HAWAII GEOTHERMAL PROJECT

ENGINEERING PROGRAM

CONCEPTUAL DESIGN OF A 10MW
REGENERATIVE ISOBUTANE
GEOTHERMAL POWER PLANT

TECHNICAL REPORT No. 18

October 15, 1976

By

Ashok K. Gupta

and

James C. S. Chou

Department of Mechanical Engineering
University of Hawaii
Honolulu, Hawaii 96822

TABLE OF CONTENTS

LIST OF TABLES

LIST OF ILLUSTRATIONS

CHAPTER I

OPTIMUM PRESSURE AND TEMPERATURE WITH ISOBUTANE

AS THE WORKING FLUID

Introduction

Binary fluid system may be used for generating power
from geothermal brines. As the name implies, two fluids
are involved in the power production process--the geothermal
fluid and the working fluid. The advantages of isobutane
as the working fluid in a closed cycle have been discussed
by Anderson (15) and Holt, Hutchinson and Cortez (12). In
a basic isobutane cycle, geothermal fluid from the wells
is used to vaporize and to superheat isobutane in a heat
exchanger. Isobutane vapor then expands through a turbine
to generate useful power. The exhaust vapor is condensed
and pumped to a heat exchanger to complete a cycle.

To improve the thermal efficiency of the basic cycle,
a regenerative heat exchanger may be used to transfer
the heat from the turbine exhaust to the compressed isobutane
entering the main heat exchanger. This results in reducing
the size of the condenser, cooling equipment, and the main
heat exchanger. Another effect is the increase of the
temperature of the waste brine so that the waste heat could
be utilized effectively.

The objective of this study is to evaluate the regenerative binary cycle, as applied to the geothermal brines for the generation of power. The selections of the cycle temperature and pressure of isobutane have been studied since they have great effects on the performance of a basic or regenerative plant. The results of the study are reported in this chapter.

The entire system of a binary fluid plant is difficult to optimize because of its complexity. In such a case, we may try to optimize the subsystems and then choose an optimum combination of these subsystems. The most critical subsystem is the main heat exchanger. The type of flow and the temperature distribution of fluids in the heat exchanger affect the size of the heat exchanger and the power output of the system, which can be evidenced by heat balance.

The inlet temperature of the hot geothermal fluid depends upon the reservior temperature and the method of bringing the fluid to the surface. In a liquid dominated field, the geothermal fluid may be delivered from a well by flashing or by pumping. If the fluid is allowed to flow through the well by natural forces, a part of the liquid will become vapor, as the pressure drops when the fluid moves up. In this case, the main heat exchanger may be divided into two sections. In the first section, isobutane is heated by condensing the flashed brine of the hot geothermal fluid, and the heat is transferred from hot

liquid brine to isobutane in the second section. If a
suitable pump can be developed to force the fluid out
through the well, the fluid can remain compressed without
flashing, and there is no need for the condensing section
of the heat exchanger. Two cases are considered separately
in determining the optimum temperatures.

Temperature Distribution in Counterflow Heat Exchanger for the Compressed Geothermal Liquid

While the types of commercial heat exchangers may be
a combination of counterflow, parallel flow, and cross-
flow, pure counterflow heat exchangers are proposed for
extracting energy from geothermal fluids since the
temperature distribution of a counterflow arrangement gives
the highest final temperature which isobutane can attain
when it is heated by the hot fluid. An additional advantage
of the counterflow arrangement is that less heating surface
is required by this arrangement than by parallel flow and
crossflow arrangements (11).

There are two difficulties in heat transfer problems
involving superheated isobutane. One is the lack of precise
data on its thermodynamic properties; another is the
peculiarity of the specific heat of superheated isobutane
vapor. To facilitate the interpolation and to increase the
precision of the calculations, the temperature-enthalpy

relationship of isobutane may be taken from a small size
chart as given in the ASHRAE Handbook (2) and replotted
with much larger scales. Figure 1-1 is the temperature-
enthalpy curve of the superheated vapor at 700 psia. There
are two points of inflection at approximately 280 and 320°F.
The slope at every point of the curve represents the
constant-pressure specific heat at the respective temperature.
It shows that the specific heat of isobutane varies greatly
with the temperature.

The properties of geothermal brine vary from one
reservoir to another. But, for estimation purposes, the
properties of geothermal brine may be approximated by those
of water, without introducing appreciable error. For a
given mass ratio of water to isobutane, the enthalpy of
liquid water in Btu per pound of isobutane can be plotted
in the same diagram. Since the specific heat of the
saturated water is nearly constant, the temperature-enthalpy
relationship is shown as a straight line, and its slope is
dictated by the mass ratio of water to isobutane. Because
of the nature of the specific heats of water and isobutane,
the concept of logarithmic mean temperature difference
cannot be applied directly to calculate the heat transfer
area requirements, and a correction factor for the
logarithmic mean temperature difference has to be applied
to take into account the effect of minimum temperature
difference between the two fluids. If the overall heat
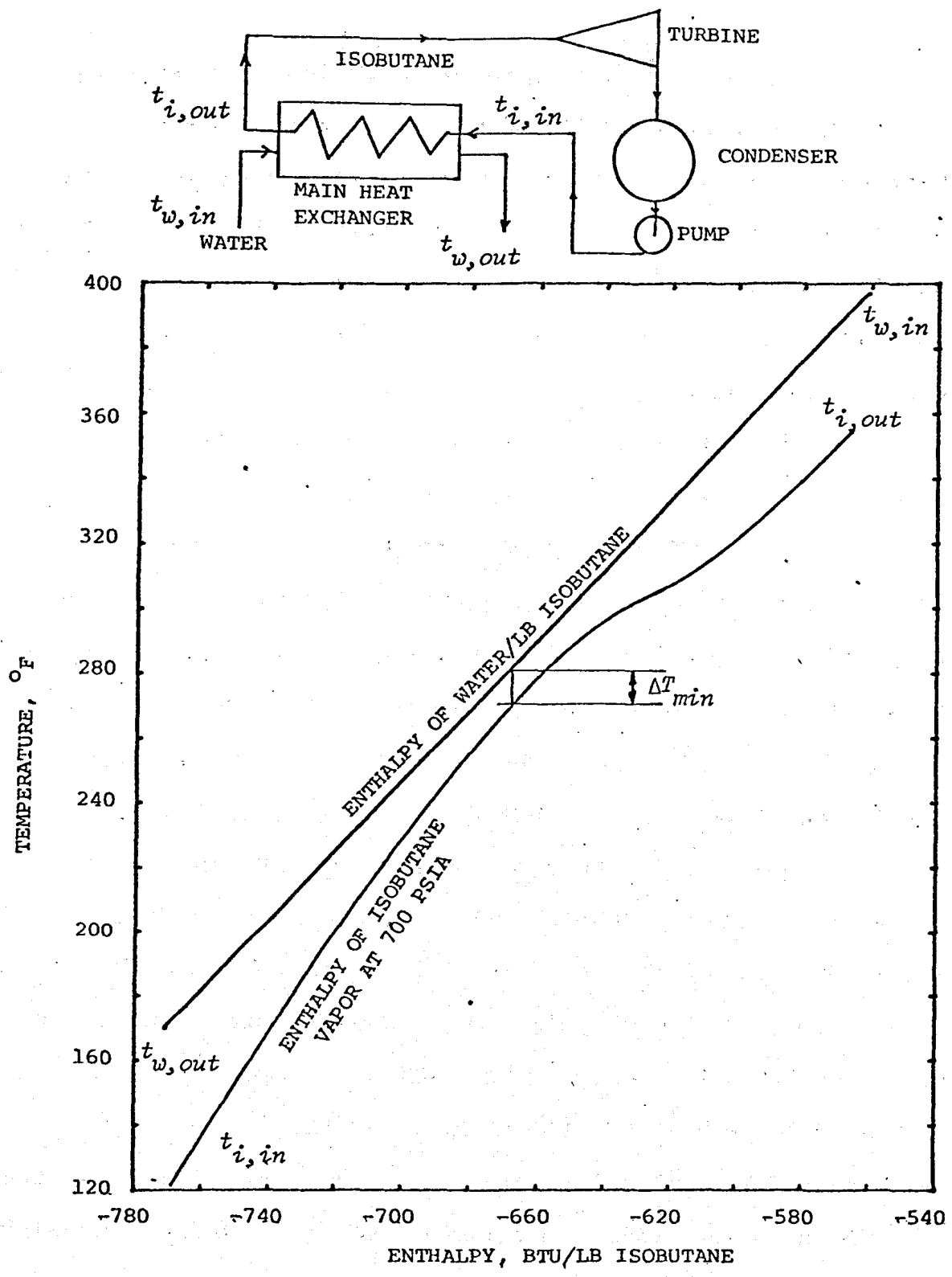
Figure 1-1.   Temperature-Enthalpy Relationship of Isobutane
              and Water in Heat Exchanger

transfer coefficient U_o is considered to be fairly constant, a numerical step-by-step integration may be used to determine the required heat transfer area A, ft^2, per pound of isobutane,

$$A = \frac{1}{U_o} \Sigma \frac{\Delta h}{\Delta T}$$

where Δh is the enthalpy change per pound of isobutane in Btu/lb. and ΔT is the mean temperature difference between the two fluids in the corresponding interval in ^{o}F.

In the analysis of an isobutane cycle, the temperature $t_{i,in}$ (Figure 1-1) of the entering isobutane is directly related to the condensing temperature of the cycle and the outlet pressure of isobutane from the pump. The inlet temperature $t_{w,in}$ of the hot geothermal fluid depends upon the reservoir temperature and the method of bringing the geothermal fluid to the surface. With the two inlet temperatures, $t_{i,in}$ and $t_{w,in}$, fixed as shown in the Figure 1-1, different patterns of temperature distribution may be obtained by varying the minimum temperature difference between the two fluids and the slope of the temperature-enthalpy line for water. The mass ratio of the two fluids prescribes the slope of the temperature-enthalpy line for water.

Effects of System Pressure on Power Production

The isobutane pressure of the binary system affects the power output greatly. The optimum isobutane pressures were found by trial heat balances. The effects of changing the isobutane pressure on the work output are shown in Figure 1-2, based on the following assumptions:

Geothermal fluid	400°F Saturated water
Condensing temperature	120°F
Pump efficiency	80%
Isobutane turbine efficiency	85%
Temperature difference	10°F

and the optimum isobutane pressure is 1,000 psia for an outlet temperature difference of 10°F.

For a given outlet temperature difference, ΔT_1, and a minimum temperature difference, there is an optimum pressure. Many heat balances were made to determine the optimum pressure as a function of water temperature at wellhead, and the results are given in Figure 1-3 for an outlet temperature difference of 10°F. For all the calculations in this study, a condensing temperature of 120°F, a pump efficiency of 80%, and an isobutane turbine efficiency of 85% are assumed. The work output in this chapter refers to the turbine work minus the work of feed pump, not including the work of downhole pump and other auxiliaries.
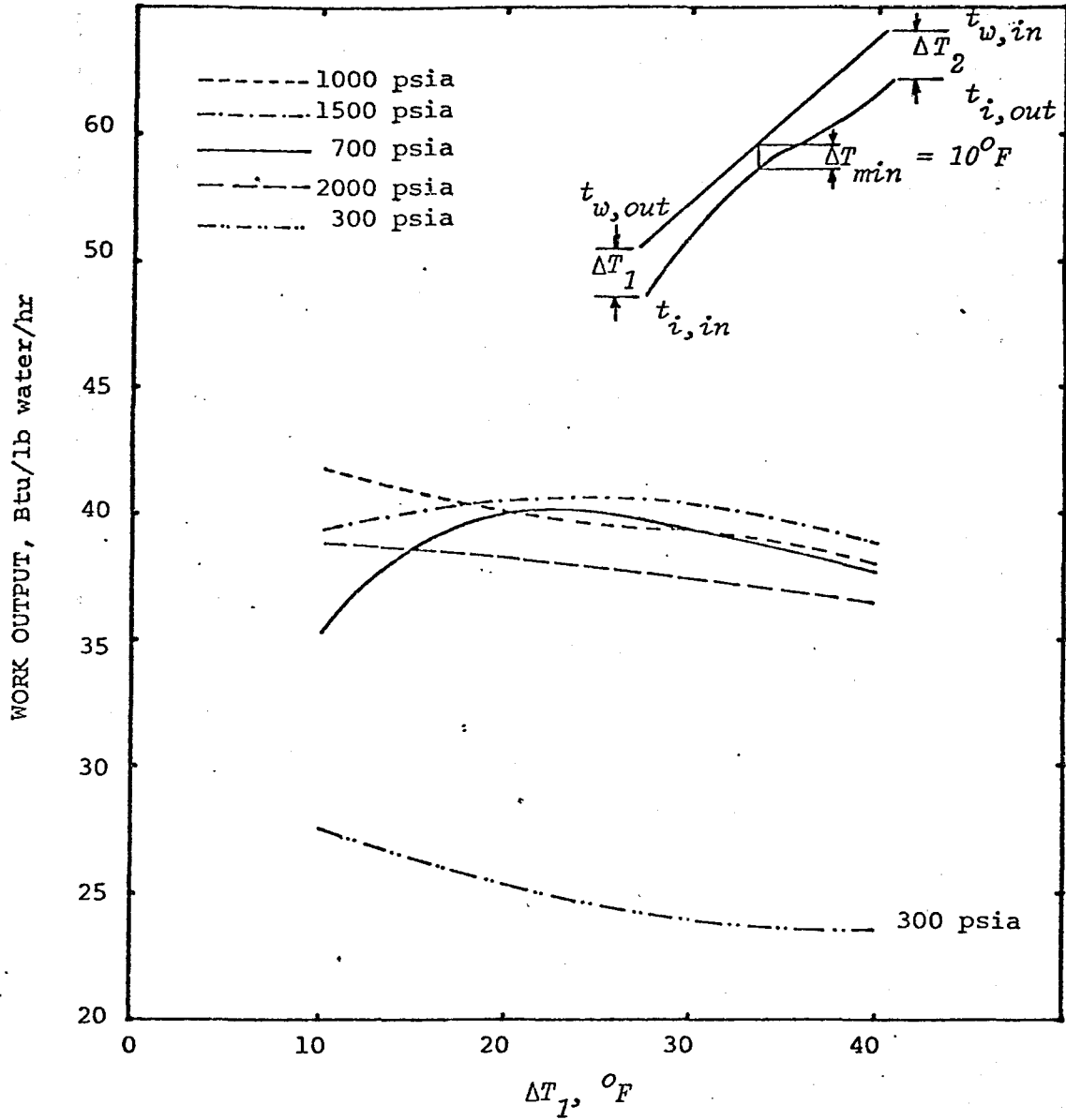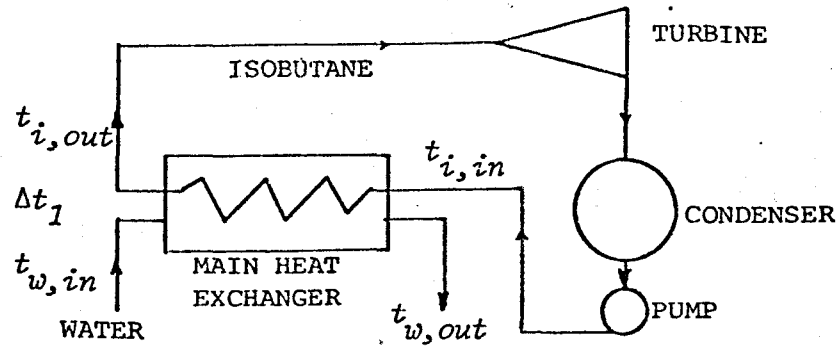
Figure 1-2. Effect of System Pressure on Work Output
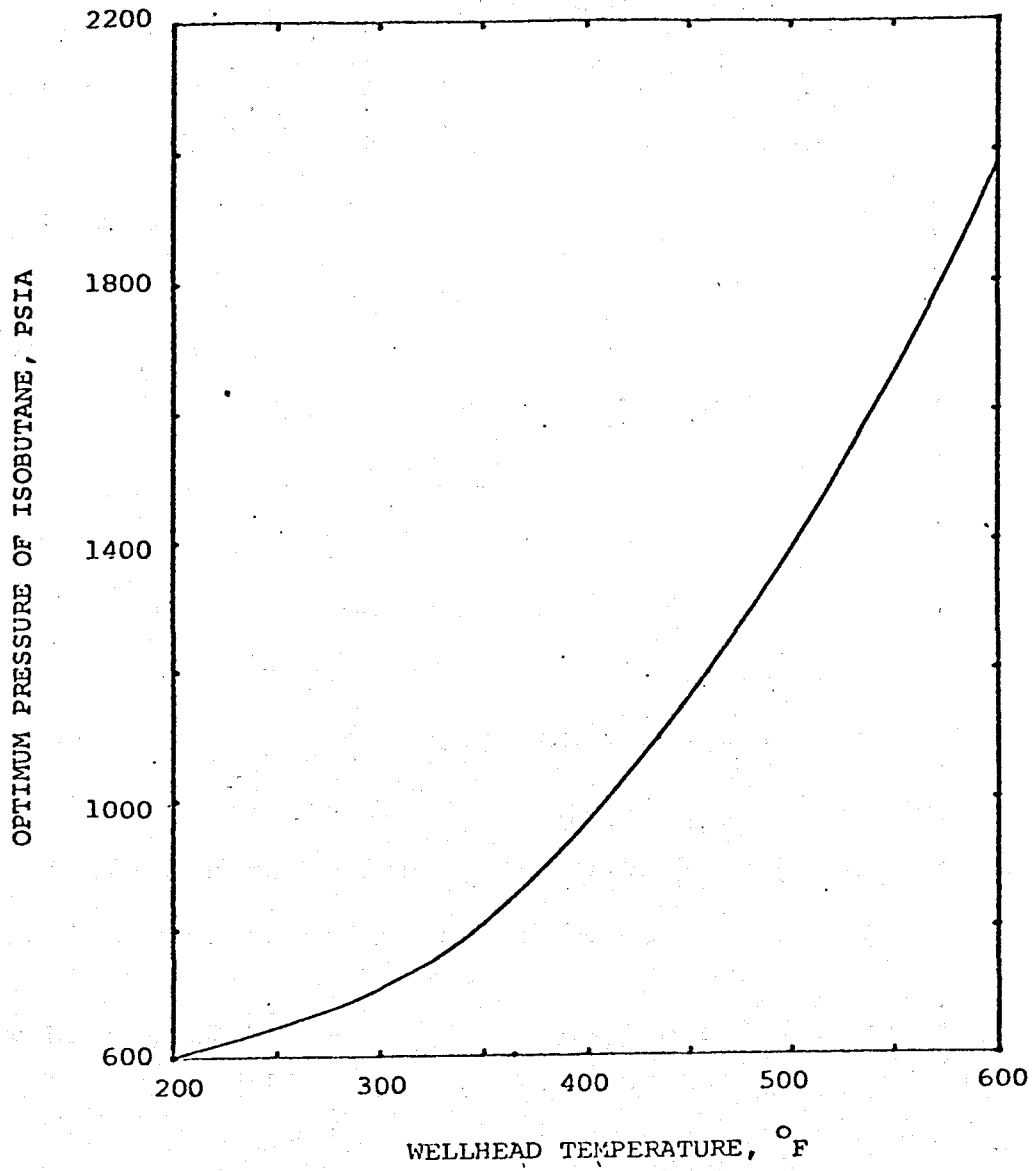
Figure 1-3. Optimum Pressure of Isobutane for Maximum
Power Output

Optimum Minimum Temperature Difference in the Heat Exchanger for Compressed Geothermal Liquid

The minimum temperature difference in the heat exchanger plays an important role. As this temperature difference increases, the size and the cost of the heat exchanger decreases. At the same time, the work output per pound of the geothermal fluid decreases because of the decrease of the outlet temperature of isobutane and the high heat content of the waste brine. The effect of changing the minimum temperature difference, ΔT_{min}, on the outlet temperature of isobutane and water, for a fixed mass flow rate ratio of water and isobutane, is shown in Figure 1-4, in which a mass ratio of 1.0 is assumed. For the fixed inlet temperatures, $t_{i,in}$ and $t_{w,in}$, the outlet temperatures, $t_{i,out}$ and $t_{w,out}$, vary significantly. The areas of heating surface were calculated by numerical step-by-step integration with the following assumptions:

Geothermal fluid	400°F Saturated water
Isobutane pressure	1000 psia
Mass ratio of fluid to isobutane	1.0
Overall heat transfer coefficient	175 Btu/(hr)(ft^2)($^\circ$F)

ISOBUTANE   TURBINE

$t_{i,out}$ $t_{i,in}$

CONDENSER

$t_{w,in}$ MAIN HEAT EXCHANGER   PUMP

WATER $t_{w,out}$

400

$t_{w,in}$

360

$t_{i,out}$

320

280

ENTHALPY OF WATER/LB ISOBUTANE

ΔT_{min}

40

30

20

5   10

240

$t_{w,out}$

ENTHALPY OF ISOBUTANE VAPOR AT 1000 PSIA

200

TEMPERATURE, °F

160

120

$t_{i,in}$

-780   -740   -700   -660   -620   -580   -540

ENTHALPY, BTU/LB ISOBUTANE
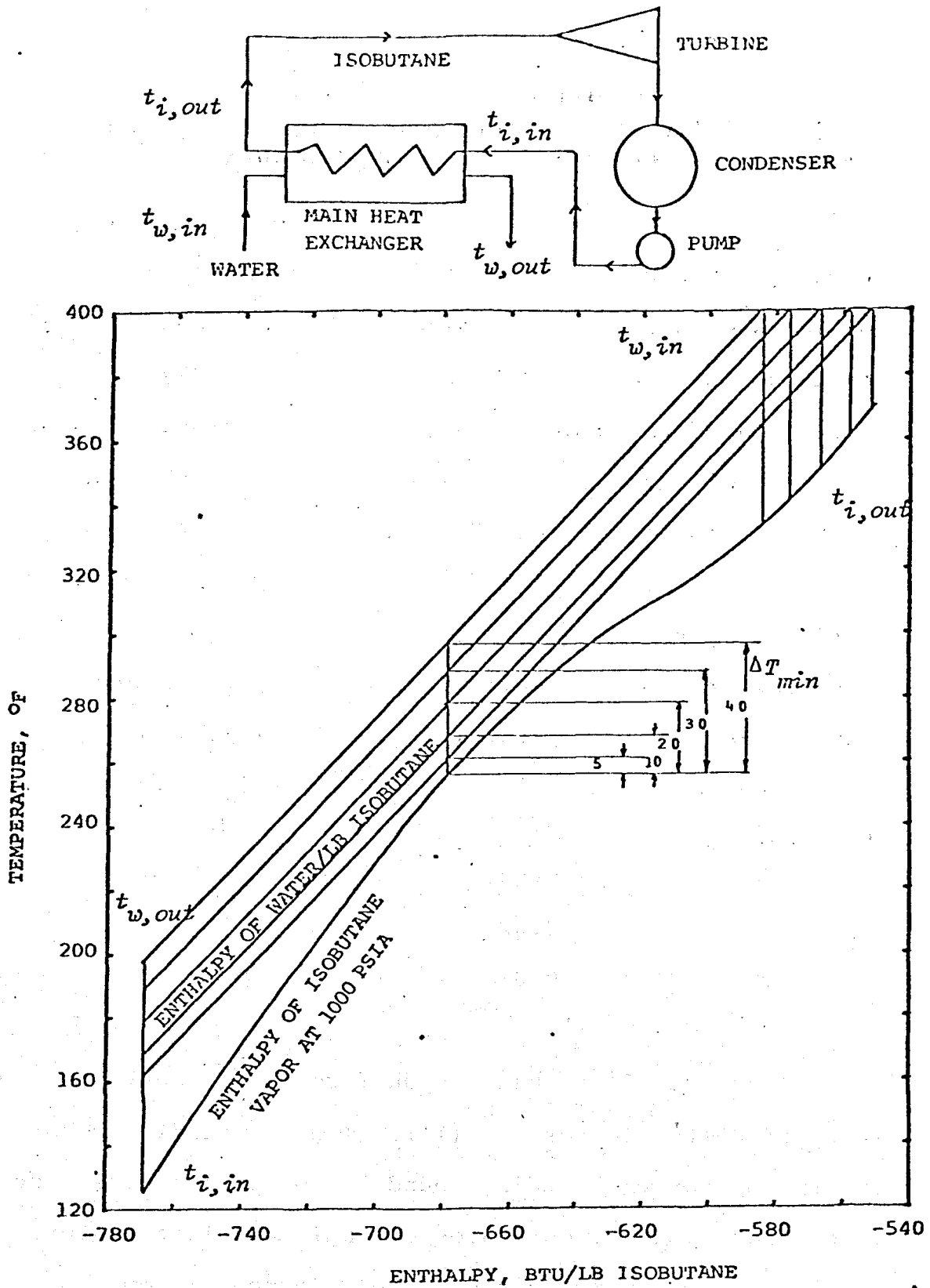
Figure 1-4. Effect of Variation of ΔT_{min} on Outlet Temperature $t_{w,out}$ and $t_{i,out}$.

The results are:

ΔT_{min}	$t_{w,out}$	$t_{i,out}$
$5\,^{\circ}F$	$176\,^{\circ}F$	$383\,^{\circ}F$
10	182	377
20	192	365
30	203	354
40	213	343

ΔT_{min}	Heating Surface	Power Production
$5\,^{\circ}F$	0.072 $ft^2/(lb$ water/hr)	9.7 Mwh/10^6lb.water
10	0.047	9.3
20	0.033	8.8
30	0.025	8.5
40	0.019	8.2

The cost of the heating surface is currently in the
range of $5 to $10 per ft^2 (11). The annual fixed charges
and the operating cost is around 12% of the initial cost;
it depends upon the estimates of equipment life, interest
rate, and other minor costs. The major cost items for
power production are the well, power plant, and transmission.
A detailed cost analysis is difficult, and the results are
expected to be different from one case to another. A

conservative estimate of the cost of power produced due to the efficient use of a heat exchanger is 1¢ per Kwh. By taking $40^{\circ}F$ minimum temperature difference as the reference point, the revenue from the additional power production and the annual cost of the heat exchanger were compared, as shown in Figure 1-5. It is interesting to note that based on these preliminary cost estimates, the optimum point of the minimum temperature difference is around $5^{\circ}F$. At a higher estimate of the revenue or a lower estimate of the heat exchanger cost, the optimum value of the minimum temperature difference will be lower. Because of the severe scaling on the heating surface of the heat exchanger, caused by the geothermal fluid, a conservative value of the heat transfer coefficient was used for the heating surface calculations. A value of U_o which is higher than 175 Btu/ (hr) (ft^2) $(^{\circ}F)$ will lower the optimum value of the minimum temperature difference.

Calculations for $600^{\circ}F$ saturated water at the inlet of heat exchanger were also performed. As the temperature of isobutane increases, the optimum pressure of the Rankine cycle also increases. For $600^{\circ}F$ saturated water, the optimum pressure of isobutane is about 2,000 psia (Figure 1-3). With all other assumptions to be the same as the previous case, the optimum value of minimum temperature difference is again in the neighborhood of 5 to $10^{\circ}F$.
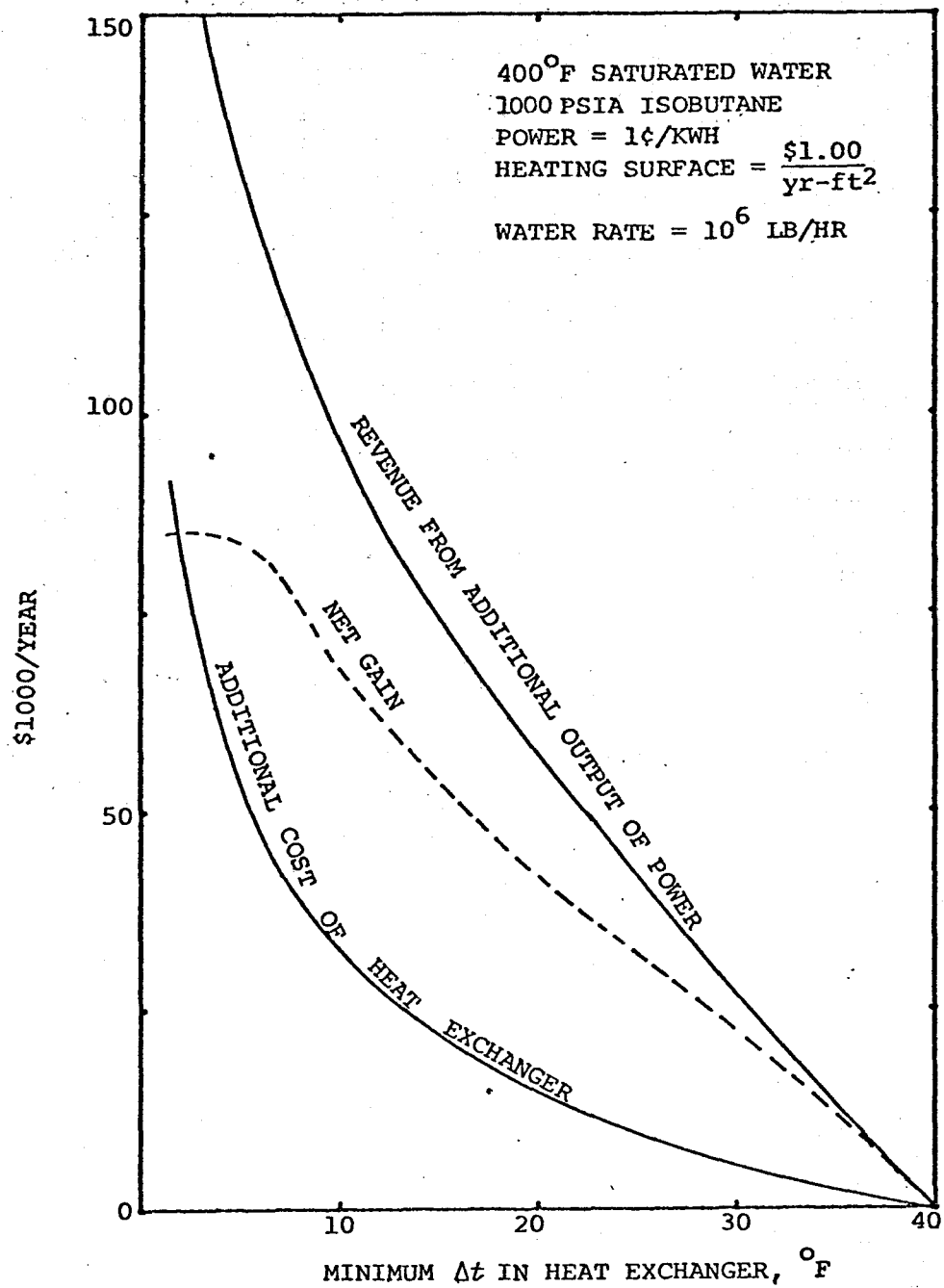
Figure 1-5. Pattern of Effects of Minimum Temperature
Difference Between Water and Isobutane
on Costs

Effects of Varying Mass Ratio of Water Flow to
Isobutane Flow for Compressed Geothermal Fluid

The variation of mass flow rate ratios of water to
isobutane, for a fixed minimum temperature difference, also
affects the heating surface area requirements and the power
output. Having decided that the minimum temperature
difference in the heat exchanger be kept at 5 to $10^{\circ}F$, the
effects of varying mass ratio were evaluated. The location
of the minimum temperature difference (Figure 1-1) varies
with the mass ratio. At a given value of minimum
temperature difference, its location moves downwards as the
slope of the temperature-enthalpy line of water increases.
At a small ratio of water mass to isobutane mass, the
minimum temperature difference could occur at the exit of
water. The advantage of lowering the exit temperature of
water is to lower the amount of waste energy; however, the
cycle efficiency drops as the exit temperature is pushed
down by tilting the enthalpy line of water. This effect
can be seen in Figure 1-6, where the minimum temperature
difference is kept to be $10^{\circ}F$, and the ratio of mass flow
rates is changed by changing the slope of temperature-
enthalpy line of water. With the same assumptions used for
determining the optimum value of minimum temperature
difference for $400^{\circ}F$ saturated water, the required area of
heating surface and the power output were determined for
$10^{\circ}F$ minimum temperature difference at various mass ratios.
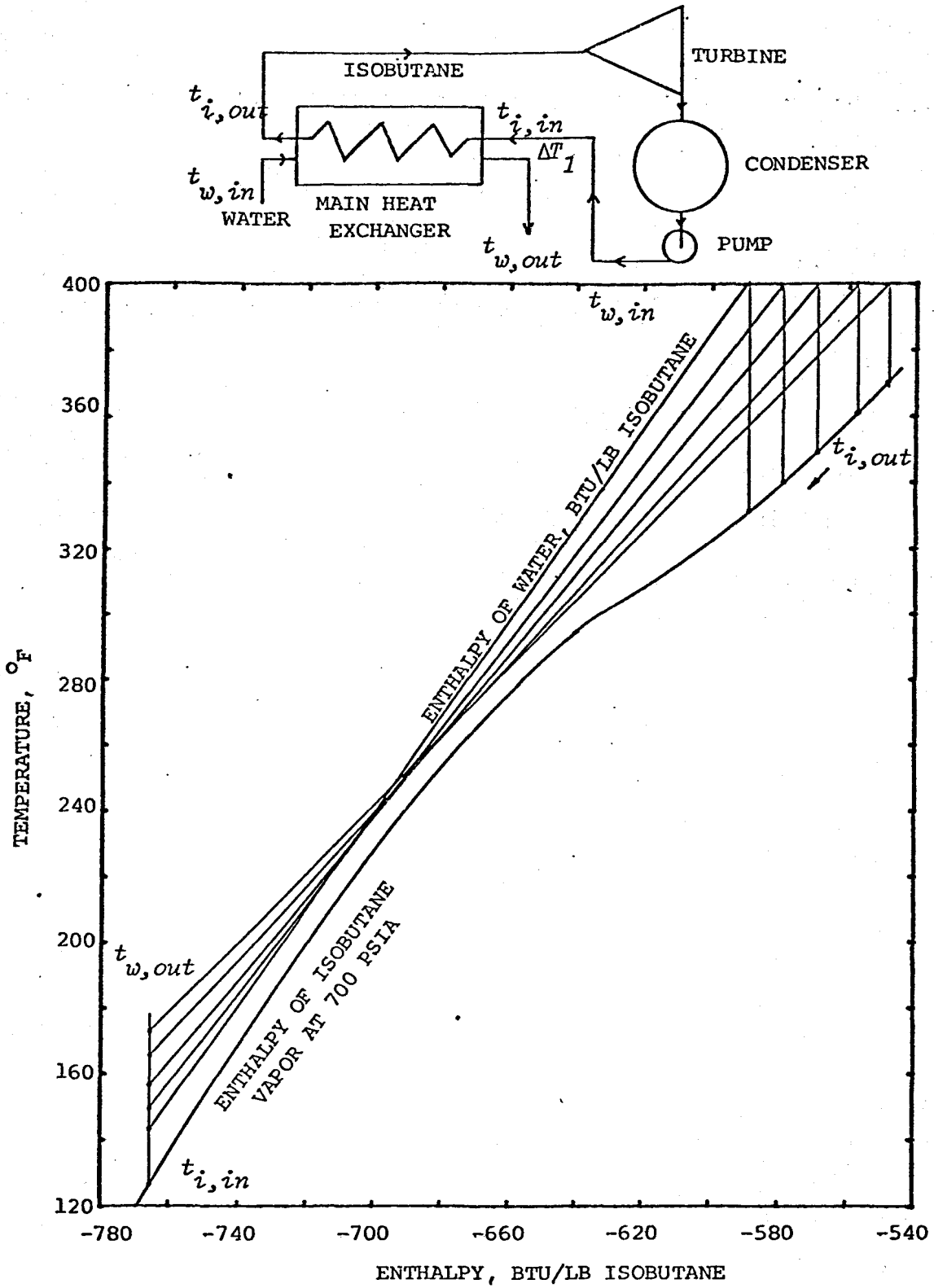
Figure 1-6.   Effect of Changing Mass Ratio on Outlet
Temperatures $t_{w,out}$ and $t_{i,out}$.

The effect of mass ratio on power output and heating surface area requirements is as follows:

Mass Ratio lb.H_2O/lb Isobutane	Exit Temperature of Water, $^\circ$F	Heating Surface ft^2/(lb.Water/hr)	Power Output Mwh/10^6lb.water
0.70	142	0.0970	10.7
0.75	147	0.0963	10.9
0.80	155	0.0955	10.8
0.85	165	0.0953	10.6
0.90	172	0.0952	10.3
1.00	179	0.0940	9.3

The optimum mass ratio is 0.74 pound of water per pound of isobutane. The effect of mass ratio on the revenue from power output is quite significant, but the variation of cost of heating surface area with mass ratio is not very large.

A similar evaluation was made for the case of 600°F saturated water with the following assumed data:

Geothermal fluid	Saturated 600°F water
Isobutane pressure	2,000 psia
Overall heat transfer coefficient	175 Btu/(hr)(ft^2)($^\circ$F)
Minimum temperature difference	10°F

The results are:

Mass Ratio lb.H_2O/lb. Isobutane	Exit Temperature of Water, ^{O}F	Heating Surface ft^2/(lb.Water/hr)	Power Output Mwh/10^6lb.water
0.56	145	0.1236	26.28
0.60	155	0.1086	25.78
0.65	165	0.1292	24.96
0.70	178	0.1390	24.10
0.75	197	0.1410	23.50

where the mass ratio of 0.56 gives the highest power output.
The variation of heating surface area with mass ratio is
different in this case as compared to one with 400°F. The
heating surface area decreases and then increases with the
change of the slope of the temperature-enthalpy line for
water. This variation is attributed to the change of the
mean temperature difference due to flatter temperature-
enthalpy curve for isobutane at 2,000 psia as compared to
one at 700 psia. In spite of this variation, the effect of
changing the mass ratio on the cost of heating surface area
requirements is not significant. However, the effect of
the variation of mass ratio on the revenue from power output
is very appreciable. No definite relationship between the
geothermal fluid temperature, mass ratio, water exit
temperature, and power output can be suggested, although it

is clear that the exit temperature of the waste water

increases and the power output decreases as the mass ratio

increases.

Optimum Minimum Temperature Difference in the Heat Exchanger for a Two-phase Geothermal Fluid

Figure 1-7 depicts the pattern of temperature

distribution of 800 psia isobutane heated by 400°F wet

steam of 29.2% quality at a mass ratio of 0.43 pounds of

steam to one pound of isobutane. If there is no heat loss

from the heat exchanger, the following equation applies:

$$\frac{1}{m} = \frac{x\ h_{fg}}{h_6 - h_5} = \frac{h_2 - h_3}{h_5 - h_4}$$

where m represents the pounds of wet saturated steam per

pound isobutane, x is the mass fraction of steam in the

geothermal fluid, h_{fg} is the latent heat of vaporization

of water in Btu/lb., h_2 and h_3 are the specific enthalpies

of liquid water in Btu/lb., h_4, h_5 and h_6 are the specific

enthalpies of isobutane in Btu/lb. at the respective points

as shown in the diagram. At a given mass ratio, the length

of line 1-2 and the slope of line 2-3 are fixed for a given

state of the steam at the inlet of the heat exchanger. By

moving these lines crosswise, the two terminal temperature
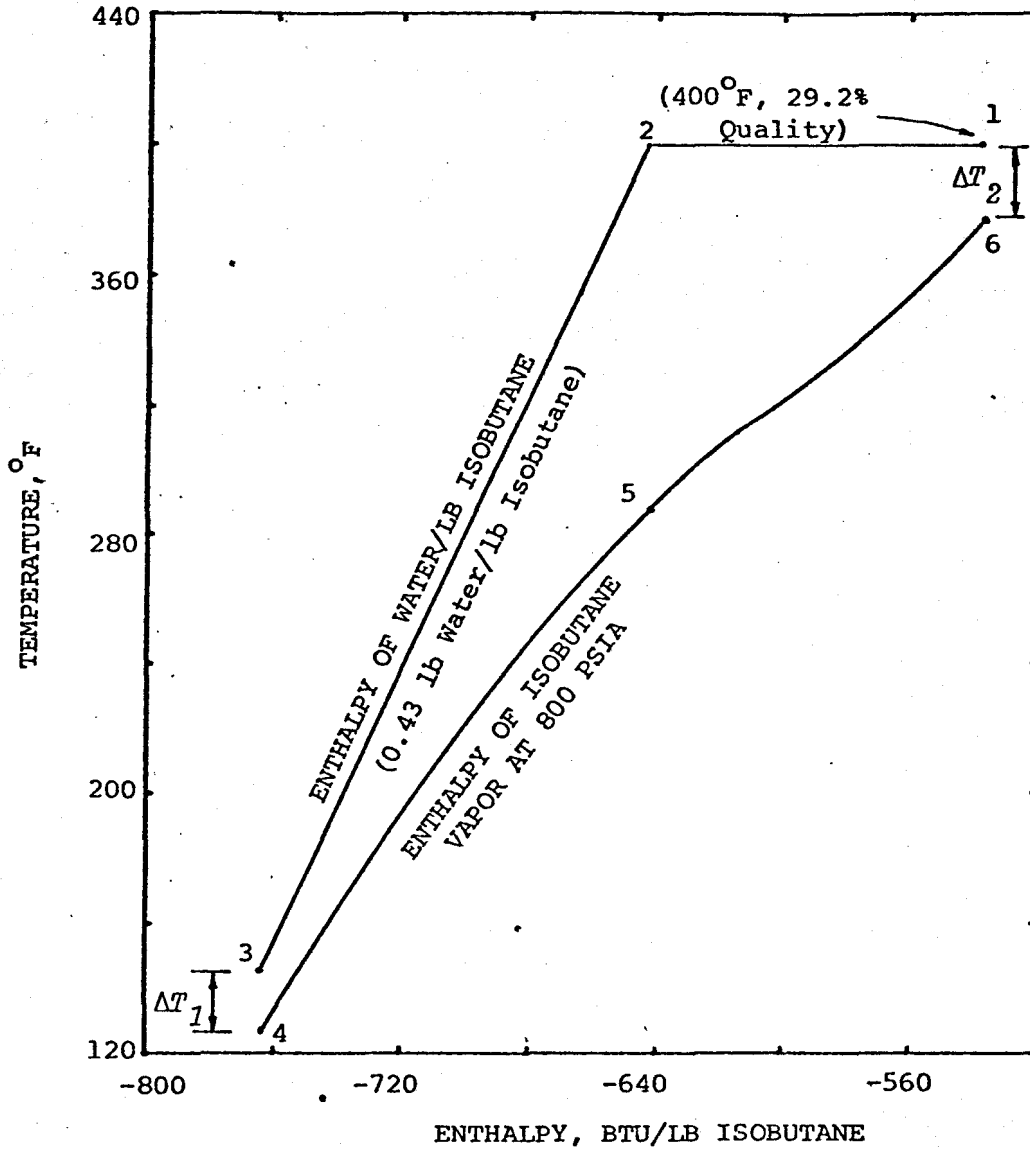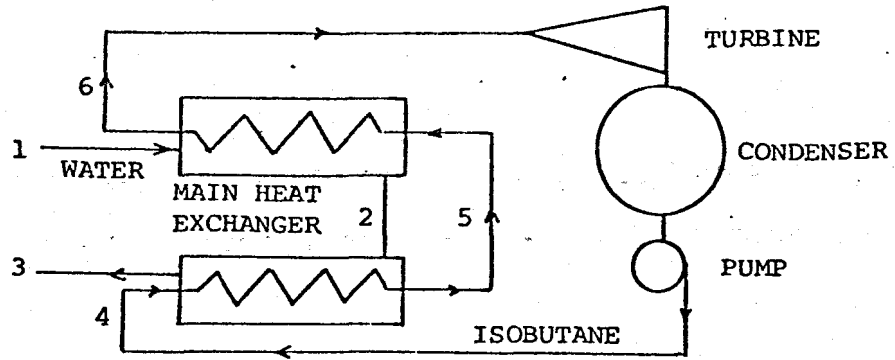
Figure 1-7. Temperature Distribution of Wet Saturated Steam and Isobutane in a Heat Exchanger

differences, ΔT_1 and ΔT_2, increase or decrease simultaneously. As the mass ratio changes, both the length of line 1-2 and the slope of line 2-3 are subject to change so that one of these two terminal temperature differences may remain in the same degrees. Various combinations of ΔT_1 and ΔT_2 can be obtained by changing the mass ratio and by moving the point 1 at a given temperature level.

Efforts were made to determine the optimum temperature differences for the plant which uses geothermal fluid in the form of liquid-vapor mixtures. From the point of high power output, the temperature of the fluid leaving the plant should be low, and the thermal efficiency of the cycle should be high. At a given pressure of isobutane for maximum efficiency, effects of temperature distribution on the cost of the heat exchanger were found to be insignificant because of the relatively small size of the heat exchanger required for transferring the heat from the wet steam at a large temperature difference. As an example, calculations were performed for the following conditions:

Geothermal fluid	400°F wet steam, 29.2% quality
Isobutane pressure	800 psia
Heat transfer coefficient,	
condensing section	500 Btu/(hr)(ft^2)($^\circ$F)
liquid-to-liquid section	175 Btu/(hr)(ft^2)($^\circ$F)
Annual cost of heating surface	\$1.00/ft^2
Power cost	1¢/Kwh

The trends of annual gross revenue from power and the annual cost of heating surface are shown in Figure 1-8. The optimum combination of the two temperature differences is around $\Delta T_1 = 10^O F$ at the water exit and $\Delta T_2 = 20^O F$ at the water inlet when the mass ratio of water to isobutane is 0.43. In another example, for wet steam at $312^O F$, 10.3% quality and isobutane at 700 psia, the maximum power occurs at $\Delta T_1 = 10^O F$, $\Delta T_2 = 18^O F$, with mass ratio = 0.48. From the above observations, it may be inferred that approximately $\Delta T_1 = 10^O F$ and $\Delta T_2 = 20^O F$ are the optimum temperature differences in general. However, additional heat balances are recommended for every special conditions.
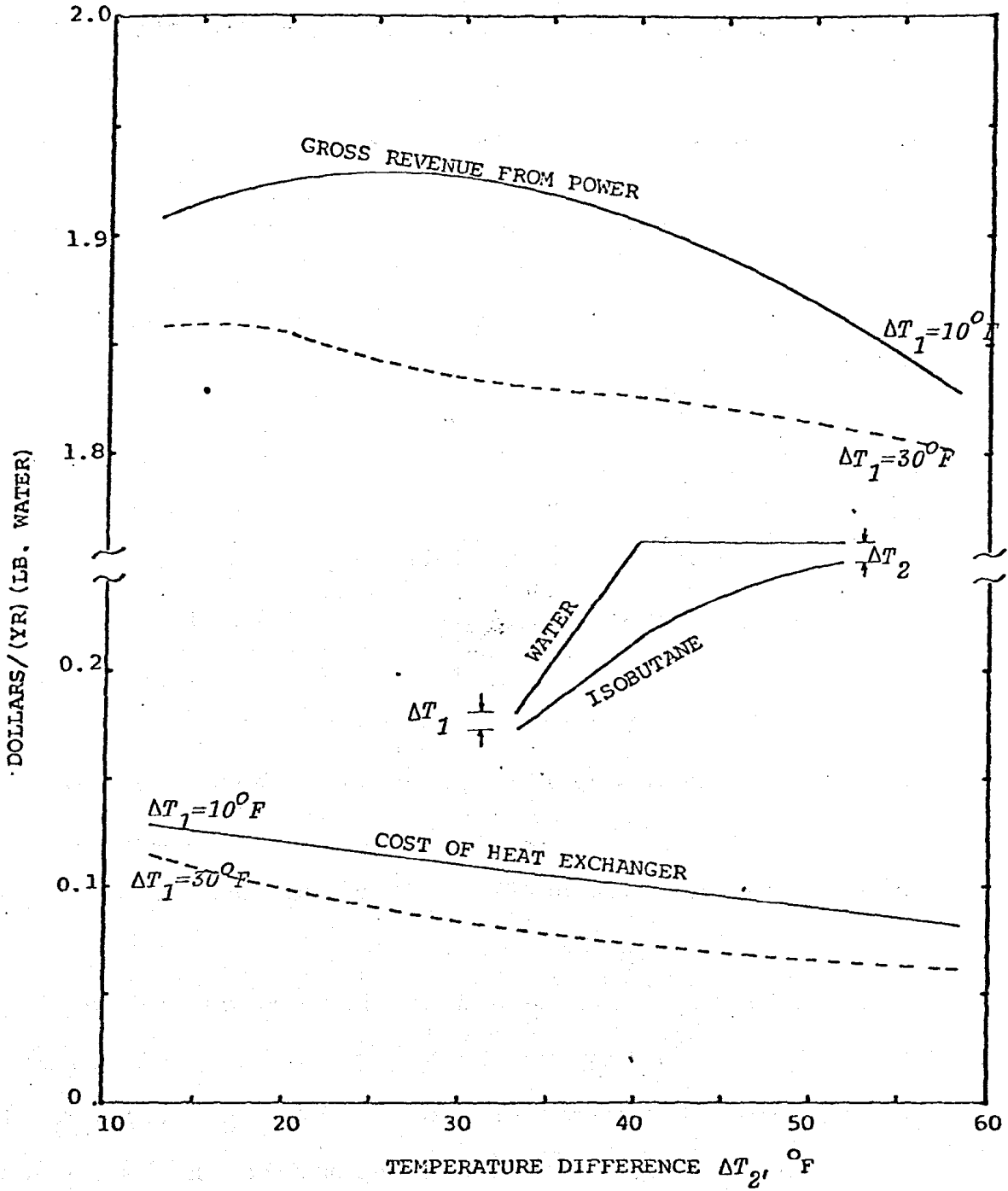
Figure 1-8. Effects of Temperature Distribution in a
Wet-Steam Heat Exchanger on Costs

CHAPTER II

COMPARISON OF POWER PRODUCTION PROCESSES

Introduction

The prevailing method of converting energy in the geothermal fluid into power is the two-stage vapor-flashing method, as illustrated in Figure 2-1A. This type of plant first became operational in New Zealand and then in several other countries. In the vapor-flashing process, steam is produced from the geothermal fluid by reducing the pressure of the fluid to a point below its vapor pressure. Steam flashed out is used directly to power a turbine which drives an electric generator. The design and performance of these plants have been reported in many publications and summarized by Chou and Ahluwalia (7). There are two other methods under serious study: the binary-fluid method with isobutane as the working fluid, and the total-flow-concept method.

The binary-fluid method has been discussed in Chapter I. A problem, which has yet to be resolved, is the scaling on the water side of the heating surface. To utilize the heat in the exhaust vapor from the isobutane turbine, the addition of a regenerative heat exchanger to the basic isobutane cycle was suggested. The consequence is that the exit temperature of fluid in the main heat
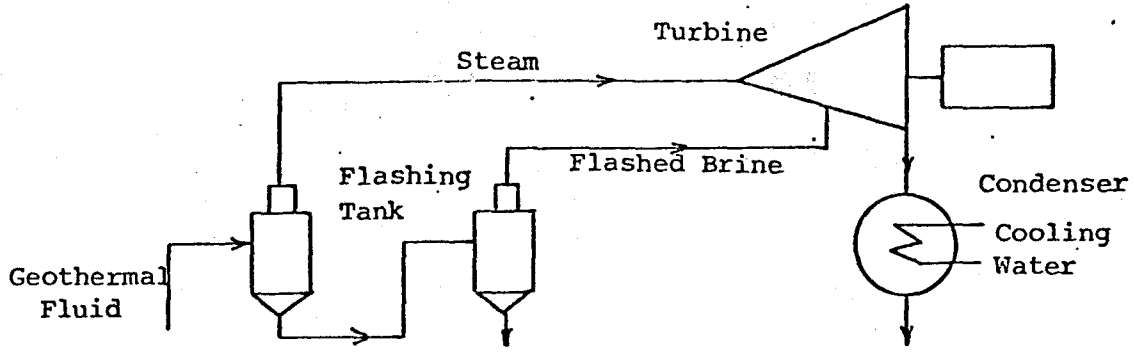
(A)   TWO-STAGE VAPOR-FLASHING SYSTEM
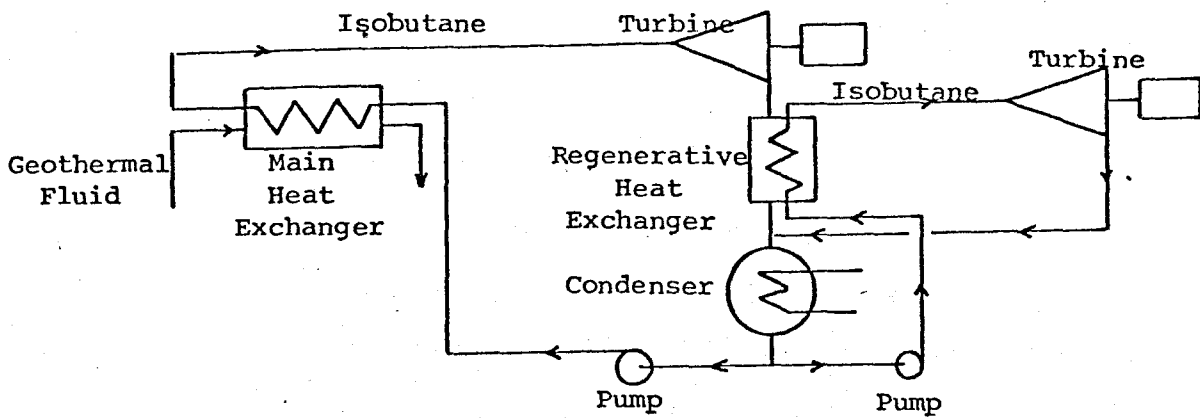
(B)   REGENERATIVE BINARY SYSTEM
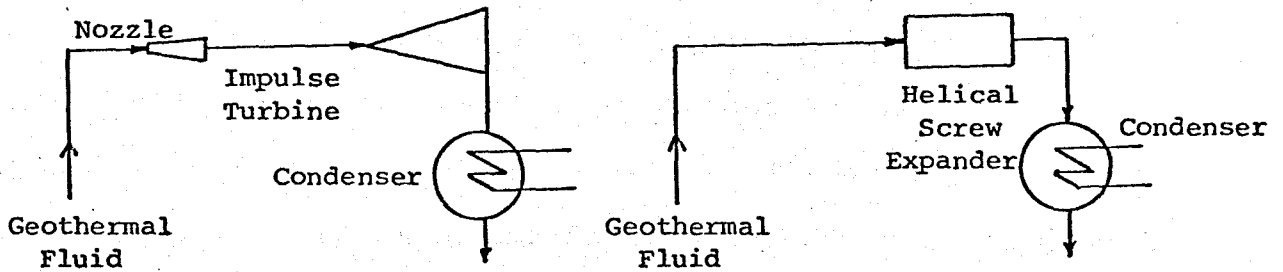
(C)   TOTAL FLOW SYSTEM

Figure 2-1.   Methods of Converting Thermal Energy in Hot
Geothermal Fluid into Useful Work

exchanger can be increased to such an extent that the waste heat in the effluent may be used for industrial heating as described by Chou, et al. (8). Figure 2-1B shows another version of a regenerative binary system. Here the regenerative heat exchanger is used to heat and to evaporate the isobutane flowing in a separate basic cycle. Then the hot isobutane vapor is expanded in the second turbine to produce additional power.

Figure 2-1C shows another method of power production from geothermal fluid, studied and advanced by Austin, et al. at the Lawrence Livermore Laboratory (3). The thermal energy of the geothermal fluid is converted to kinetic energy by the expansion of the fluid through a converging-diverging nozzle, and then the kinetic energy is transformed into power in an impulse turbine; or the fluid is directly expanded in some suitable machine such as a helical rotary screw expander (Figure 2-1C). The basic advantage of this method is its simplicity. The degree of success of the total flow concept depends upon the development of a reliable machine with sufficiently high thermal efficiency. A comparison of the power productions of the three processes, with and without a downhole pump, was made for providing some information on the selection of the processes, as given in the subsequent section.

Effects of Downhole Pump on Power Production

In a liquid-dominated reservoir, the hot geothermal fluid is in a compressed liquid state. If the hot fluid is driven out by natural forces through a well, the vapor pressure of the fluid drops due to the wall friction and the change of potential energy; thus the vapor flashes out while the temperature decreases. As a result of this transition, the available energy of the fluid decreases as it moves up. Both the flow rate and the change of available energy per unit mass of the geothermal fluid are functions of vapor pressure drop. To maximize the power production, the optimum value of vapor pressure at the wellhead must be determined in the early stages of plant design. For plants currently in operation, wellhead pressures are in the range of 50 to 100 psia (13). Pumping hot geothermal fluid from a reservoir increases the production rate of a well, eliminates scale formation on the well surface, lessens the surface fouling in the heat exchangers of a binary cycle plant, and preserves the available energy of the geothermal fluid. Several projects to develop downhole pumps are in progress under the sponsorship of the government. Although the future use of pumps depends upon the development of reliable pumps which are capable of handling hot geothermal fluids and the favorable economic analysis of a specific project, the effects of pumping on power production per unit

mass of fluid can be estimated within a small uncertainty.

In an actual situation, the enthalpy of the fluid in a reservoir is different from that at the wellhead due to the change of potential energy, the pump work, if any, and the heat loss from the well to its surroundings. However, by examining the magnitude of each term in energy equation for a typical well, one may accept the assumption that the enthalpy of the fluid is approximately constant throughout the well. For example, if the depth of the well is 1,000 ft., the change of potential energy is 1.28 Btu/lb. of geothermal fluid. After the well is in operation for a length of time, the heat transfer to the surroundings may be assumed to be negligible. The shaft work is about 2.5 Btu/lb. of geothermal fluid, and the change of kinetic energy is also insignificant. Comparing these to the total enthalpies in the range of 375 to 625 Btu/lb., it is reasonable to assume that the enthalpy in a well remains constant. For simplicity, this assumption is used for the comparisons of power production per unit mass of geothermal fluid with or without a downhole pump in the well. Other general assumptions on the fluids are that the thermodynamic properties of geothermal fluid may be approximated by those of water, and that the pressure effects on the properties of liquid may be neglected.

Based on a paper by Anderson (15), the turbine efficiencies of isobutane turbines are expected to be higher

than those of steam turbine of the same ratings. For the analyses of system performances here, the efficiencies of isobutane and steam turbines are assumed to be 85% and 75%, respectively. All the performance calculations in this section are based on 80% pump efficiency and $120^{\circ}F$ condensing temperature.

The heat balances of binary systems have been made with guidelines discussed in the previous chapter for the selection of the temperature distribution in main heat exchanger and the isobutane pressure. As to the regenerative heat exchanger, the temperature difference at the end of the high pressure isobutane inlet was assumed to be $10^{\circ}F$. Under these assumptions, the work output in Kwh per 1,000 pounds of geothermal fluid was calculated at various reservoir temperatures, and the results were plotted in Figure 2-2 at reservoir temperatures from 400 to $600^{\circ}F$. The wellhead pressure was assumed to be 100 psia without downhole pump. As expected, a downhole pump can improve the performance significantly. Regenerative isobutane system with downhole pump exhibits the best performance at high reservoir temperatures.

If the geothermal fluid temperature is less than $380^{\circ}F$, the degree of superheat of the exhaust isobutane will be too low to justify the addition of a regenerative heat exchanger to the binary system. When there is a demand for the low temperature heat in the waste fluid, a regenerative isobutane system can function at a wellhead
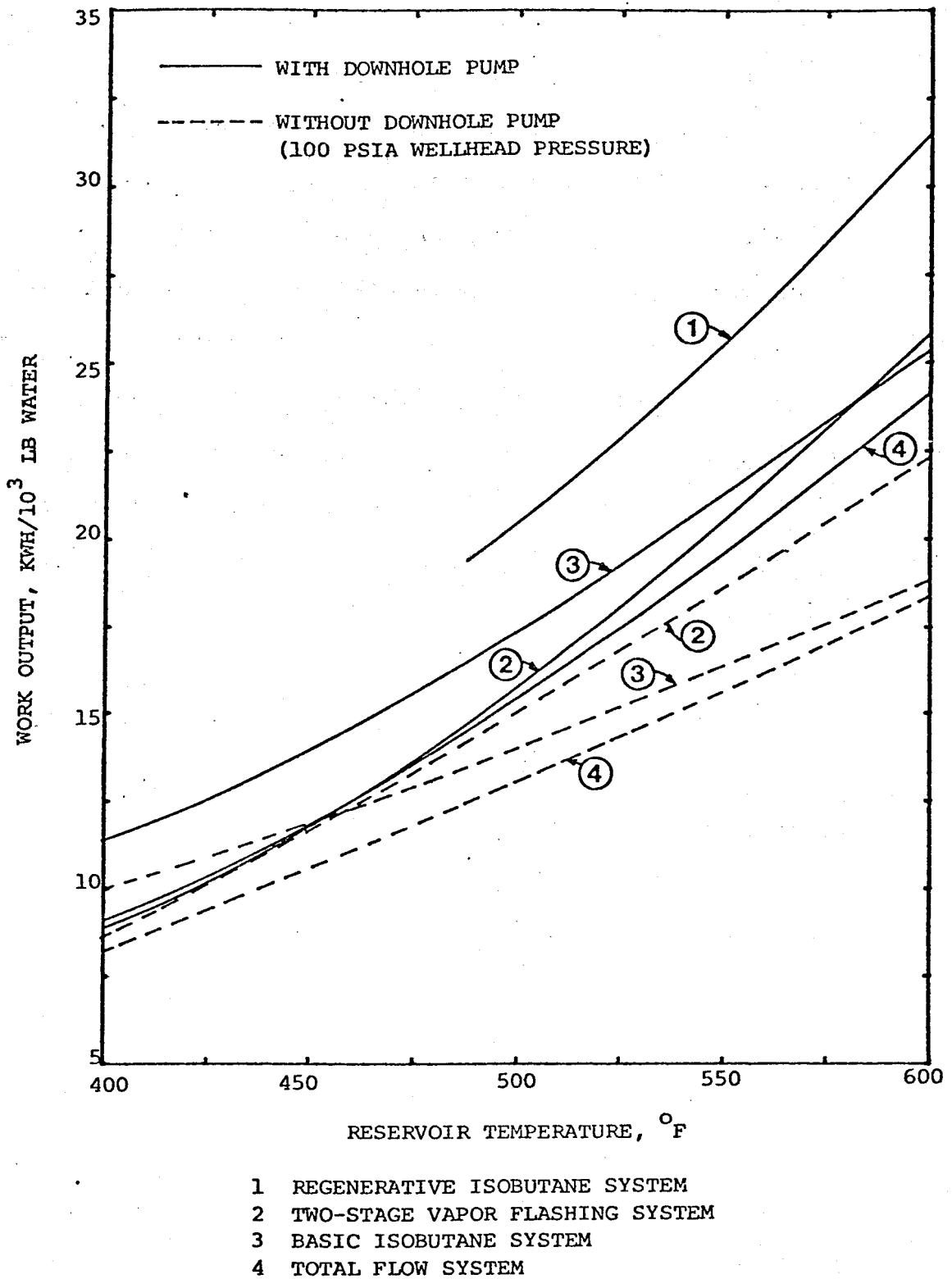
Figure 2-2. Effects of Downhole Pump on Work Output

pressure around 200 psia without a downhole pump. In such a case, the work output per unit mass of fluid is almost equivalent to that from a two-stage vapor-flashing system at 100 psia without a downhole pump. With a downhole pump, the work output from a regenerative isobutane system is shown by the Curve 1 in Figure 2-2. This curve is terminated at 485°F reservoir temperature because the liquid content in the exhaust vapor of the second isobutane turbine should be limited to 12%. If a suitable working fluid could be selected for the second closed cycle to avoid the high liquid content in the exhaust vapor. The Curve 1 may be extended to a reservoir temperature lower than 485°F.

Basic isobutane system with downhole pump gives the highest power output at low temperatures. The addition of a downhole pump exhibits significant improvement in performance in general as the reservoir temperature increases. Also, the formation of scale in the well and in the heat exchanger can be suppressed because the fluid is kept under pressure with the pump. At low reservoir temperatures, basic isobutane cycle may be used with 100 psia wellhead pressure without much loss of power output (Curve 3). In the performance calculations for a vapor-flashing system, the temperature of vapor from the second cyclone separator was taken to be the average of the condensing temperature and the saturated temperature of geothermal fluid at the wellhead. The performance of the vapor-flashing system is

not much improved by using a downhole pump (Curves 2).

The total flow concept system yields the least work output (Curve 4) because its performance is based on the engine efficiency of 55% according to the current technology (20). Through research it is likely that a nozzle and an impulse turbine may be developed to give the engine efficiency much higher than 55%. The total flow concept system can not be recommended until its efficiency is improved.

In all the previous calculations with a downhole pump, the pump work has not been accounted since the depth of the pump and its performance are uncertain. Assuming the depth of the pump to be 2,000 ft. and a combined efficiency of motor and pump to be 50%, the work input is 5.14 Btu per pound water, or 1.5 Kwh per 1,000 lb. water. So far no reliable pump has been developed to handle hot brine at such a depth. The application of binary system depends upon the successful development of downhole pump.

CHAPTER III

REGENERATIVE HEAT EXCHANGER DESIGN

Introduction

The regenerative heat exchanger is an important and costly equipment in the regenerative binary system. Its cost should be justified before adopting the regenerative isobutane system. An analysis is being attempted here to provide the estimates of the overall heat transfer coefficient, the pressure drops in the shell and tubes, and the cost.

The finned-tube configuration for the heat exchanger provides significant advantages over the plain-tube one when the heat transfer coefficient of one side is very small as compared to the other side. Finned surfaces of wide variety are manufactured commercially, and the three basic types are: cross-fins, pin-fins, and longitudinal-fins. The selection of a particular type depends on three factors: allowable pressure drop, velocity limitations, and fluid properties.

The amount of accurate and reliable data on finned-tube heat exchanger is extremely limited, particularly for units of large sizes. The research program at the University of Delaware has contributed significantly to an understanding of heat transfer and fluid-flow mechanisms on

the shell-side of a shell-and-tube heat exchanger containing plain tubes. Based on the results from this program, Bell (5) devised a procedure which could be used to predict the shell-side heat transfer coefficient and the pressure drop. In his procedure, the effects of leakage between tube bundle and shell, between tube and baffle, and between baffle and shell on heat transfer coefficient and pressure drop were shown to be important. Briggs et al. (6) investigated the technique of Bell and concluded that the Bell's procedure can be applicable in the shell-side correlation for finned-tube also. They utilized the method which was based on the Bell's procedure for the shell-side calculations of heat transfer coefficient and pressure drop, and their method is used in this study. For determining the shell-side heat exchanger tube-sheet geometry, Dunkelberg's method (9) is used.

Tube-Sheet Geometry

The performance of a regenerative heat exchanger depends greatly upon the tube configuration and the characteristics of the fins attached to the tubes.

The baffled shell-and-tube heat exchanger implements flows normal to the tube banks in a configuration which provides convenient piping access. This is accomplished

by inserting the tube bank longitudinally into the cylindrical shell and providing segmented baffles normal to the tube bank to guide the flow. The tube-side fluid enters and leaves through headers and the shell-side fluid enters and leaves through the nozzles welded directly on the shell.

This type of arrangement destroys the ideal normal flow. The flow through the baffle segment is parallel to the tubes, and the flow approaching each normal flow section is neither uniform nor normal. In addition, practical design requires that there be small clearances between the baffles and the shell, and between the baffles and the tubes. Also, if there is a substantial clearance between the outer row of tubes and the shell, it leads to significant bypassing to decrease the heat exchanger effectiveness.

Aluminum is an excellent material for heat exchanger tubes and fins. It is cheap, easy to fabricate, and light in weight. It has high thermal conductivity and does not react with isobutane chemically. Copper is a good alternative, but its cost is higher than aluminum.

Cross-fins are most suitable for the finned-tubes of a heat exchanger which handles large flow with small allowable pressure drop. They can be placed at right angle to the axis of each tube, or to a group of tubes. Included in this catagory are the helical fin and annular fin. Helical fins provide excellent heat transfer

36

performance, but the fin height is limited by the amount
that the fin can be stretched in wrapping operation.
Annular fins provide ease of fabrication and cause less
pressure drop.

The fin spacing is usually determined by manufacturing
tolerance; the value is usually from 6 to 15 fins per inch
depending on the incentive to minimize the volume of the
heat transfer matrix. As the number of fins per inch is
increased, the unit can be made progressively more compact.
At the same time, however, it becomes more sensitive to
clogging by dirt and small irregularities in the fin
spacing. Ten to twelve fins per inch represent a good
compromise.

As the regenerative heat exchanger operates with
clean fluids at all times, relatively small tubes which
offer high heat transfer rate may be used to make the heat
exchanger compact. However, the pressure drop of fluid
increases as the diameter of tube decreases. Arbitrarily,
tubes of 1 in. outside diameter are selected.

The fin efficiency is a function of temperature
distribution in the fin, and in turn it depends on the
thermal conductivity, the dimensions of the fins and the
heat transfer coefficient between the fin surface and the
fluid. The influential parameter is $H_f\sqrt{h/Kb}$, where H_f
is the fin height in ft., h is the heat transfer coefficient
in Btu/hr-ft-$^\circ$F, and K is the thermal conductivity of the

fin material in Btu/hr-ft-^{O}F, and b is the fin thickness
in ft. Typical fin height is ¼ to ½ in. for general
purposes. The fin efficiency can be adversely affected if
the fin is not integrally or metallurgically bonded to the
tube by soldering, brazing, or welding.

The tube array plays an important role. It may be
classified as either in-line or staggered. The in-line
tubes tend to give a somewhat lower pressure drop and
poorer heat transfer because the flow tends to be channeled
into high velocity regions in the center of the lane
between the tube rows. The staggered tubes produce better
mixing of the flow over the tube banks and give a higher
pressure drop.

Tube-pitch affects the heat transfer coefficient also.
In practice, a pitch of 1.25 times the outer diameter of
tubes is generally adopted for plain tubes. For finned-
tube bundle, the pitch to tube diameter ratio must be
higher because of the protrusion of the fins.

To investigate the problem of designing a regenerative
heat exchanger, sample calculations were made for a plant
which processes 600°F saturated water at 10^6 lb. per hr. as
shown in Figure 3-1. Figure 3-2 shows the temperature-
enthalpy relationship of isobutane inside the regenerative
heat exchanger. Superheated isobutane at 95.7 psia and
321°F enters the shell side of the regenerative heat
exchanger. It transfers heat to the compressed liquid
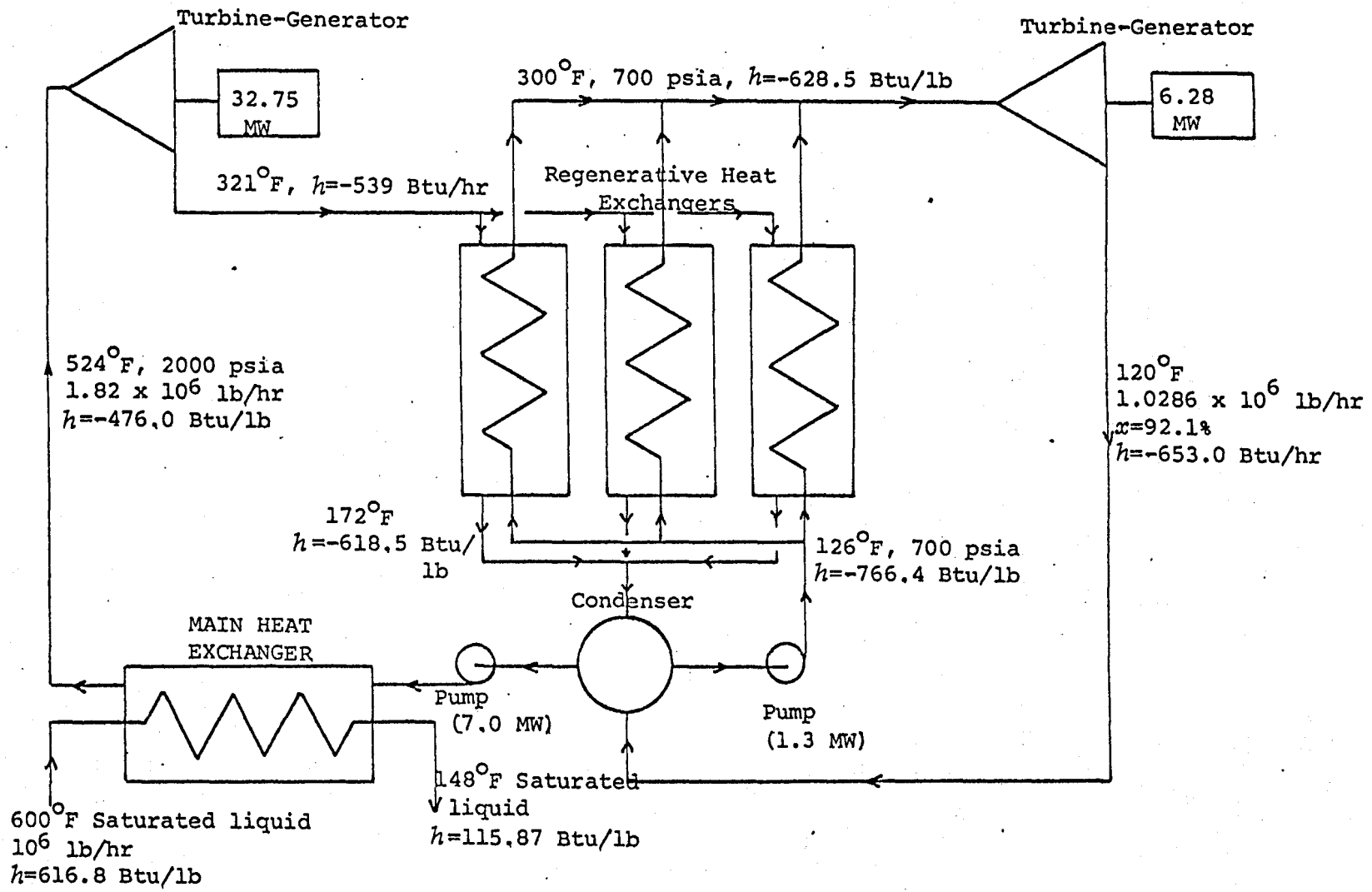
Figure 3-1. Flow Diagram of a Regenerative Isobutane System
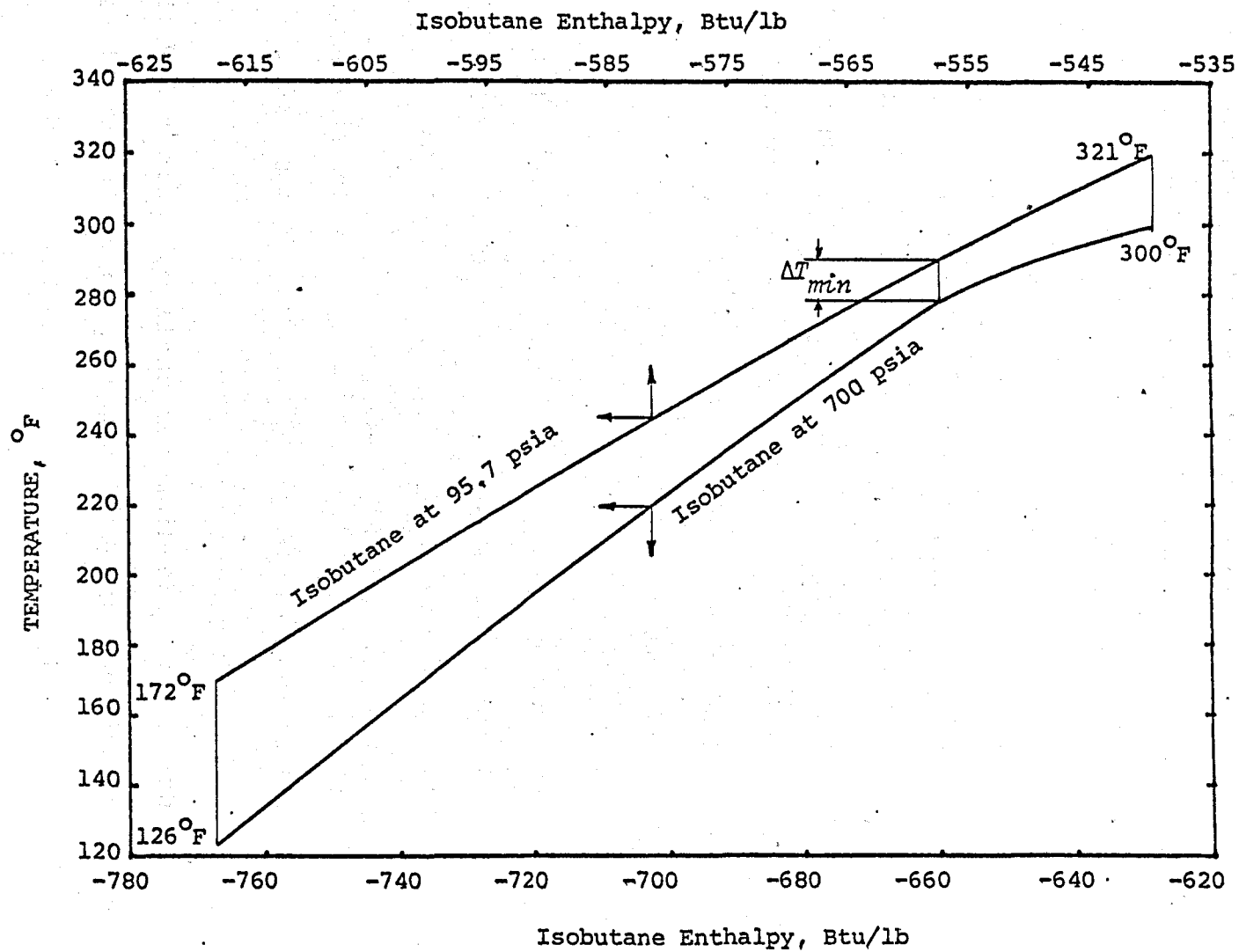
Figure 3-2. Temperature-Enthalpy Relationship of Isobutane in a Regenerative Heat Exchanger

isobutane and exists from the heat exchanger at 172°F. The liquid isobutane enters the tube side at 126°F and 700 psia and comes out at 300°F.

Based on the aforementioned considerations on the tube configuration and characteristics of fins, the following basic design data pertaining to the tube sheet geometry were established:

Type of fins	Annular
Inside tube diameter, d_i	0.884 in.
Outside tube diameter, d_o	1.00 in.
Tube thickness, t_{tube}	0.058 in.
Fin diameter, W	1.875 in.
Fin thickness, b	0.015 in.
Number of fins/inch, N_{Nf}	11
Fin height, H_f	0.4375 in.
Outside surface area, A_o	3.884 ft^2/ft
Inside surface area, A_i	0.2314 ft^2/ft
Surface area ratio, A_o/A_i	16.83
Thermal conductivity of aluminum alloy, K	100 Btu/(hr)(ft^2)(°F)
Equilateral triangular pitch, p	2.8 in.

A 60 in. outside diameter shell was selected with two passes on the tube side and single pass on the shell-side. Utilizing the method devised by Dunkelberg (9) for

determining the tube-sheet geometry, and following the TEMA
Standards (19), the variables which prescribe the tube
sheet geometry were determined as follows:

Shell thickness, t_s	0.5 in.
Shell inside diameter, d_s	59.0 in.
Allowable clearance between the outside tubes and shell, q	0.475 in.
Shell material	steel
Baffle cut, B	40 % of shell inside diameter
Window area, SW	1038 sq. in.
Mean cross-flow area, S_m	954 sq. in.
Total number of tubes in the exchanger, N	367
Baffle spacing, LL_B	39 in.
Number of tubes in the window, N_{wt}	137
Tube rows in the window section, N_{wr}	8.5
Tube rows in the cross-flow section, N_{cfr}	5.0

Figure 3-3 depicts the tube layout. In the
calculations, the following properties of isobutane at
the bulk mean temperature were used (10):

Scale 10:1

Tube Pitch	=	2.8"
Shell Thickness	=	0.5"
Number of Tubes	=	367
Tube Outer Dia.	=	1.00"
Tube Inner Dia.	=	0.884"
Fin Outer Dia.	=	1.875"
Baffle Pitch	=	39"

Flow of

Shell-side

Fluid

½" thick

60" Diameter

2.8"

Baffle-Cut

1.875"
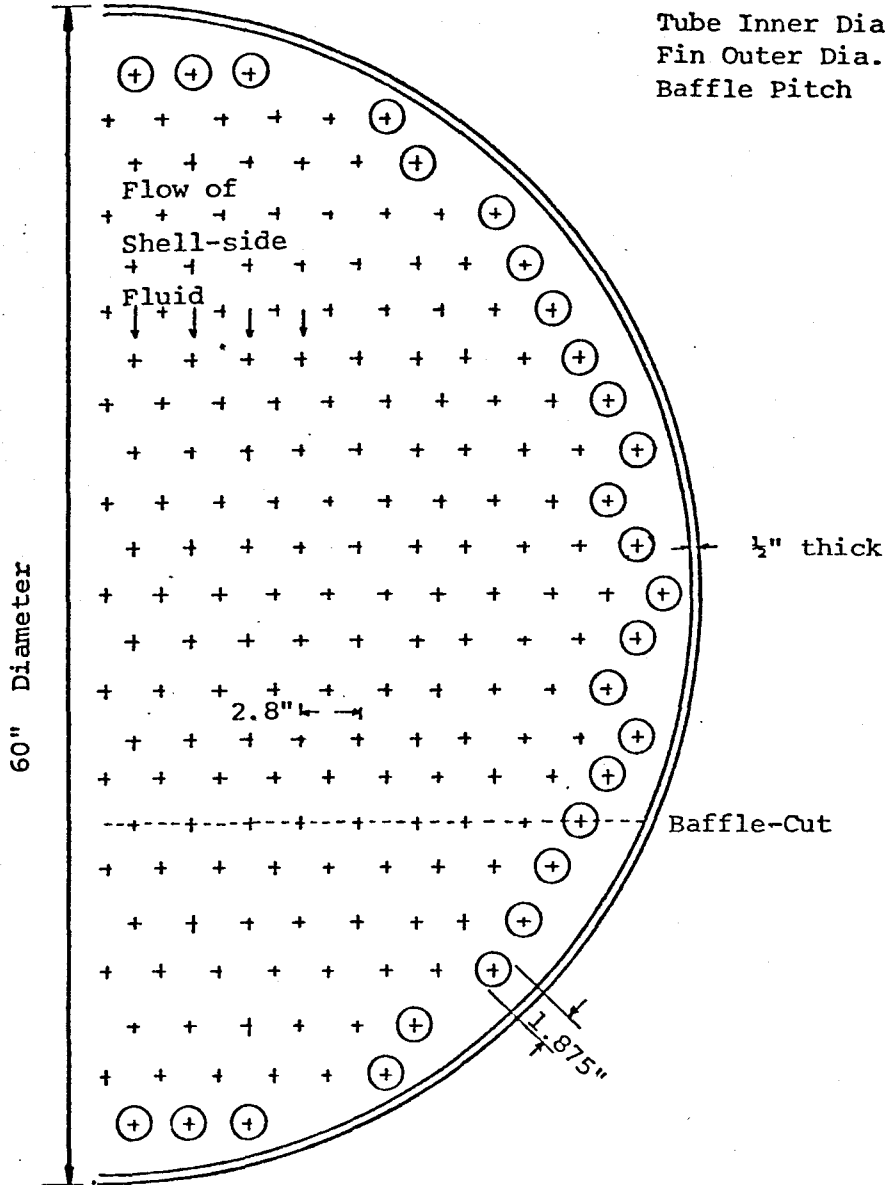
Figure 3-3. Layout of Tubes in Shell

	Tube-Side (213°F, 700 psia)	Shell-side (247°F, 95.7 psia)
Viscosity, μ, lb/hr-ft	0.1694	0.1161
Thermal conductivity, K, Btu/hr-ft-°F	0.042	0.034
Density, ρ, lb/ft^3	28.57	0.8
Constant-pressure specific heat, C_p, Btu/lb-°F	0.775	0.525

Overall Heat Transfer Coefficient, U_O

The overall heat transfer coefficient U_O can be expressed in terms of the individual resistances, as shown by Briggs et al. (6):

$$1/U_o = 1/h_L + rf + ro + (A_o/A_i)(1/h_i) + (A_o + A_i)(ri) + rm \qquad (3-1)$$

where h_L is the shell-side heat transfer coefficient for the heat exchanger with bypass and leakage in Btu/hr-ft^2-°F, rf is the fin resistance in hr-ft^2-°F/Btu, ro is the outside fouling factor in hr-ft^2-°F/Btu, A_i is the inside heat transfer area in ft^2, A_o is the total outside heat transfer area in ft^2, h_i is the inside heat transfer coefficient in Btu/hr-ft^2-°F, ri is the inside fouling factor in hr-ft^2-°F/Btu and rm is the metal resistance in hr-ft^2-°F/Btu. For a finned-tube configuration, the outside heat transfer area

includes the surface area of the fins. The various terms involved in equation (3-1) were calculated with the procedure given by Briggs et al., except the uses of Sieder and Tate equation and Briggs and Young equation with some modifications for the calculations of the inside coefficient h_i. This is because of lack of the experimental data for the geometry under the present consideration. It is believed that the modified Sieder and Tate equation, and Briggs and Young equation used for the calculations of h_i are more accurate and applicable. The results of the calculations are:

Hydraulic radius, rh	0.00983 ft.
Velocity of fluid in the tubes, V_i	4.5 ft/sec.
Reynold's number for the fluid in the shell, Re	30960
Colburn modulus, j	0.0129
Ratio of window heat transfer area to total heat transfer area, r	0.77183
Correction factor for the effect of baffling on shell side coefficient, Φ	0.7105
Effective number of crossflow constrictions in series, N_C'	30.0
Correction factor for row number effect on heat transfer coefficient, χ_t	0.9432
Bypass area, A_{BP}	0.2573 sq. ft.

Fractional bypass area factor, FF_{BP} 0.0388

Viscosity ratio correction
factor, $(\mu/\mu_w)^{0.14}$ 1.0

Shell side heat transfer
coefficient with no bypassing
or leakage, h_{NL} 300.0 Btu/(hr)
(ft^2) ($^\circ$F)

Diameter of the hole in the
baffle, d_{hB} 1.90625 in.

Number of tubes through each
baffle, N_{BT} 218

Baffle-to-tube leakage area, S_{TB} 0.1405 sq. ft.

Central angle, θ 158.1°

Baffle diameter, d_B 58.7 in.

Leakage area between baffle
and shell, S_{SB} 0.108 sq. ft.

Total baffle leakage area, S_L 0.2485 sq. ft.

Ratio of S_L/S_m 0.0375

Shell side heat transfer
coefficient with bypass
and leakage, h_L 257.03 Btu/(hr)
(ft^2) ($^\circ$F)

Assumed fin efficiency, ϕ_f 0.8

Fin resistance, rf 0.001024 (hr)
(ft^2) ($^\circ$F)/Btu

Inside fouling factor, $ri(A_o/A_i)$ 0.008415 (hr)
(ft^2) ($^\circ$F)/Btu

Log-mean area of the tube
wall, A_{mean} 0.2463 sq. ft.

Metal resistance, rm 0.000762 (hr)
(ft^2) ($^\circ$F)/Btu

Tube side heat transfer
coefficient, h_i 393.6 Btu/(hr)
(ft^2) ($^\circ$F)

Overall heat transfer
coefficient, U_o 17.45 Btu/(hr)
(ft^2) ($^\circ$F)

Total Heat Transfer Area $A_{o,t}$ and Tube Length L per Pass

As shown in Figure 3-1, the flow of isobutane is accomodated by three sets of heat exchangers in parallel. Each set consists of 4 single shell-pass, double tube-passes heat exchangers and is equivalent to a 4 shell-passes, 8 tube-passes heat exchanger, as shown in Figure 3-4. The heat transfer Q of each set of heat exchangers is 48.26 x 10^6 Btu/hr.

The logarithmic mean temperature difference was found in terms of the terminal temperature difference as shown in Figure 3-2, and it must be compensated later for the irregularities of the specific heats of isobutane. The corrected mean temperature difference is the product of the logarithmic mean temperature difference and the correction factor F, which is a function of heat capacity ratio R and the required heat exchanger effectiveness P(19). The number of passes affects the value of F.

Knowing the value of overall heat transfer coefficient U_o in Btu/hr-ft^2-^{o}F, the total heat transfer requirements Q in Btu/hr, and the corrected mean temperature difference $\Delta T_{m,corr}$ in ^{o}F, the total surface area required for the heat exchanger $A_{o,t}$ in ft^2 was calculated. The total length of tubing required is equal to $A_{o,t}/A_o$, where A_o is the outside surface area per unit length of tube in ft^2. The length L of the tube per pass in ft. can be found by dividing the total length of the tubing required by the number of tube passes.
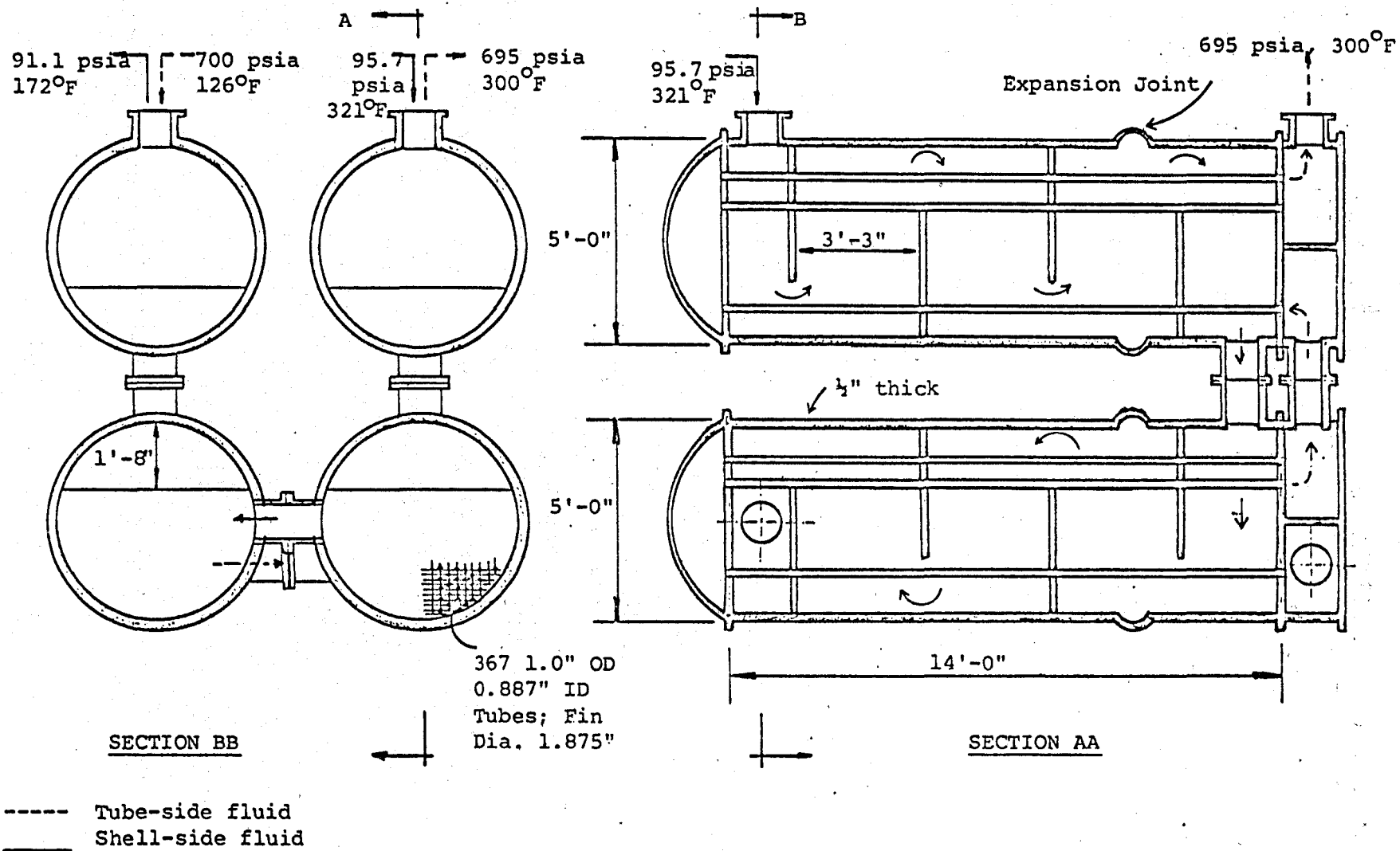
Figure 3-4. Schematic Diagram of a 4 Shell-Passes
8 Tube-Passes Regenerative Heat Exchanger

47

The irregular variation of specific heat of isobutane at constant pressure causes the minimum temperature difference ΔT_{min} to be located in the middle of the heat exchanger (Figure 3-2). Because of this effect, a further correction was made, and the corrected mean temperature difference becomes $18^{O}F$.

With the aforementioned assumptions, the following values were calculated for each equivalent 4 shell-passes, 8 tube-passes regenerative heat exchanger:

Heat capacity ratio, R	0.85
Heat excahnger effectiveness, P	0.89
Log-mean temperature difference, ΔT_{mean}	$32.0^{O}F$
Corrected mean temperature difference, $\Delta T_{m,corr}$	$18.0^{O}F$
Total outside heat transfer area required, $A_{o,t}$	132,966 sq. ft.
Length of tube per pass, L	14.0 ft.
Actual outisde heat transfer area	154,428 ft^2

Pressure Drop on the Shell Side

The pressure drop on the shell side was calculated in the same way as originated by Briggs et al., except that the friction factor f_s in the shell was calculated by

Robinson and Briggs equation (18). The calculated values
for the shell side pressure drop are:

Velocity of isobutane in the window, V_W	31.7 ft/sec.
Cross-flow velocity based on S_m, V_m	31.7 ft/sec.
Geometric mean velocity, $V_z = (V_m \times V_W)^{\frac{1}{2}}$	31.7 ft/sec.
Fricition factor for the flow of fluid in the shell, f_s	0.299
Pressure drop through the window with the correction factor for leakage, ΔP_W	0.1925 psia
Pressure drop through each end, ΔP_E	0.623 psia
Total pressure drop in the shell, ΔP_s	4.6 psia

Pressure Drop on the Tube Side

The friction factor for the flow of fluids inside
the tubes can be found from the Moody's diagram in terms
of the Reynold's number. The calculated values for
determining the tube side pressure drop are:

Mass velocity of the fluid in the tubes, G_i	462534.7 lb/ hr-ft^2
Velocity of the fluid inside the tube, V_i	4.5 ft/sec.

Friction factor of flow on the tube side, f_i	0.024
Pressure drop through the tubes, ΔP_i	2.27 psia
Pressure drop through the headers, ΔP_h	1.99 psia
Total tube side pressure drop, ΔP_{total}	4.26 psia

Cost Estimation

The actual cost of a large heat exchanger has to be quoted by the suppliers after the design is completed. At present, it is impossible to make a precise estimate of the cost; however, the unit cost is believed to be in the range of $0.50 per sq. ft. of the heating surface area approximately (11). Thus, the total cost of the three heat exchangers is about $200,000, not including the installation cost. The sample design in this section is intended to demonstrate the feasibility of the construction of a large regenerative heat exchanger. The designs were not optimized. By applying the techniques of optimization (16), the heat transfer area and the cost could be significantly reduced.

CHAPTER IV

CONCEPTUAL DESIGN OF A REGENERATIVE ISOBUTANE SYSTEM

Introduction

The geographical location of the plant can have a
strong influence on its success as a commercial venture.
But, the site selection of a geothermal power plant is
unusually inflexible and depends basically on the location
of producing reservoirs. The factors that must be evaluated
in a study of plant location are: availability of energy
source, geographical considerations, market for power
produced, ecology effect, environmental impact, and cost of
the plant. No site can satisfy all the requirements, and
the final selection has to be based on comparing the
favorable and unfavorable features of each site.

This chapter describes the major components of a
10 MW regenerative isobutane plant, assuming that the
saturated water enters the plant at $600^{\circ}F$.

Flow Diagram

The operating pressures and temperatures were
determined by heat balances as discussed in Chapter I. The
major components, their connecting piping, important valves

and controls are shown in Figure 4-1. There are three
fluids in the system. The first is the geothermal fluid
that supplies heat energy for the power generation. The
second is the working fluid isobutane and the third is the
cooling water. Heat is rejected by the exhausted isobutane
in a condenser to the cooling water which is recycled
through a cooling tower. Table 4-1 lists the pressure,
temperature, enthalpies and the flow rates at the
respective points as noted in Figure 4-1.

Consideration was given to the size of the plant.
The experimental plant must be large enough to experience
all the problems of a production plant. The size of the
plant must fit the normal production rate from at least one
well. On the other hand, an unnecessarily large pilot
plant might become wasteful. Hence, the unit is designed
for a net power of 10 MW. Heat is transferred from the
geothermal fluid to the isobutane at a rate of 175×10^6
Btu/hr., which gives the gross output of 13.7 MW from the
two turbo-generators. The power requirement of the pumps
and other auxiliaries is 3.7 MW.

The saturated 600°F geothermal brine flows into the
plant at 0.35×10^6 lb/hr. The pressure is maintained
above its saturation pressure by a brine pump in order to
prevent flashing throughout the heat exchanger. This pump
has sufficient capability to force the used brine into the
reservoir through the reinjection well. Four main heat

① Brine Pump

600°F Saturated Liquid Water 0.35 x 10^6 lb/hr

FC

Main Heat Exchangers

TC PC

④ Isobutane 1975 psia, 524°F 0.62 x 10^6 lb/hr

③ Isobutane 2000 psia,138°F

NOTATIONS:

TC Temperature Control
PC Pressure Control
SC Speed Control
FC Flow Control
LC Level Control
MWC Load Control
CC Conductivity Control

⋈ Normally Open Valve
⋈ Normally Closed Valve
⋈ Control Valve

② To reinjection well Saturated Liquid Water

To Condenser

To condenser

Turbine-Generator
SC MWC
11.5 MW

Knockout Drum
LC

Turbine-Generator
SC MWC
2.2 MW

Main Feed Pump (2.5 MW)

Knockout Drum
LC

To Receiver

⑨

To Receiver

⑩

Regenerative Heat Exchanger
⑤ ⑥
TC PC
⑧

Condenser Vent

Cooling Tower
∞

LC Low-pressure Feed-pump (0.5 MW)

To Flare Stack

⑦

Receiver LC

Isobutane Fill

Make-up Water

Blow-Down

CC

Cooling Water 80°F, 4.6 x 10^6 lb/hr

To Receiver

53

Figure 4-1. Flow Diagram of 10 MW Regenerative Isobutane Power Plant

Table 4-1. Pressures, Temperatures, Enthalpies and Flow Rates of a 10 MW Plant

Point	Pressure, psia	Temperature, $^\circ$F	Enthalpy, Btu/lb	Flow Rate, 10^6 lb/hr
1.	1,800	600	616.8	0.35
2.	1,750	148	116.0	0.35
3.	2,000	138	−757.1	0.62
4.	1,975	524	−476.0	0.62
5.	96	321	−538.9	0.62
6.	91	172	−618.5	0.62
7.	96	120	−766.4	0.98
8.	700	126	−770.7	0.36
9.	695	300	−628.5	0.36
10.	96	120	−653.0	0.36

exchangers are provided in series with the valves so that any of the exchangers can be taken out of service for cleaning or repair while the plant is in operation. A bypass is provided to allow all the hot brine to be returned to the reinjection line in case of emergency.

A receiver is placed underground below the condenser to store liquid isobutane. During normal operation, the main feed pump circulates 0.62×10^6 pounds of isobutane per hour at a total head of 2,000 psia. A knockout drum is used to prevent liquid isobutane from entering the turbine. Superheated vapor at 1975 psia and 524^OF expands through the isobutane turbine to drive the main generator, which produces 11.5 MW power. The exhaust vapor from the turbine at 321^OF, 95.7 psia flows through the shell of the regenerative heat exchanger. It then condenses in a shell-and-tube condenser, and the condensate returns to the underground receiver by gravity.

The low-pressure feed pump draws isobutane from the receiver at a flow rate of 0.36×10^6 lb/hr and pressurizes it to 700 psia. The compressed isobutane passes through the tubes of the regenerative heat exchanger and comes out as superheated vapor at 300^OF and 695 psia. The superheated isobutane then expands in the second isobutane turbine to produce 2.2 MW power. The exhausted vapor from the second turbine is at 120^OF and 95.7 psia and has 92.1% quality.

Cooling water at 80^OF enters the tube-side of the

condenser at a flow rate of 4.6×10^6 lb/hr and leaves at
110°F. To conserve the supply of cooling water, cooling
towers are used. Because the evaporation process increases
the concentration of dissolved solids in cooling water,
continuous blow-down and make-up water are provided.

Specifications of Equipment

1. Regenerative Heat Exchanger

In Chapter III, a preliminary design of 3 regenerative
heat exchangers for a saturated 600°F geothermal brine at
10^6 lb/hr has been made. A separate design of a regenerative
heat exchanger for a saturated 600°F geothermal brine at
0.35×10^6 lb/hr is not attempted here since the present
design serves as a preliminary estimate only. One heat
exchanger as described in Chapter III shall be used in
the conceptual plant. It consists of four 5 ft. diameter
shells. Each shell has two tubes passes. There are 367
tubes of 14 ft. length per shell. The tubes are of 1 in.
outside diameter and 0.058 in. thick. The arrangement of
tubes and heat exchangers is shown in Figures 3-3 and 3-4.
Each tube has 11 1.875 in. O.D. annular fins per inch,
and the fin thickness is 0.015 in. The tubes and the fins
are made of aluminum alloy. The equilateral triangular
pitch for the tubes is 2.8 in. The shell is made of carbon

steel and is 0.5 in. thick. The baffle cut of the heat
exchanger is 40% of the shell inside diameter. The overall
heat transfer coefficient of the heat exchanger surface is
17.45 $Btu/hr-ft^2-^oF$, and the total area of the heating
surface is 154,428 ft^2. The isobutane vapor at 700 psia
from the low pressure feed pump flows through the tubes of
the heat exchanger at 0.36 x 10^6 lb/hr. The exhaust
isobutane at 321^oF, 95.7 psia from the main turbine flows
at a rate of 0.62 x 10^6 lb/hr to heat the isobutane in the
tubes. An expansion joint on the shell of each heat
exchanger is recommended to compensate for the different
thermal expansions of the shell and tubes.

2. Main Heat Exchangers

    The temperature distribution in a main heat exchanger
was discussed in Chapter I, and the guidelines of the design
was adopted from the proof-of-concept design by Bechtel
Corporation (4). There are three 4 ft. 6 in. diameter main
heat exchangers of counterflow, shell-and-tube type in series
for normal operation, and the fourth one is a stand-by
for repair and for periodic cleaning by chemical process.
The scale-forming brine flows in the shell for easy cleaning
while isobutane flows at a rate of 0.62 x 10^6 lb/hr in the
tubes. Each heat exchanger has a heat transfer surface
area of 10,435 ft^2, consisting of 2,292 tubes of 3/4 inch

diameter by 16 BWG thick by 24 ft. long. The tube material is 90-10 copper-nickel. Isobutane comes out of the exchanger at 524°F and 1,975 psia. The total heat load for the three heat exchangers is 175×10^6 Btu/hr. The overall heat transfer coefficient for the heat exchangers is estimated to be 175 Btu/hr-ft^2-°F during normal operation.

3. Receiver

The receiver is a steel underground tank of 40,000 gallons capacity. During operation, most of the working fluid is in circulation. Valves are provided so that all the isobutane can be drained into the receiver. The system shall be purged with carbon dioxide before the isobutane is charged into the system.

4. Isobutane Turbines

The overall efficiency of isobutane turbine is estimated at 85%. According to Anderson (15), this efficiency can be easily attained, and there is no technological problem to build a reliable isobutane turbine. Radial inflow type turbines with rotational speed around 7,200 rpm may be used. The two turbines are geared to their respective generators which operate at 3,600 rpm by

the use of a speed-reduction gearbox. Special shaft seals

have to be provided to avoided leakage of isobutane, and

a high pressure lubrication system is necessary.

5. Condenser

The condenser shall be conventional shell-and-tube

type with isobutane flowing through the shell side. The

recommended tube diameter for the condenser is from 3/4 to

$1\frac{1}{4}$ in., with velocities of cooling water from 5 to 7.5 ft/

sec. The number of tube passes could be either one or two.

The effective tube length is from 25 to 30 ft. per tube pass.

The condensing temperature is $120^{O}F$ for normal operation.

The total heat load for the condenser is 136×10^{6} Btu/hr.

The flow rate of isobutane in the condenser is 0.98×10^{6}

lb/hr, assuming $30^{O}F$ temperature rise of the cooling water.

Aluminum-brass or other aluminum alloys are the recommended

material for the tubes. The material of shell may be

carbon steel.

6. Cooling Towers

The dry cooling towers and the natural draft towers

are not practical for a 10 MW power plant. A mechanical

draft tower operating on forced or induced fan is more

desirable. For an induced draft tower, the crossflow design offers certain distinct advantages over the counterflow arrangement because the distance of air travel is independent of the fill height. Both the height of the fill and the distance of air travel of a crossflow tower can be adjusted to minimize the loss of draft. Also, the pumping head of crossflow tower is lower than that of counterflow tower. The forced draft tower has to encounter the disadvantage of non-uniform air distribution and partial recirculation of vapors. An induced draft crossflow tower is, thus, recommended for a 10 MW geothermal power plant.

The usual approach temperature is between 14 and $22^\circ F$, with cooling range between 15 and $35^\circ F$ (1). The make-up water can be supplied to the concrete basin of the tower from a canal or a storage pond. Provision for chemical treatment should be made, if necessary. The blowdown should be deaerated and then pumped into the brine reinjection line, in order to minimize the corrosion in pipeline and reinjection well.

7. Knockout Drums

In order to ensure that no dirt particles or droplets of liquid isobutane may enter the isobutane turbines, two knockout drums should be provided. The knockout drum is a

cylindrical carbon steel vessel with internal baffles.
Liquid isobutane from the drums is drained to the receiver
by level control.

8. Piping

Carbon steel is suggested for isobutane and brine
pipes. The main isobutane lines should be of 15 in. diameter
for the liquid and 21 in. for the vapor. For brine flow,
12 in. diameter pipes are recommended (14). These values
were calculated by assuming that the velocity of liquids
in pipe is 5 ft/sec., and that the allowable pressure drop
for the isobutane vapor is 1 psi per 100 ft. of pipe length.
Insulations of suitable type and thickness should be
provided according to the temperature and service of the
pipe, finished with mechanical protection such as metal
or waterproofing jackets. Provisions should be made for
thermal expansion of high temperature pipes.

Controls and Instrumentation

In the start-up sequence, the cooling system is
started first. It provides the heat sink for the process.
The brine system is started next, and then the isobutane
system. The controls and instrumentation for each sub-

system are:

A. Brine System

In case of the shut-down of the plant, the by-pass valve is opened, and the main brine valve is closed. Pressure, temperature controls and recorders are provided at the inlet as well as the outlet of the main heat exchangers. A modulating valve at the outlet of the main heat exchangers adjusts the brine flow rate to control the temperature of isobutane as it leaves the main heat exchangers.

B. Isobutane System

The receiver should be monitored by a level control. A pressure relief valve connects the receiver to a flare stack. A valve at the inlet of the main heat exchanger along with a trip valve stops the flow of isobutane during an emergency shutdown. Temperature and pressure controls are provided at the isobutane outlet of the main heat exchangers to sense the condition of isobutane. The pressure controller adjusts the main feed pump speed to maintain the designed pressure. The temperature controller adjusts the main turbine control valve which regulates the isobutane flow rate to maintain designed temperature

conditions. Level controls are provided at the knockout drums. The main turbine as well as the secondary turbine should be equipped with a stop valve and a control valve. Regulation is accomplished by a turbine generator controller that senses the speed of the turbine, electrical load, and temperature of isobutane at the outlet of the main heat exchangers or the regenerative heat exchanger. Turbine trip signals include high turbine speed, high exhaust pressure, low oil pressure, high bearing temperature, and generator trip. Upon trip the main turbine stop valve and the stop valve at the isobutane entrance to the heat exchangers close. The turbine bypass valve opens to allow the flow of isobutane to the condenser. For condenser, controlled vent valve allows the discharge of gases to the flare stack. Normally during shutdown, isobutane system has liquid in the receiver and vapor throughout the rest of the system. However, for initial start-up or after the system has been opened for maintenance or repair, an additional procedure is required to purge the air that has been introduced into the system. To accomplish this, the air is first evacuated with a vacuum pump, then carbon dioxide is introduced into the piping system, and all piping paths are sequentially opened and closed, with the carbon dioxide and residual air being vented out through the flare stack. The flare stack is provided with an electric ignitor type pilot flame to monitor isobutane concentration.

C. Cooling System

Local cooling water temperature and pressure indicators should be provided at the both ends of the condenser. The conductivity control for the cooling tower is used to control the salt content in the cooling water.

Cost Estimation

The estimated capital cost of a 10 MW basic isobutane plant for $380^{\circ}F$ saturated liquid water with additional facility for experimentation, conceptually designed by Bechtel Corporation (4), is $23 million, of which $7.5 million is for experimental facilities. The equipment cost of major components is estimated to be $6,190,000.

A preliminary estimate of the equipment cost of the 10 MW regenerative isobutane power plant under this study is given in Table 4-2. The unit costs from the reference (11,17) were scaled up to take care of the inflation at about 6% annually. Without any experimental facilities, the capital costs of the plant is expected to be $10,915,000. It is relatively low in comparison with the conceptual plant of Bechtel Corporation's design because the costs of engineering, contingency and some equipment are different.

For the same flow rate of the same geothermal fluid, the net power production of the basic isobutane plant is

8.5 MW with the same main heat exchanger, assuming that the superheat of exhaust isobutane is removed by a desuperheater. The total installed cost is lowered to $9,375,000 because of the omission of the second turbine, the regenerative heat exchanger and its auxiliaries, and the itemized costs are given in Table 4-3. If the power cost is 2¢/Kwh, the revenue loss of the basic plant is $262,000 per year. At 12% of the investment charge, the payback period of the additional equipment for regenerative plant is 11 years.

Table 4-2. Capital Cost of the Major Components of a 10 MW
Regenerative Isobutane Power Plant

Description	Installed Cost ($1,000)
Main Heat Exchangers (a)	900
Regenerative Heat Exchanger (b)	150
Turbine-Generator Modules (c)	1,500
Condenser (a)	500
Receiver (a)	30
Pumps and Drives (c)	525
Cooling Tower (a)	250
Knockout Drums (a)	75
Auxiliary Systems (c)	220
Electrical Equipment (c)	530
Concrete and Earthwork (c)	720
Piping, Insulation and Valves (c)	2,320
Instrumentation (c)	270
Site Improvement (c)	220
Building Costs (c)	805
Engineering Costs	1,000
Contingency	500
Total Installed Cost	10,915

Bases for Estimates:

a = Modern Cost-Engineering Techniques by Popper (17).

b = Heat Exchanger Design by Ozisik and Fraas (11).

c = Electrical Power Generation Using Geothermal Brine
    Resources for a Proof-of-Concept Facility by
    Bechtel Corporation (4).

Table 4-3. Capital Cost of the Major Components of a
8.5 MW Basic Isobutane Power Plant

Description	Installed Cost ($1,000)
Main Heat Exchangers (a)	900
Turbine-Generator Module (c)	1,000
Condenser (a)	500
Receiver (a)	30
Pumps and Drives (c)	350
Cooling Tower (a)	250
Knockout Drum (a)	35
Auxiliary Systems (c)	220
Electrical Equipment (c)	530
Concrete and Earthwork (c)	720
Electrical Bulk Materials (c)	400
Piping, Insulation and Valves	1,800
Instrumentation (c)	270
Site Improvement (c)	220
Building Costs	750
Engineering Costs	950
Contingency	450
Total Installed Cost	9,375

Bases for Estimates:

a = Modern Cost-Engineering Techniques by Popper (17).

b = Heat Exchanger Design by Ozisik and Fraas (11).

c = Electrical Power Generation Using Geothermal Brine
Resources for a Proof-of-Concept Facility by
Bechtel Corporation (4).

CHAPTER V

CONCLUSIONS

At present, there are basically three different systems for converting energy in geothermal fluid into power: vapor-flashing system, total flow system, and binary system. A comparison of the power production processes was made on the basis of work output in Kwh per 1,000 pounds of geothermal fluid for self flowing wells with wellhead pressure of 100 psia and for wells with downhole pumps. For simplicity, the assumptions were made that the enthalpy of the geothermal fluid in the reservoir is approximately equal to that at the wellhead, that the thermodynamic properties of geothermal fluid may be approximated by those of water, and that the pressure effects on the properties of fluid are negligible. The results showed that the performance of the two-stage vapor-flashing system is not appreciably improved by using a downhole pump (Figure 2-2). The total flow system is simple, but its success depends mainly on the development of a reliable machine with sufficiently high thermal efficiency. The regenerative isobutane system is impractical, if the geothermal fluid temperature is below $380^{\circ}F$. But, when the brine temperatures range from 485 to $600^{\circ}F$, the regenerative isobutane system with downhole pump exhibits superior performance as compared

to two-stage vapor-flashing system, basic isobutane system, or total flow system. One of the important components of an isobutane binary system is the main heat exchanger. Both the size of the heat exchanger and the power output of the system are greatly affected by the temperature distribution of fluids in the heat exchanger.

The temperature difference between isobutane and geothermal fluid in the heat exchanger is a vital concern to the economics of the plant operation. Both the magnitude and the location of the minimum temperature difference can affect the system performance if the geothermal fluid is liquid when it enters the heat exchanger. The optimal value of the minimum temperature difference has been investigated under a given set of assumptions: cost of power produced = 1¢/Kwh, cost of heating surface = $1.0/yr.-$ft^2$, condensing temperature = $120^\circ F$, isobutane pressure =1000 psia, mass ratio of isobutane to water = 1.0, overall heat transfer coefficient = 175 Btu/hr-ft^2-$^\circ$F; the magnitude of the optimum value was found to be 5 to $10^\circ F$ when the geothermal fluid is a saturated liquid at $400^\circ F$ (Figure 1-5). For a higher estimate of the power cost, a higher value of overall heat transfer coefficient, or a lower estimate of the heat exchanger cost, the optimal value of the minimum temperature difference will be lower. Calculations were also made for saturated geothermal liquids at higher

temperatures under the same assumptions, and the results indicate that the optimal value remains around 5 to $10^{o}F$. The location of the minimum temperature difference can be adjusted by varying the mass ratio of water to isobutane (Figure 1-6). It was found that the point of minimum temperature difference should be as close to the water exit as possible in most cases; however, this is not a rule. As to the cases of two-phase geothermal fluid, the heat transfer area was calculated by dividing the heat exchanger into a condensing section and a non-condensing section. In such cases, the optimal temperature differences at the exit and the inlet of the heat exchanger (Figure 1-8) were evaluated. The heat transfer coefficient for the condensing section was assumed to be 500 $Btu/hr-ft^2-^{o}F$. For one calculation, the other assumptions were: mass ratio of water to isobutane = 0.43, temperature of geothermal fluid = $400^{o}F$, mass fraction of vapor in geothermal fluid = 0.292, power cost = 1¢/Kwh, heating surface cost = $1.0/yr-ft^2$, condensing temperature = $120^{o}F$. It was found that the optimal temperature differences were $10^{o}F$ at the exit and $20^{o}F$ at the inlet. In another calculation, the geothermal fluid was assumed to be at $312^{o}F$, 10.3% quality, and there were no significant changes in the values of optimal temperature differences. Thus, it can be concluded that for preliminary design the suggested minimum temperature difference in the heat exchanger is in the range of 5 to $10^{o}F$ at a location

near the water exit when the geothermal fluid is pumped out
from a reservoir, and that the suggested temperature
differences at the exit and the inlet of a heat exchanger
are $10^{\circ}F$ and $20^{\circ}F$ respectively if the geothermal fluid
emerges from a well by natural forces. Additional heat
balances and economic analyses are recommended for every
specific situation.

The power output of the system is affected
significantly by the isobutane pressure. The optimum
pressures as a function of the water temperatures at the
wellhead were determined by heat balances as given in
Figure 1-3.

The equipment of regenerative isobutane system is
similar to that of basic system except the addition of
regenerative heat exchanger. The designs of regenerative
heat exchanger were studied, and the method of Briggs et al.
(6) was found suitable for calculating the overall heat
transfer coefficient and the pressure drops in the shell
and tubes. For a hypothetical plant which is supplied with
$600^{\circ}F$ saturated geothermal liquid, the overall heat transfer
coefficient was found to be 17.5 $Btu/hr\text{-}ft^{2}\text{-}^{\circ}F$. In the
conceptual design of a 10 MW plant, one heat exchanger of
14 ft. tube length was used. It has 4 shell-passes and
8 tube-passes as shown in Figure 3-4. Aluminum alloys
were suggested for the tubes and the fins of the heat
exchanger since they do not react chemically with isobutane

and are cheaper than copper. The size of heat exchanger
may be reduced by applying the techniques of optimization.
Although the design is conservative, it does demonstrate
the technical feasibility of constructing a large
regenerative heat exchanger, and it furnishes a basis for
economic evaluation.

At the end of this study, a conceptual design was
made for a 10 MW isobutane regenerative plant which uses
$600^{O}F$ saturated geothermal liquid as the energy source.
The flow diagram of the plant is given in Figure 4-1 to
show the arrangement of counterflow heat exchangers,
turbines, knockout drums, shell-and-tube condenser,
isobutane feed pumps, cooling tower, regenerative heat
exchanger, and underground receiver. The capital cost of
the plant is estimated at $10,915,000. As compared to the
estimated cost of the basic isobutane plant, the adoption
of the regenerative system can definitely be justified if
the power cost is 2¢/Kwh or higher.

REFERENCES

1. Ahluwalia, R.K., M.S. Thesis, University of Hawaii, December 1974.

2. American Society of Heating, Refrigeration and Air-Conditioning Engineers Handbook of Fundamentals, American Society of Heating, Refrigeration and Air-Conditioning Engineers, Incorporated, New York, 1972, p. 629.

3. Austin, A.L., G.H. Higgins and J.H. Howard. The Total-Flow Concept for Recovery of Energy from Geothermal Hot Brine Deposits, UCRL-51366, Lawrence Livermore Laboratory, University of California, April 1973.

4. Bechtel Corporation, Electric Power Generation Using Geothermal Brine Resources for a Proof-of-Concept Facility, A report submitted to Energy Research and Development Administration, May 1975.

5. Bell, K.J. Final Report of the Co-operative Research Program on Shell and Tube Heat Exchangers, Bulletin No. 5, University of Delaware Engineering Experimental Station, Delaware, January 1963.

6. Briggs, D.E., D.L. Katz and E.H. Young. "How to Design Finned-Tube Heat Exchangers", Chemical Engineering Progress, Vol. 59, No. 11, November 1963, p. 49.

7. Chou, J.C.S. and R.K. Ahluwalia. Characteristics of Vapor-Flashing Geothermal Plant, Hawaii Geothermal Project, Technical Report No. 6, University of Hawaii, November 1974.

8. Chou, J.C.S., R.K. Ahluwalia and E.Y.K. Woo. "Regenerative Vapor Cycle with Isobutane as Working Fluid", Geothermics, Vol. 3, No. 3, 1974.

9. Dunkelberg, Stephen B. "A Dimensionless Method for Determining Shell-Tube Heat Exchanger Tube-Sheet Geometry", Journal of American Society of Heating, Refrigeration and Air-Conditioning Engineers, Vol. 4, December 1962, p. 48.

10. Engineering Standards, Philips Petroleum Company, May 1967.

11. Fraas, Arthur P. and M. Necati Ozisik. Heat Exchanger
    Design, John Wiley and Sons, New York, 1965, p. 141.

12. Holt, B., A.J.L. Hutchinson and Douglas S. Cortez.
    "Geothermal Power Generation Using the Binary
    Cycle", Geothermal Energy Magazine, Vol. 1, 1973,
    p. 43.

13. James, R. "Optimum Wellhead Pressure for Geothermal
    Power", New Zealand Engineering, Vol. 22, 1967,
    p. 221.

14. Kuong, J.F. Applied Nomography, Vol. 2, Gulf
    Publishing Company, Texas, 1968.

15. Kruger, P., C. Otte (editors). "The Vapor-Turbine
    Cycle for Geothermal Power Generation", Geothermal
    Energy, Stanford University Press, Stanford,
    California, 1973, Chapter 8, p. 163.

16. Peters, M.S. and Klaus D. Timmerhaus. Plant Design
    and Economics for Chemical Engineers, McGraw-Hill
    Book Company, New York 1968, p. 281.

17. Popper, Herbert and the Staff of Chemical Engineering.
    Modern Cost-Engineering Techniques, McGraw-Hill
    Book Company, New York, 1970.

18. Robinson, K.K. and D.E. Briggs. "Pressure Drop of Air
    Flowing Across Triangular Pitch Banks of Finned
    Tubes", Chemical Engineering Progress Symposium
    Series, Vol. 62, 1962, p. 2.

19. Standards of Tubular Exchanger Manufacturers
    Association, Tubular Exchanger Manufacturers
    Association, New York, Fifth Edition, 1968.

20. Weiss, H., R. Steidel and A. Lundberg. Performance
    Test of a Lysholm Engine, UCRL-51861, Lawrence
    Livermore Laboratory, University of California,
    July 1975.