAL REPORT No. 4
June 10, 1974



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ENGINEERING PROGRAM

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I. INTRODUCTION

In New Zealand and Mexico, vapor flashing plants have been successfully used to produce power from geothermal fluids emitted from self-flowing wells in liquid-dominated fields. The term self-flowing means that a mixture of hot water and flash steam emerges from the well by natural forces. In a vapor flashing plant, the vapor separated from brine expands in a turbine for power generation and then condenses in a direct-contact condenser. Another type of plant which may be used in a liquid-dominated field is known as the binary plant. The word binary is used because two fluids are involved in the power production process, the geothermal fluid and the working fluid. Using Freon as a working fluid, small binary power plants have been constructed in Japan and Russia to harness geothermal energy. Recently, San Diego Gas and Electric Company has experimented with isobutane as a working fluid to produce power from hot brine with a high concentration of salts. In a basic binary cycle, the working fluid extracts heat from geothermal fluid and operates in a Rankine cycle to produce power. Owing to the properties of Freon or isobutane, the cycle operates at supercritical pressure in the temperature range of 300°F to 500°F, and the Rankine cycle efficiency can be satisfactory. One of the advantageous features of the Freon or isobutane systems is that the construction of the turbine can be very compact and simple and the turbine efficiency can be higher than the efficiency of steam turbine of the same rating.

This report contains the investigation of a proposed modification of the basic isobutane cycle with a regenerative heat exchanger. The performance of the regenerative cycle appears very favorable as compared with that of the basic cycle. Although the system is complicated by the addition of a regenerative

heat exchanger, the increase in power production and the reduction of the sizes of cooling equipment could justify the extra cost of the regenerative heat exchanger.

Geothermal fluid from a well may exist in two phases, vapor and liquid, if the fluid is self-flowing. To avoid scaling, a submersible pump may be used to force the fluid out of the well. Effects of pumping on the power output and heat rejection of a binary cycle have also been evaluated with a specific set of given conditions, and the calculated results are included in this report.

II. DESCRIPTION OF CYCLE

Geothermal fluids from wells consist of steam, liquid water, and various impurities in a wide range of concentrations, temperatures, and pressures. Because of the non-uniformity of the fluids, the geothermal power plants differ from one locality to another. Since the energy stored in liquid-dominated fields of the world is believed to be many times larger than in vapor-dominated fields, there is considerable interest in the design of power plants for utilization of heat from hot brine at 350°F to 500°F. A recent effort has been focused on using isobutane as the working fluid to operate in a closed cycle. The expected performances and the advantages of such a cycle have been discussed in detail by Anderson [1] and Holt, et al.[2]. In a basic cycle shown in Fig. 1, geothermal fluid from wells is used to vaporize and to superheat isobutane in a heat exchanger. Isobutane vapor then expands through a turbine to generate power, and the exhaust vapor condenses in a water-cooled condenser before it enters the heat exchanger to complete a cycle. Anderson pointed out many advantages of the isobutane cycle over the flashed-steam cycle, which has been applied to the geothermal plants in New Zealand for decades. Holt, et al. evaluated the effects of working pressures of isobutane on the thermal efficiency of the basic cycle and found that the optimal working pressure largely depends on the temperature of available geothermal fluid. The optimal working pressure increases with the increase of brine temperature, and the isobutane vapor from the turbine has to be at highly superheated conditions in order to achieve high thermal efficiency.

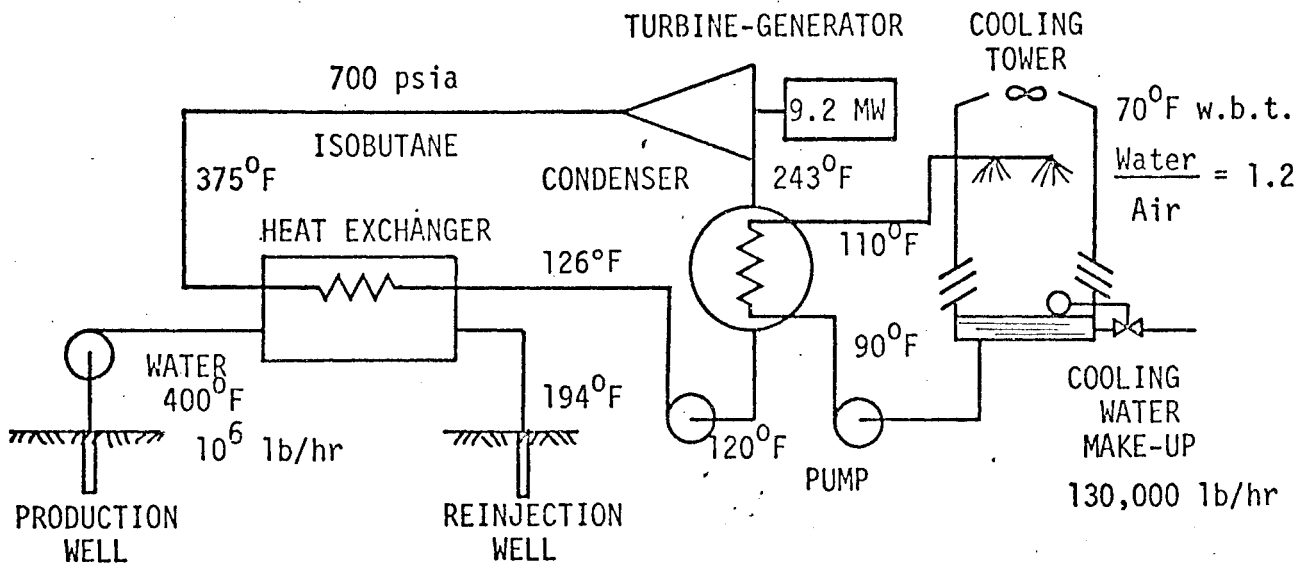


FIG. 1 BASIC ISOBUTANE CYCLE

To improve the thermal efficiency of a simple gas turbine cycle, a regenerator may be used to allow the interchange of energy between the turbine exhaust and the compressed air entering the combustion chamber. This principle can be applied profitably to the basic isobutane cycle for a geothermal power plant as depicted in Fig. 2. The superheated exhaust vapor from the turbine is passed through a regenerator to heat the condensate before it enters the main heat exchanger. By adding the regenerator to the system, the sizes of the condenser, cooling-water pump, cooling tower, and main heat exchanger can be significantly reduced. Furthermore, the discharge temperature of brine of the regenerative cycle is noticeably higher than that of the basic cycle; thus the waste heat can be used for industrial uses, such as producing fresh water with a multiple-effect evaporator as illustrated in the flow diagram.

The values of temperature, pressure, and flow rate in Figs. 1 and 2 refer to a sample calculation under the following assumptions:

- a. The efficiency of isobutane turbine-generator is 85%, and the efficiency of feed pump is 80%.
- b. One million pounds of hot water flows into the power plant at 400°F with heat content of 375 BTU per lb.
- c. The minimum temperature difference between the hot water and the isobutane is taken to be 10°F in a counter-flow heat exchanger. For example, the temperature differences between the fluids are shown in Fig. 3 for a turbine throttle pressure of 700 psia.
- d. The condensing temperature of isobutane is 120°F.
- e. The terminal temperature difference of isobutane in the regenerator is 10°F. Pressure drops and heat losses of the fluids through pipes and heat exchangers are neglected.

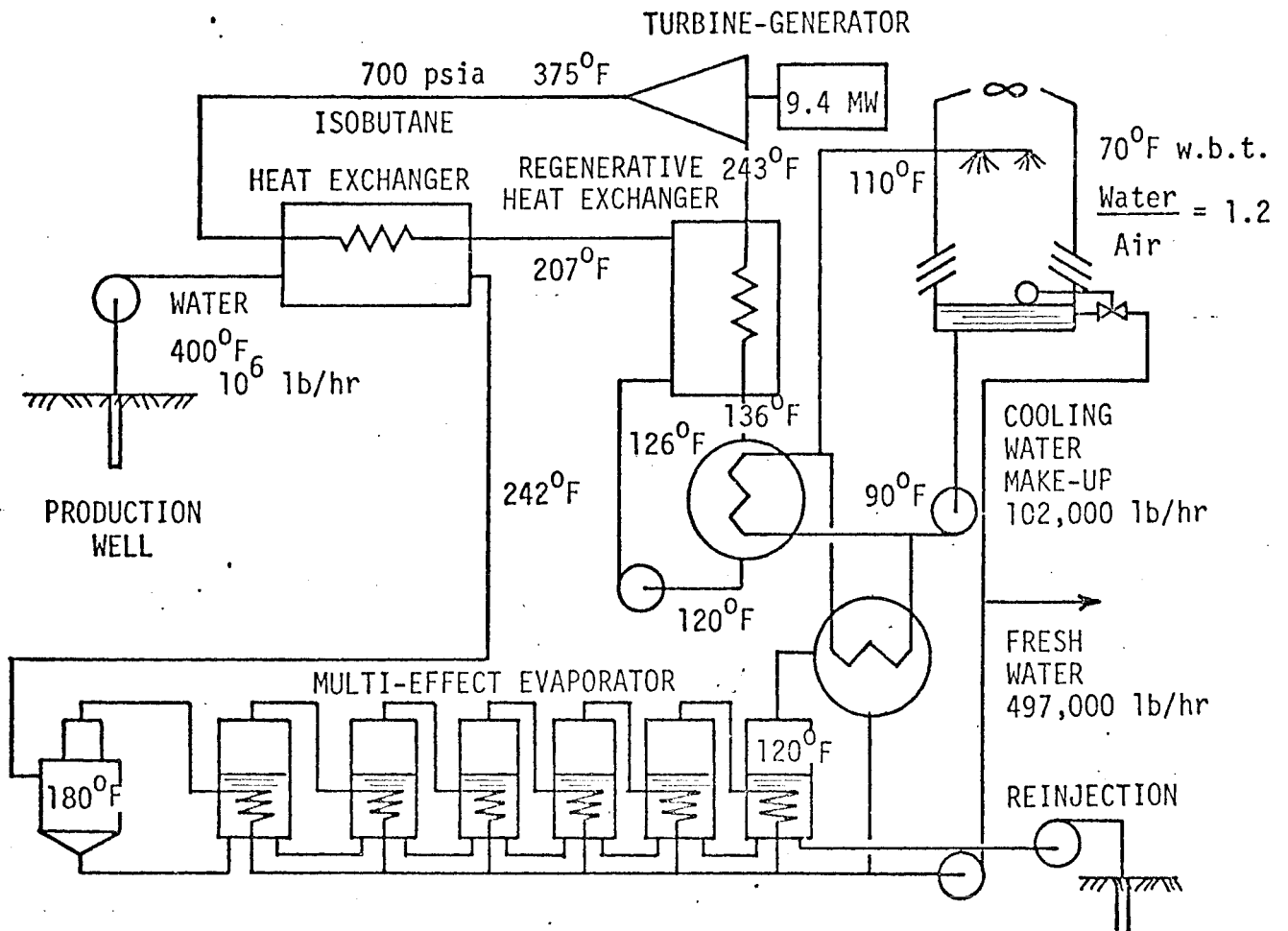


FIG. 2 REGENERATIVE ISOBUTANE CYCLE AND MULTI-EFFECT EVAPORATOR

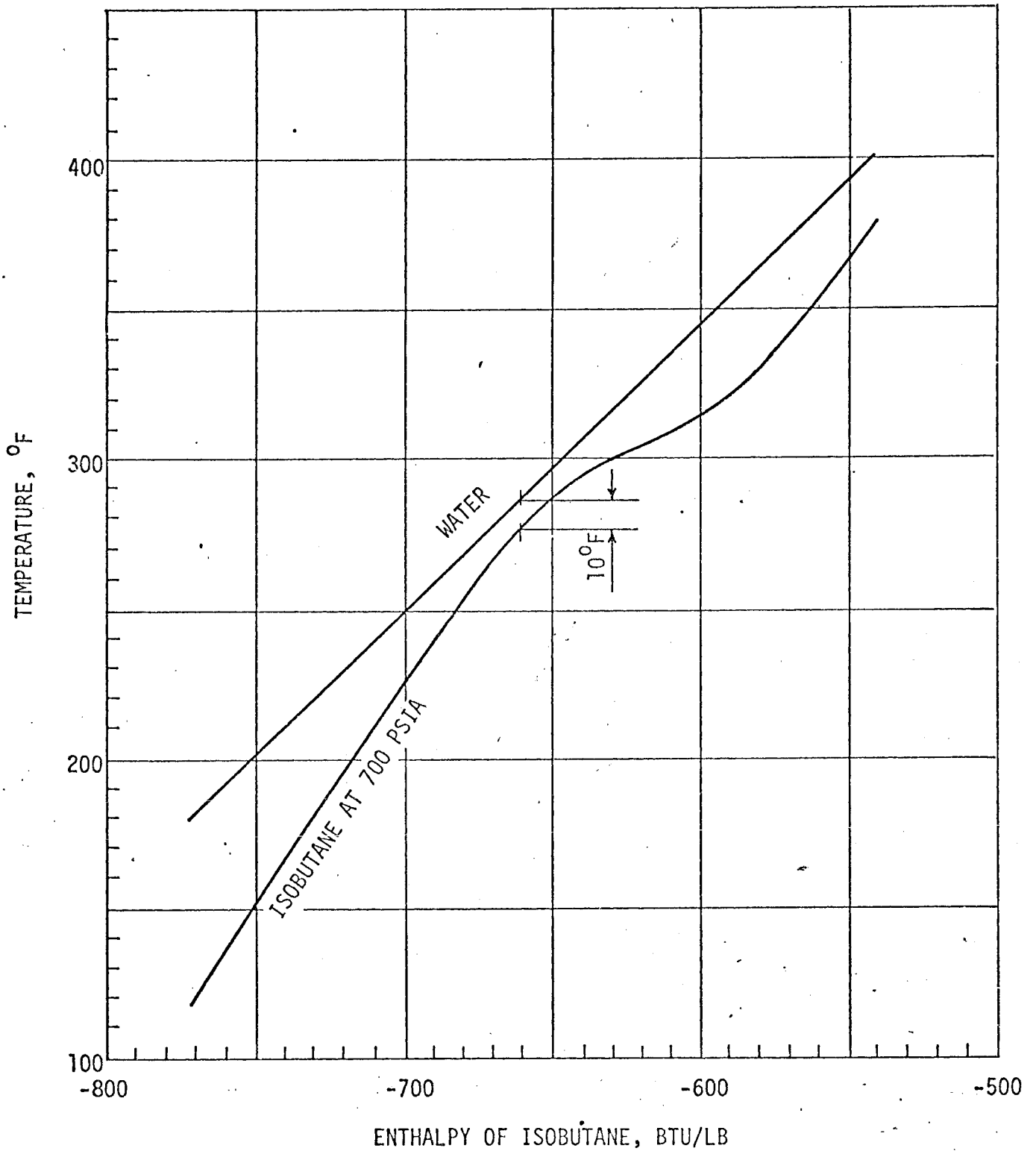


FIG. 3 TYPICAL TEMPERATURE DIFFERENCES BETWEEN WATER AND ISOBUTANE IN COUNTERFLOW HEAT EXCHANGER

To illustrate the significance of the results, power output and heat rejection are plotted in Figs. 4 and 5. In this case, the maximum increase in power production by using a regenerator is only in the order of 2% of the power produced by the basic cycle at the optimal operating pressure of 700 psia. However, the rates of heat which must be rejected through the cooling system are greatly different between the two cycles. Moreover, the temperature of water leaving the main heat exchanger of the regenerative cycle is much higher than that of the basic cycle as indicated in Figs. 1 and 2. Thus the waste heat can be economically extracted from the discharge of hot water for useful purposes. With a six-effect evaporator, there is 497,000 lbs/hr of fresh water to be exported in this example.

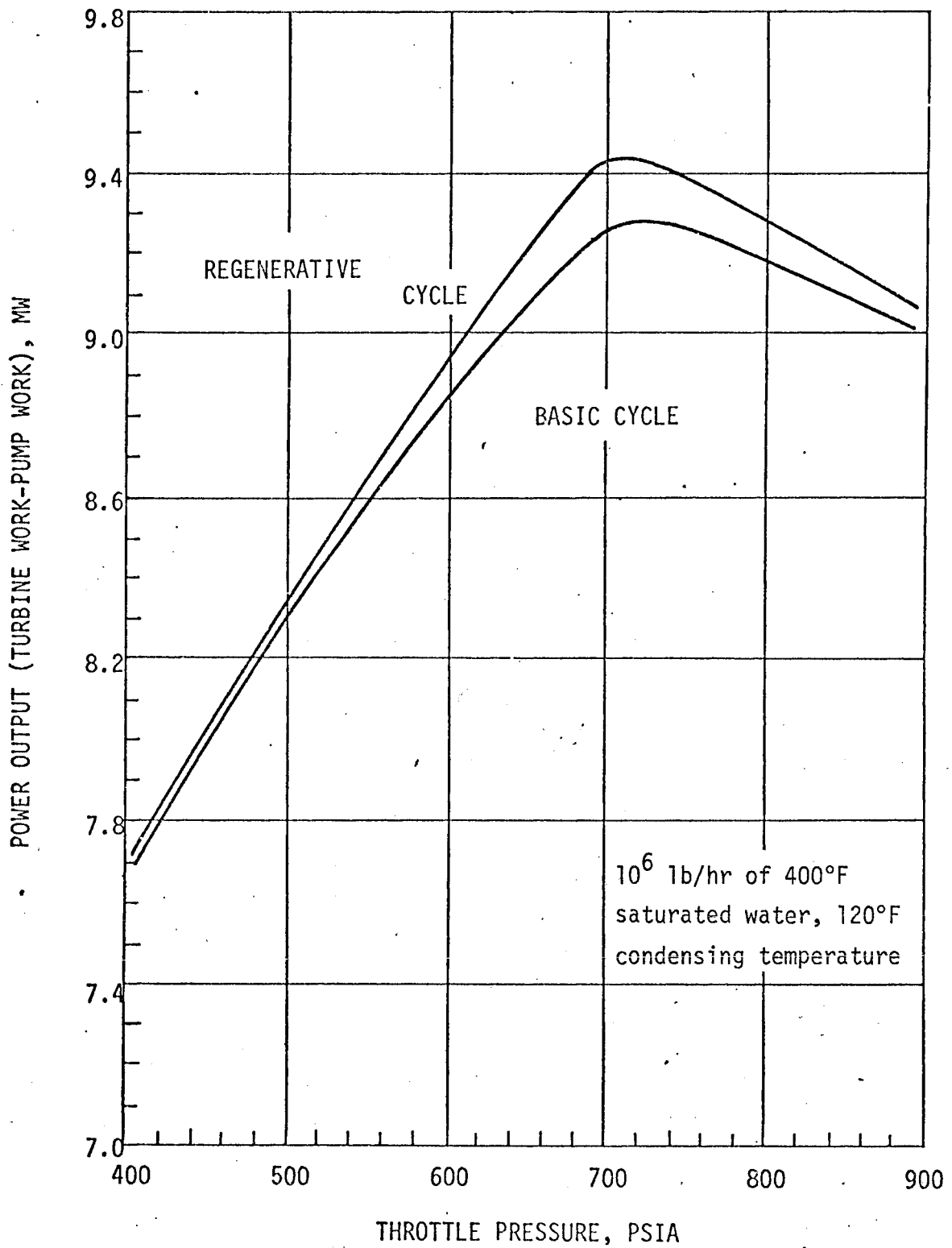


FIG. 4 POWER OUTPUT WITH 400°F SATURATED WATER

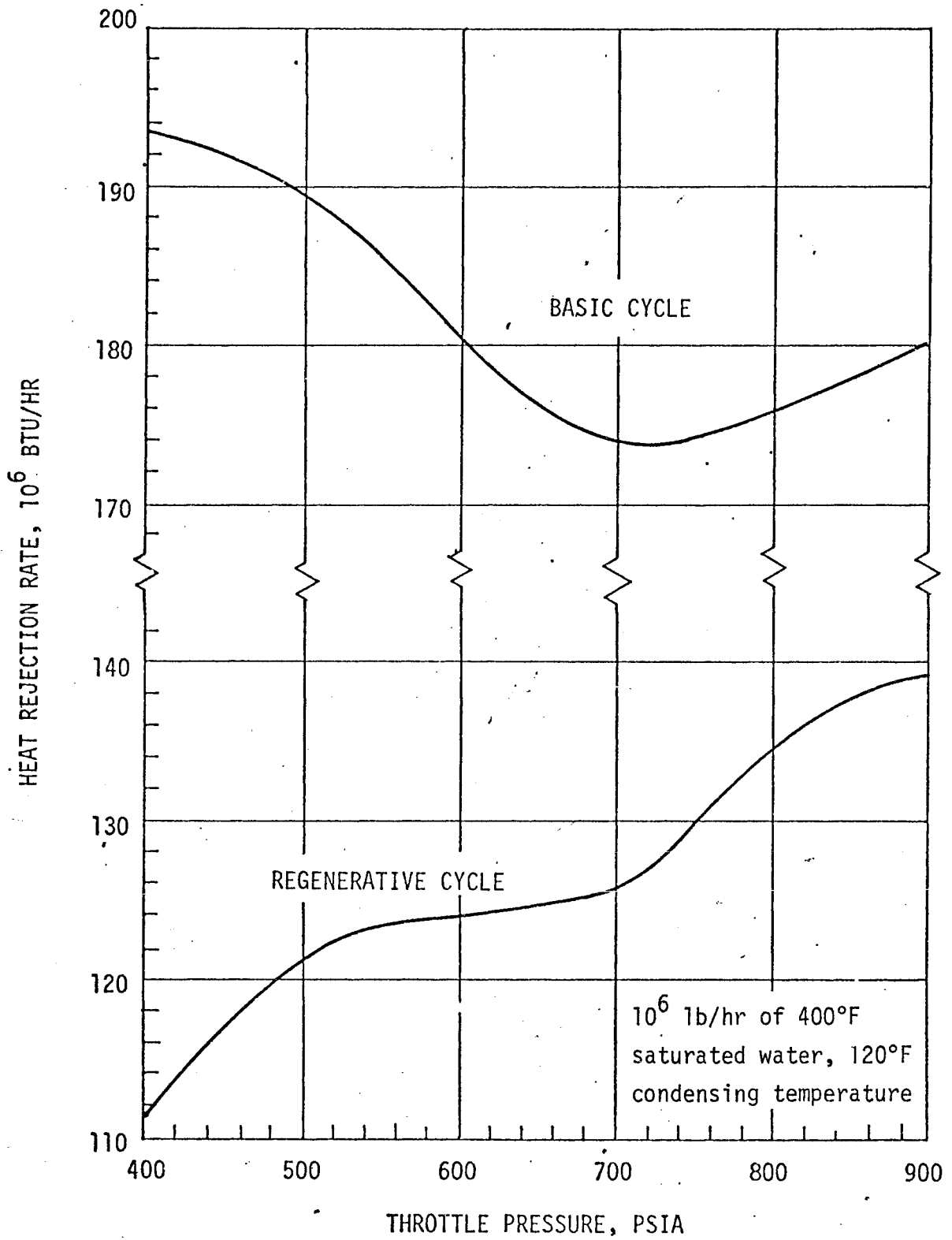


FIG. 5 HEAT REJECTION RATE WITH 400°F SATURATED WATER

III. USING LIQUID-VAPOR MIXTURE OF GEOTHERMAL FLUID

In a liquid dominated field, the geothermal fluid may be delivered from a well by flashing or by pumping. The temperature of the fluid inside the reservoir is nearly saturated, and the corresponding vapor pressure is much higher than the atmospheric pressure. If the fluid is allowed to flow freely through the well, a part of the liquid will become vapor as the fluid loses some of its vapor pressure. This loss of vapor pressure is due to the pressure drop caused by friction and to the change of potential energy on its way moving up. The amount of vapor separated depends upon the corresponding pressure drop. This process is called flashing. With a self-flowing well, the geothermal fluid appears in two phases at the wellhead. Consequently, the isobutane in a binary system should be heated in two stages, first by the liquid and then by the flashed vapor. Fig. 6 shows the arrangement of the regenerative cycle with the isobutane heated in two stages. The flashed vapor loses all of its heat of vaporization in the first heat exchanger and becomes saturated liquid having the same heat content as the liquid directly from the separator, since the pressure drop through the heat exchanger is assumed to be negligible. The two streams of liquid join together to enter the second heat exchanger where they release their heat.

To study the performance of a regenerative cycle as compared with that of a basic cycle, the following conditions were assumed for the calculations of power output and heat rejection:

Rate of flow	10^6 lb _m /hr,
Temperature of water	400°F,
Composition of water by weight	20% vapor and 80% liquid,
Turbine efficiency	85%,
Feed pump efficiency	80%,
Minimum temperature difference in heat exchanger	10°F,
Condensing temperature	120°F.

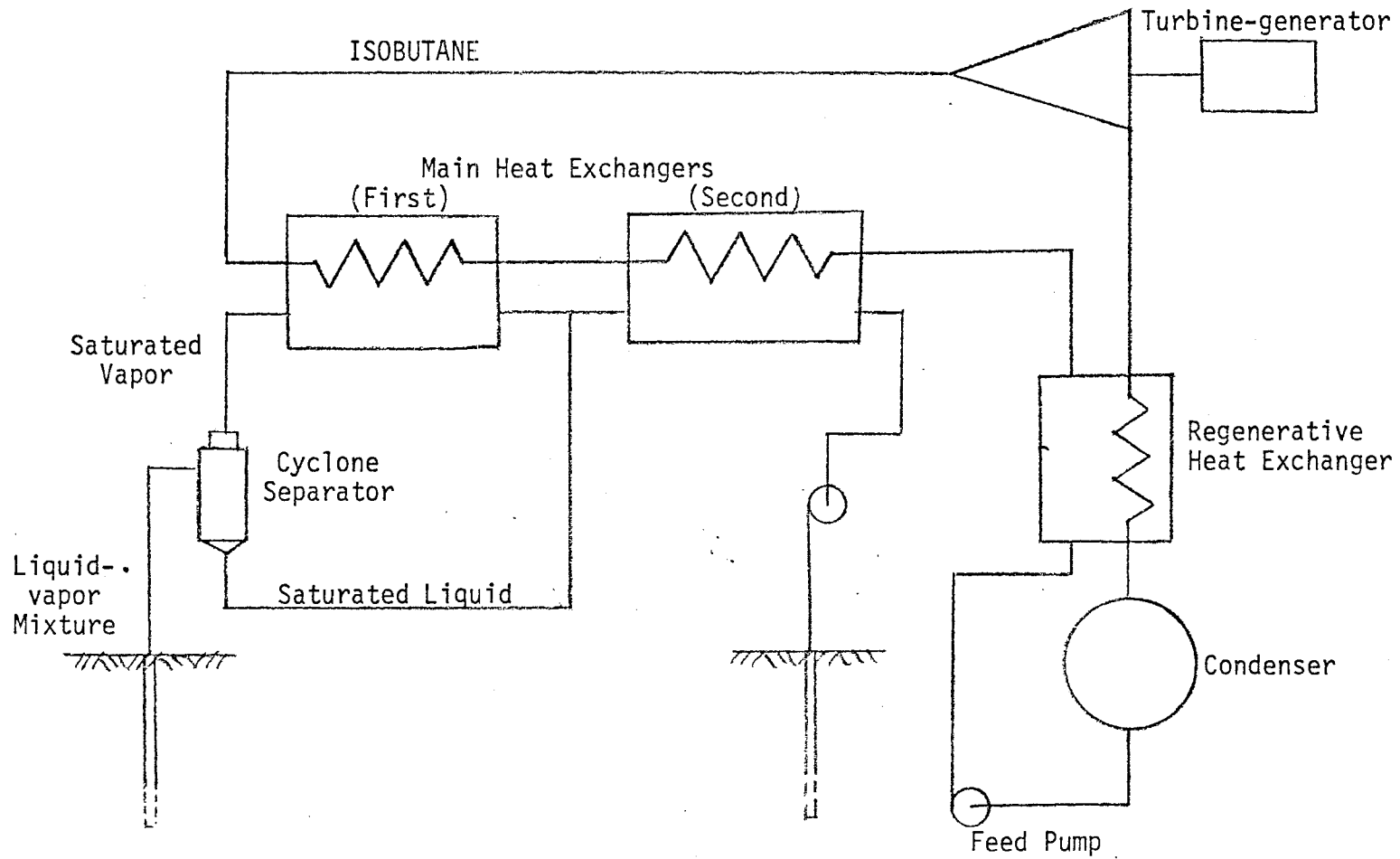


FIG. 6 REGENERATIVE CYCLE WITH TWO-STAGE HEATING

At the start of calculation, the temperature-enthalpy relationship between the two-phase water and the isobutane must be ascertained, as shown in Fig. 7, which applies to the operating pressure of 700 psia only. The temperature-enthalpy line of isobutane was plotted according to the data from Handbook of Fundamentals by the American Society of Heating, Refrigerating and Air-Conditioning Engineers [3]. Unlike isobutane, liquid water has a nearly linear relationship between temperature and enthalpy. The slope of the temperature-enthalpy line of water depends upon the ratio of water flow to isobutane flow. The dotted horizontal line depicts that the steam is losing its heat of vaporization to the isobutane in the first heat exchanger at constant temperature. The sloped line, which follows the horizontal line, represents the temperature drop of water in the second heat exchanger. Since there is a considerable portion of vapor in the vapor-liquid mixture, the 10°F minimal temperature difference occurs at the exit of the heat exchanger.

The performances of the basic cycle and the regenerative cycle under the aforementioned assumptions have been calculated and tabulated in Tables 1 and 2, respectively. In the second columns of the tables are the optimum throttle temperatures. For every working pressure, there is an optimum throttle temperature determined by the method of exhaustive search, which gives the highest power output. For example, at 700 psia, the optimum throttle temperature is 375°F and the maximum power output is 19.535 MW, which is equal to the turbine power minus feed-pump power. In general, the power output of a regenerative cycle is higher than that of a basic cycle, as illustrated in Fig. 8. As to the rates of heat rejection, it is also in favor of the regenerative cycle. At the optimum working pressure of 700 psia, the temperature of waste water leaving the regenerative system is 217°F, which is in the proper range for industrial uses, while the temperature of waste water from the basic cycle is too low for any economical recovery of waste heat.

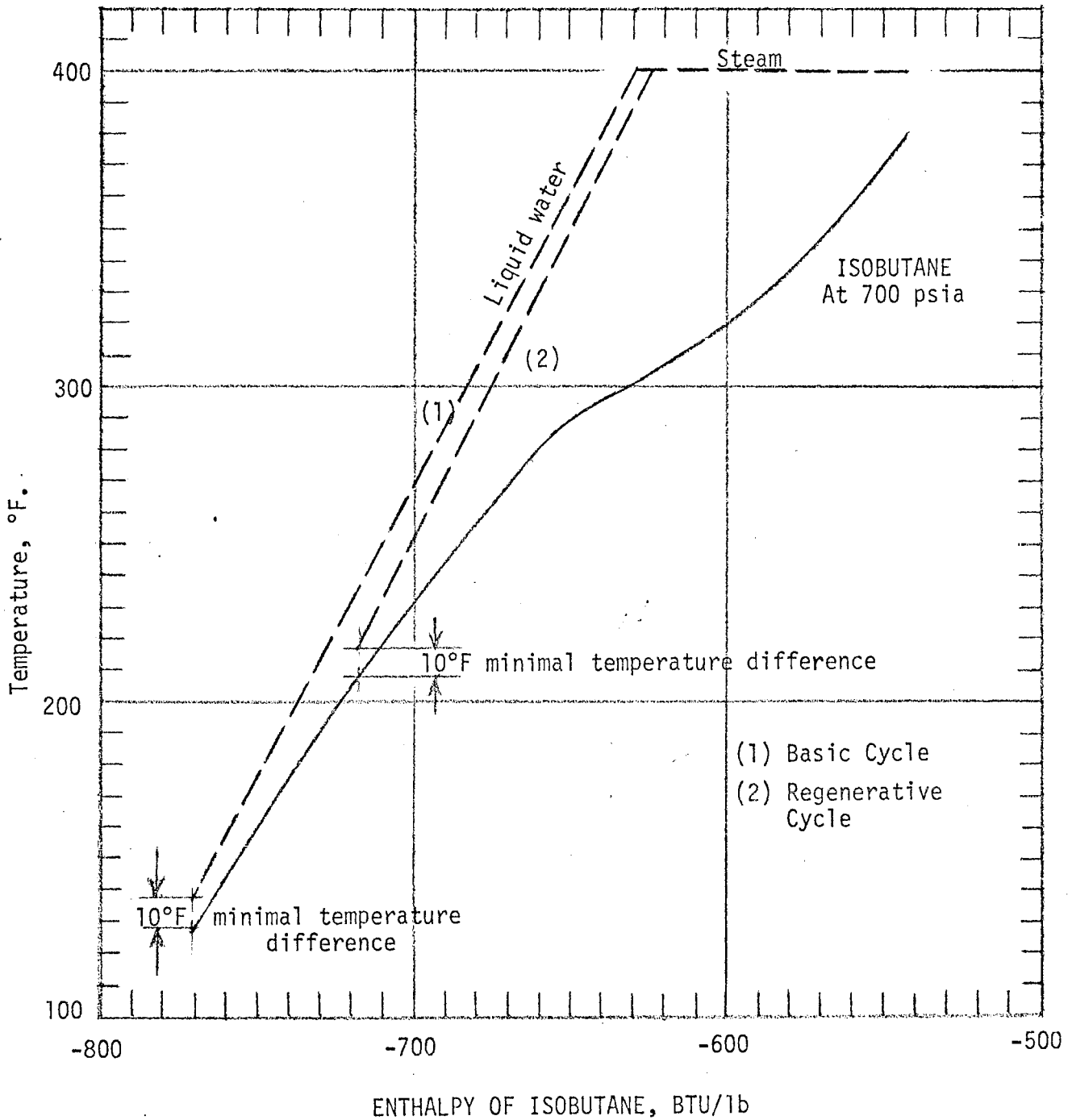


FIG. 7 TEMPERATURE-ENTHALPY DIAGRAM FOR ISOBUTANE AT 700 PSIA WITH TWO-STAGE HEATING

TABLE 1 PERFORMANCE OF BASIC CYCLE WITH 400°F WATER (20% VAPOR, 80% LIQUID)

WORKING PRESSURE psia	OPTIMUM THROTTLE TEMP., °F	ISOBUTANE FLOW RATE 10 ⁶ lb/hr	POWER OUTPUT MW	HEAT REJECTION 10 ⁶ BTU/hr	WATER LEAVING TEMP., °F
500	380	1.810	16.33	381.22	135
600	390	1.901	17.70	376.49	135
700	375	1.948	19.535	369.12	136
800	375	2.028	18.398	371.09	138
900	375	2.065	18.443	369.93	139

For 10⁶ lb/hr of water (20% vapor, 80% liquid) at 400°F; 120°F condensing temperature.

TABLE 2 PERFORMANCE OF REGENERATIVE CYCLE WITH 400°F WATER (20% vapor, 80% liquid)

WORKING PRESSURE psia	OPTIMUM THROTTLE TEMP., °F	ISOBUTANE FLOW RATE 10 ⁶ lb/hr	POWER OUTPUT MW	HEAT REJECTION 10 ⁶ BTU/hr	WATER LEAVING TEMP., °F
500	380	1.996	18.013	264.87	245
600	390	2.076	19.33	275.52	230
700	375	2.101	21.07	283.01	217
800	375	2.183	19.819	294.02	210
900	380	2.223	19.862	303.94	200

For 10⁶ lb/hr of water (20% vapor, 80% liquid) at 400°F; 120°F condensing temperature.

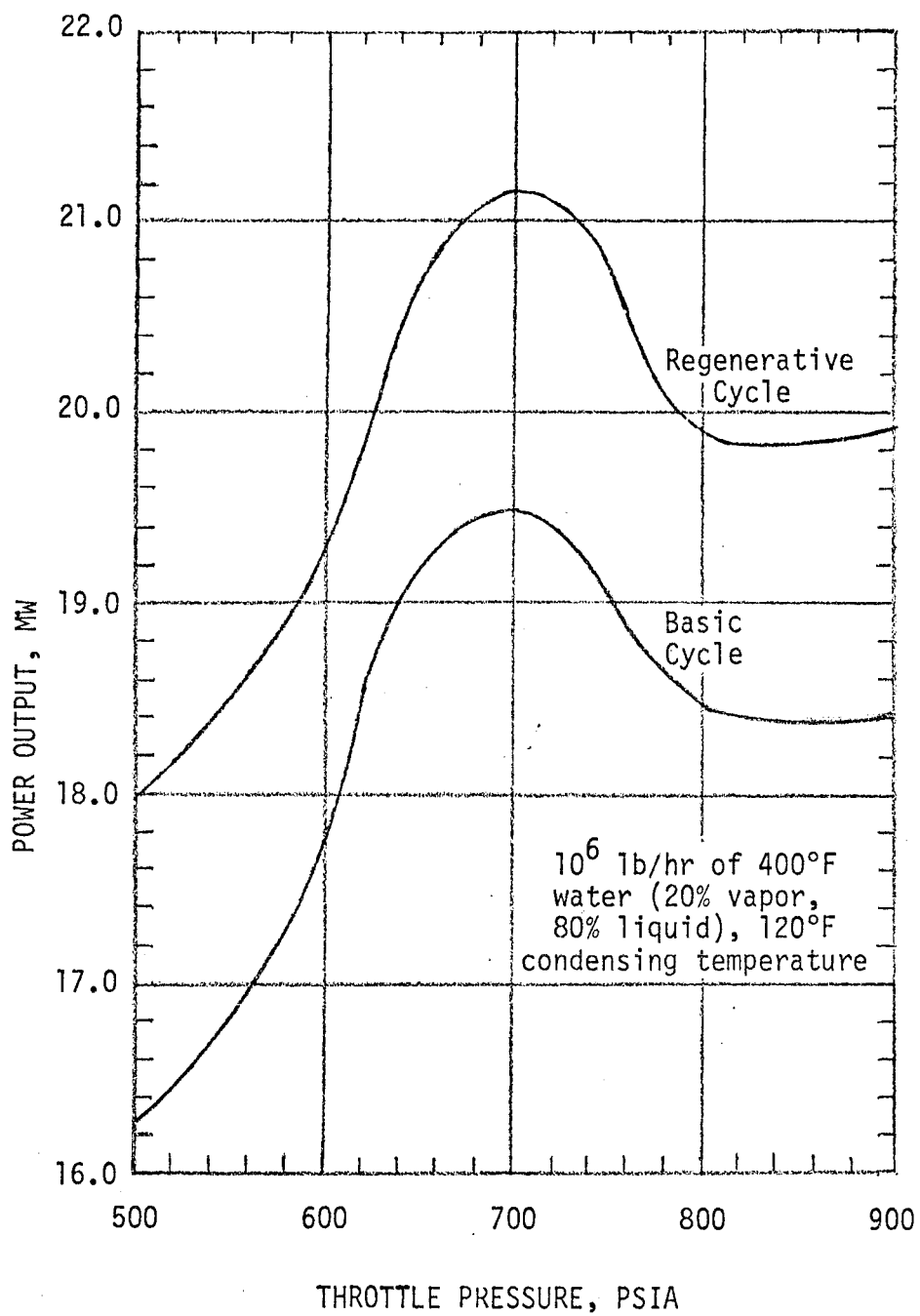


FIG. 8 POWER OUTPUT WITH 400°F WATER
(20% VAPOR, 80% LIQUID)

IV. USING GEOTHERMAL LIQUID ONLY

When a geothermal fluid flows through a well by natural forces, some of the liquid would become vapor, and some of the dissolved solids in the liquid may precipitate during the flashing to cause scaling on the inside surface of the well. If the content of solids is high, a pump must be used to force the liquid out to prevent it from flashing. When a pump is used, there is only liquid entering the main heat exchanger.

In the preceding section, the performances of two cycles are reported for geothermal fluid which flashes out from a well and consists of 20% vapor and 80% liquid water by weight at 400°F when the mixture enters the main heat exchanger. It should be of interest to determine the effects on the performance by pumping the water out. The enthalpy of the 400°F vapor-liquid mixture is 540.48 BTU/lb. The temperature and vapor pressure of a saturated liquid water, having the equivalent value of the enthalpy of 540.48 BTU/lb, are 543.24°F and 987.8 psia respectively. Assuming that the water is delivered to the main heat exchanger at 543.24°F as a saturated liquid by pumping, the throttle temperature of isobutane was determined by plotting the temperature-enthalpy diagram which was similar in shape to Fig. 3. The minimal temperature difference between the two fluids was also 10°F. For working pressures from 600 to 1000 psia, the optimum throttle temperature of isobutane is virtually at the same point, 520°F. The calculated values of power output and heat rejection are given in Tables 3 and 4 for a basic cycle and a regenerative cycle, respectively. The gain in power output is graphically illustrated in Fig. 9. To compare the curves with those in Fig. 8, obviously there is no advantage in power production by pumping the fluid from a well to keep it in a liquid state. However, the handling of well-water by pumping can prevent the flashing of water in the well and reduce the pressure

TABLE 3 PERFORMANCE OF BASIC CYCLE WITH 543°F LIQUID WATER

WORKING PRESSURE psia	THROTTLE TEMP. °F	ISOBUTANE FLOW RATE $10^6 \text{ lb}_m/\text{hr}$	POWER OUTPUT MW	HEAT REJECTION 10^6 BTU/hr	WATER LEAVING TEMP., °F
600	520	1.24	14.779	362.09	160
700	520	1.25	15.700	355.37	163
800	520	1.24	16.752	345.35	170
900	520	1.26	17.400	345.14	168
1000	520	1.27	17.505	338.77	174

120°F condensing temperature, 10^6 lb/hr water.

TABLE 4 PERFORMANCE OF REGENERATIVE CYCLE WITH 543°F LIQUID WATER

WORKING PRESSURE psia	THROTTLE TEMP. °F	ISOBUTANE FLOW RATE 10^6 lb _m /hr	- POWER OUTPUT MW	HEAT REJECTION 10^6 BTU/hr	WATER LEAVING TEMP., °F
600	520	1.298	15.448	172.28	344
700	520	1.290	16.224	178.47	335
800	520	1.314	17.519	176.99	332
900	520	1.310	18.036	179.14	329
1000	520	1.326	18.311	181.33	326

120°F condensing temperature, 10^6 lb/hr water.

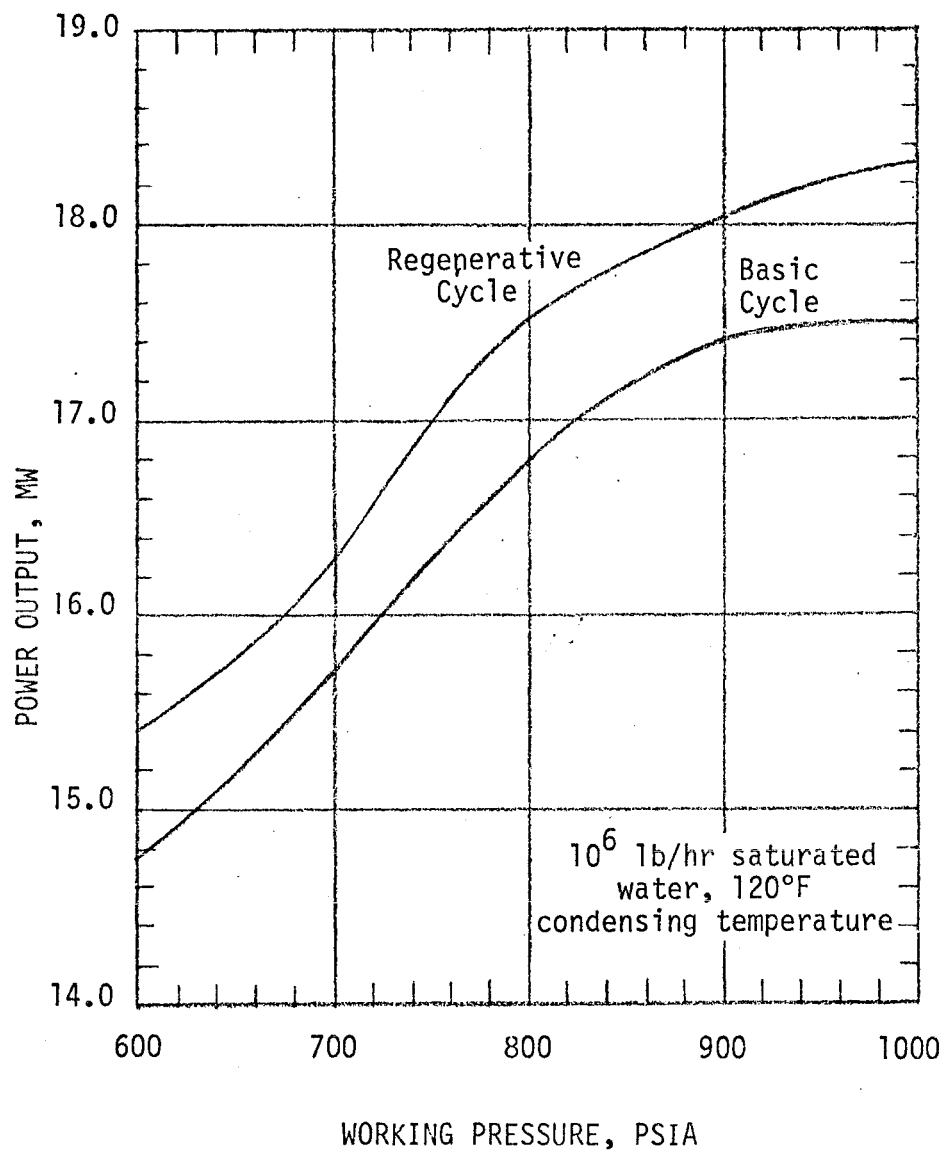


FIG. 9 POWER OUTPUT WITH 543°F SATURATED WATER

difference in the main heat exchanger. The two outstanding features of a regenerative cycle are the low heat rejection to cooling water and the high heat content of the waste brine. This is demonstrated again in the calculated results shown in the last two columns of Table 4, in which the high temperature of water implies the high heat content.

V. CONCLUSION

The selection of the type of power plant depends primarily upon the thermodynamic and chemical characteristics of the fluid in a reservoir in addition to the economic justifications. With improved designs of heat exchangers in the future, the binary plant may become feasible in some geothermal areas. The thermal efficiencies of regenerative binary cycles are higher than those of basic cycles. The main advantage of the regenerative cycle lies in the reduction of heat rejection to cooling water. The basic cycle can easily be converted to a regenerative binary cycle by the addition of a regenerative heat exchanger, and such a conversion can result in about 30% reduction in the sizes of condenser and cooling tower. It is recommended here that a plant designer include the regenerative cycle as one of the alternatives in the process of selecting a plant system, should he be interested in a binary system. The cost of heat rejection equipment may amount to one-third of the total cost of a geothermal power plant. A reduction in the rate of heat rejection by adopting a regenerative binary cycle may significantly lower the capital cost of the plant as well as the unit cost of power generated.

The heating surface of the regenerative heat exchanger is free from scaling and corrosion, which usually cause problems in the main heat exchanger in dealing with brine. Although the convective heat transfer coefficient on the side of super-heated vapor is low, the capital cost of the regenerative heat exchanger can be largely offset by the savings which occur due to the reduction in size of the main heat exchanger; also, the maintenance cost of the regenerative heat exchanger should be lower than that of the main heat exchanger.

The regenerative binary cycle is characterized by the high heat content of the waste brine. The temperatures of water leaving the heat exchanger of the regenerative cycle are much higher than those of the basic cycle in all the cases of the sample calculations. At an elevated temperature of waste brine, the waste heat can be economically extracted for industrial uses. Many geothermal reservoirs are located in areas where the supply of fresh water is inadequate. The regenerative cycle offers a great potential for the simultaneous production of fresh water and electrical power. A possible scheme of combining a six-effect evaporator with a regenerative power plant is given in Fig. 2, to show that fresh water can be produced as a by-product of a geothermal power plant.

If hot brine is pumped out from a well, the problem of scaling could be minimized, and the thickness of the heating surface may be decreased because of the reduction of the pressure difference between the brine and the isobutane. With a regenerative binary system to extract energy from pumped brine, the temperature of waste brine leaving the system is extremely high, and there is great potential in the utilization of this waste heat. However, the cost of pumping must be carefully weighed.

Conditions of reservoirs and requirements of power and water vary from one locality to another. For every case, all the possible alternatives of plant arrangement should be carefully evaluated so that the most feasible solution can be determined. The addition of a regenerative heat exchanger to a closed vapor cycle could be an attractive alternative.

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