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REPORT OU/ID-1719-3

DEVELOPMENT OF GEOTHERMAL BINARY
CYCLE WORKING FLUID PROPERTIES
INFORMATION AND ANALYSIS OF CYCLES

Annual Report

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ABSTRACT

The research discussed in this report was performed at the University of Oklahoma during the period January 1, 1979 through December 31, 1979. Efforts were directed principally to the following tasks: (1) comparisons of mixture and pure fluid cascade cycles, (2) development of guidelines for working fluid selection for single boiler cycles, (3) continued evaluation of mixtures as working fluids, (4) working fluid thermophysical property correlation and presentation of properties information.

TABLE OF CONTENTS

	<u>PAGE</u>
ABSTRACT	i
1. INTRODUCTION AND OVERALL SUMMARY	1
2. COMPARISONS OF MIXTURE AND PURE FLUID CASCADE CYCLES	3
3. GUIDELINES FOR SELECTION OF WORKING FLUIDS FOR GEOTHERMAL BINARY CYCLES	4
4. CONTINUED EVALUATION OF MIXTURES AS WORKING FLUIDS IN GEOTHERMAL BINARY CYCLES	4
5. IDENTIFICATION OF GEOTHERMAL INDUSTRY NEEDS FOR WORKING FLUID THERMODYNAMIC AND TRANSPORT PROPERTY DATA.	5
6. DEVELOPMENT AND PRESENTATION OF THERMODYNAMIC AND TRANSPORT PROPERTY INFORMATION FOR SELECTED WORKING FLUIDS OF INTEREST FOR GEOTHERMAL CYCLES	6
7. REFERENCES	7
APPENDIX I	I-1
APPENDIX II	II-1
APPENDIX III	III-1

1. INTRODUCTION AND OVERALL SUMMARY

The majority of the work discussed in this report was performed at the University of Oklahoma during the period January 1, 1979 through December 31, 1979. Efforts were directed to the following tasks:

- (1) Comparisons of Mixture and Pure Fluid Cascade Cycles
- (2) Development of Guidelines for Working Fluid Selection for Single Boiler Cycles
- (3) Continued Evaluation of Mixtures as Working Fluids
- (4) Identification of Working Fluid Thermophysical Property Data Needs
- (5) Working Fluid Thermophysical Property Correlation and Presentation of Properties Information

The initial comparisons of mixtures and pure fluids in cascade cycles (task 1) were summarized in a report to DOE-Idaho project personnel in October, 1978. This work indicated that for low temperature georesources such as Raft River, dual boiler mixture cycles may achieve resource utilization levels as great as triple boiler pure fluid cycles. This area of study has been further addressed in the M.S. thesis research of C.T. Chu (Reference 1) and is discussed in some detail in Appendix I. The subject of development of guidelines for working fluid selection for single boiler cycles (task 2) was addressed in a paper presented at the Thirteenth Intersociety Energy Conversion Conference in San Diego in August, 1978 (Reference 2). This work indicated there may be numerous fluids as good or better than isobutane in

geothermal binary cycles. A total of 100 fluids have been considered in the more recent research. This work has been further addressed in the M.S. thesis research of S. Ramaswamy (Reference 3) and is summarized in Appendix II. The evaluation of mixtures as working fluids in geothermal binary cycles (task 3), has been a continuing study at the University of Oklahoma since 1973.

This work has indicated that if countercurrent flow can be achieved in heat exchangers, mixtures have definite advantages over pure fluids in subcritical binary cycles. Discussion of thermodynamic, economic and equipment considerations has been presented by K.Z. Iqbal in his Ph.D. thesis (Reference 4); portions of this thesis are reproduced in Appendix III. Task 4 is the continuation of an effort initiated in 1976 to identify working fluid thermophysical property data needs. A cooperative effort among different geothermal research groups led to an identification of isobutane thermodynamic properties data needs which in turn led to an experimental data procurement program at the National Bureau of Standards, in 1978. Our more recent efforts have been directed to identifying isopentane thermodynamic data needs and identifying viscosity and thermal conductivity data needs for isobutane and isopentane. Also, thermodynamic property data needs were considered for the isobutane-isopentane mixture originally considered for the proposed Heber plant. Task 5 has involved the determination of correlation parameters for the MBWR equation of state for isobutane and isopentane and parameters in correlations of viscosity and thermal conductivity for isobutane and viscosity for isopentane. Because thermal conductivity data for isopentane in the

region of cycle operating conditions are not available, a generalized correlation was used to provide a reasonable means for estimating isopentane thermal conductivity. The data judged to be the best available have been correlated for isobutane and isopentane. The results of the efforts on the determination of data needs and correlation of working fluid thermophysical properties were delayed somewhat so that pressure-enthalpy thermodynamic properties diagrams in SI units for a number of hydrocarbon working fluids could be prepared and included in Report OU/ID-1719-4 (Reference 5).

2. COMPARISONS OF MIXTURE AND PURE FLUID CASCADE CYCLES

It is well known that compared to single boiler cycles, the resource utilization of low temperature Rankine cycles can be improved by using cascade cycles. In this study, a parametric analysis of cascade cycles was performed. Isobutane and several mixtures were employed as working fluids in the analysis. This study shows that for resource temperatures of 300°F and below, the dual boiler cycle has 20% greater resource utilization than the conventional single boiler cycle. The improvement in resource utilization from the dual to the triple boiler cycle is only about 5%. For a low temperature resource such as 250°F, dual and triple boiler cycles have lower cost per KW power output than the single boiler cycle. A mixture dual boiler cycle yields 6% greater resource utilization than a pure dual boiler cycle, but the capital investment in \$/kw is about the same. The M.S. thesis research of C.T. Chu (Reference 1), addresses the subject of the use of mixtures in cascade cycles. A summary of this research is presented in Appendix I.

3. GUIDELINES FOR SELECTION OF WORKING FLUIDS FOR GEOTHERMAL BINARY CYCLES

A procedure for working fluid selection for single boiler geothermal binary cycles has been developed. One hundred fluids were screened to find fluids with the best properties for Rankine cycle power recovery using heat available at 300-500°F. The first phase of the two phase screening procedure permits a quick evaluation, with potential working fluids identified for more detailed analysis. The second phase screening procedure allows one to rank potential working fluids on the basis of economics merely from knowledge of a few characterization parameters. These parameters are vapor pressure, specific volume and I-Factor for the saturated vapor at 100°F. From this study, it is concluded that the best working fluids are the fluids which have moderate vapor pressure and specific volume of saturated vapor at 100°F and an I-factor at 100°F close to 1.0. These fluids include 1-2-butadiene, 1-3-butadiene, cis-2 butene and trans-2 butene. This work, which was the M.S. thesis research of S. Ramaswamy (Reference 3), is summarized in Appendix II.

4. CONTINUED EVALUATION OF MIXTURES AS WORKING FLUIDS IN GEOTHERMAL BINARY CYCLES

Although a considerable amount of work has been carried out at the University of Oklahoma since 1973 on the subject of mixture working fluid geothermal binary cycles, the Ph.D. thesis of K.Z.

Iqbal (Reference 4), provides valuable elucidation regarding mixture cycles. Pages 175-238 of K.Z. Iqbal's thesis are reproduced in Appendix III to provide a summary of the conclusions from this work regarding the advantages and disadvantages of mixtures compared to pure fluids in geothermal cycles.

5. IDENTIFICATION OF GEOTHERMAL INDUSTRY NEEDS FOR WORKING FLUID THERMODYNAMIC AND TRANSPORT PROPERTY DATA

Although most of this work was performed prior to 1979, much of the effort has not been reported previously and so a review is appropriate here. This task involved searching the literature for data, evaluation of the available data and recommendation of the data procurement and correlation efforts needed to meet geothermal industry needs. This work was carried out as a cooperative effort with other centers for geothermal research, principally Brown University, the Idaho National Engineering Laboratory and other Department of Energy offices and laboratories, the National Bureau of Standards and the Electric Power Research Institute. The working fluids considered to be prime candidates for geothermal binary cycles in 1978 were emphasized in this effort. The data considered were thermodynamic property data, viscosity data and thermal conductivity data. Two fluids were considered in some detail, isobutane and isopentane. Isobutane is the working fluid for the first 5MW Raft River power plant, while isopentane is a prime candidate working fluid for the second Raft River 5MW power plant, which will use direct contact heat exchange. Some work also was done to evaluate thermodynamic property data needs for an isobutane-isopentane mixture considered as the working fluid in design

studies for the 50MW Heber power plant. However, the effort for this mixture was stopped when plans for the Heber plant became uncertain early in 1978. The effort to identify data needs for these fluids are presented, along with correlations of the data, in Report OU/ID-1719-4 (Reference 5).

6. DEVELOPMENT AND PRESENTATION OF THERMODYNAMIC AND TRANSPORT PROPERTY INFORMATION FOR SELECTED WORKING FLUIDS OF INTEREST FOR GEOTHERMAL CYCLES

The objective of this effort has been to provide the geothermal industry with useful thermophysical properties information for selected working fluids. The major focus in this work was on the working fluids isobutane and isopentane. For both fluids, the thermodynamic properties data which are considered to be the most accurate available have been correlated using the MBWR (modified Benedict-Webb-Rubin) equation of state. Viscosity data for both fluids and thermal conductivity data for isobutane were correlated using correlations recently developed at the University of Oklahoma. Because of a lack of experimental thermal conductivity data for isopentane, a generalized correlation was used to generate an isopentane thermal conductivity tabulation. This generalized correlation, which was developed recently at the University of Oklahoma, is a three parameter corresponding states correlation. It is expected to be accurate for the thermal conductivity of isopentane within about 5% for most temperature-pressure conditions encountered in geothermal cycles (excluding the critical region).

APPENDIX I

IMPROVEMENT OF RESOURCE UTILIZATION FOR LOW TEMPERATURE RANKINE CYCLES USING CASCADE CYCLES

IMPROVEMENT OF RESOURCE UTILIZATION FOR
LOW TEMPERATURE RANKINE CYCLES USING
CASCADE CYCLES

by

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and
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ABSTRACT

The resource utilization of low temperature Rankine cycles can be improved by using cascade cycles. A parametric analysis of cascade cycles was performed. Guidelines for selecting the working fluid for low temperature Rankine cycles are also discussed. Isobutane and several mixtures were employed as working fluids in the analysis. This study shows that for resource temperatures of 300°F and below, the dual boiler cycle has 20% greater resource utilization than the conventional single boiler cycle. The improvement in resource utilization from the dual to the triple boiler cycle is only about 5%. For a low temperature resource such as 250°F, dual and triple boiler cycles have lower cost per KW power output than the single boiler cycle. A mixture dual boiler cycle yields 6% greater resource utilization than a pure dual boiler cycle, but the capital investment in \$/kw is about the same.

INTRODUCTION

Since the price of energy is going up rapidly, the efficient use of all energy resources is becoming more important. Some low temperature energy resources such as geothermal energy, solar energy and waste heat energy are being actively considered for widespread development in the United States and in many other countries throughout the world. One way to utilize these low temperature energy resources is to produce useful electrical or shaft power using a Rankine cycle system. However, the efficiency of this energy conversion process is limited by the temperature of the resource and by prevailing ambient conditions for heat rejection. Moreover, the investment in obtaining the energy resource can be quite large. These factors create strong incentives for optimizing the performance of any power conversion scheme involving these resources.

In general, increasing the turbine inlet pressure in a Rankine cycle will increase the turbine work, and hence the thermal efficiency of the cycle. However, the working fluid bubble point temperature in the boiler is also increased because of the increase in pressure. As a result, the resource fluid exits from the boiler at an elevated temperature and available energy in the resource is wasted. However, if a second cycle is added to utilize the previously wasted energy, the resultant dual boiler cycle will be more efficient than a conventional single boiler Rankine cycle with respect to resource utilization.

There are different ways to arrange the cascade cycle. A double boiling geothermal binary cycle⁽¹⁾, studied by Ingvarsson

and Madsen, shows great improvement in resource utilization from a single boiling binary cycle for geothermal fluid temperatures of 320°F and below. A dual cycle arrangement and a topping/bottoming cycle are believed to yield substantial increases in conversion efficiency for relatively high temperature resources, for example, $536^{\circ}\text{F}^{(2)}$.

Figure 1 shows the dual boiler cycle arrangement that is studied herein. Since the resource temperature range considered is 250° to 350°F , the configuration of the cycle is similar to that of Ingvarsson's double boiling binary cycle. The working fluid enters boiler 2, where it is partially boiled by heat exchange with the resource fluid exiting from high pressure boiler 1. The working fluid leaves boiler 2 as a mixture of vapor and liquid. The low pressure vapor comes out the top of the separator and is expanded through low pressure turbine 2. The liquid part of the working fluid leaves the bottom of the separator, is pumped to a higher pressure, and then passes through high pressure boiler 1. The working fluid in boiler 1 is heated to saturated or superheated vapor by the incoming resource fluid. This high pressure vapor is then expanded in turbine 1 to generate work. The two outlet streams of working fluid from the turbines are condensed to saturated liquid by the cooling water in two separate condensers. Both streams of liquid working fluid are compressed by pumps 2 and 3 to the inlet pressure of boiler 2, are then combined and enter boiler 2 to complete the cycle. The configuration of a triple or quadruple boiler cycle is essentially the same as the dual boiler cycle.

Figure 1 Dual Boiler Rankine Cycle

One-pass tube and shell heat exchangers are used as boilers and condensers in this study. The turbine-generator and pump efficiencies are assumed to be 86% and 85%, respectively. The geothermal fluid, or simply the brine, is employed as the low temperature energy resource.

A guideline for selecting working fluids for low temperature Rankine cycles is discussed herein. Isobutane is chosen as the working fluid. A parametric analysis is performed in order to find the optimum operating conditions for dual and triple boiler cycles. The thermal efficiency and the resource utilization of cascades cycles are then compared with those of a conventional single boiler Rankine cycle. The capital investment for the cascade cycles is also estimated and compared to that of a single boiler cycle. The use of mixtures as working fluids in the low temperature Rankine cycle is believed to have some advantages over pure fluids^(3,4,5). Several binary mixtures are studied for the dual boiler cycle. Mixture cycle performances are compared with the pure isobutane dual boiler cycle.

A mixture thermodynamic properties computer program, HSGC⁽⁶⁾, a geothermal binary cycle simulator, GEO4⁽⁷⁾, and a cascade cycle calculation program⁽⁸⁾ were employed to perform the calculations in this study.

WORKING FLUID SELECTION

Working fluid selection usually involves a compromise between thermal efficiency, heat transfer properties, chemical and thermal stability, toxicity, flammability, pressure and cost. Considerable

work has been done on methods for selecting working fluids for the low temperature Rankine cycle system^(9,10,11,12,13). A large number of organic compounds can be used as working fluids in the Rankine cycle. Because of the lack of thermodynamic properties for many fluids and the complexity of selecting an optimum working fluid, only a few working fluids have been examined in detail to date.

To determine an optimum working fluid, both thermodynamic and economic analyses ought to be performed. However, before getting into complicated analysis, a preliminary screening is needed to narrow the study to a few potential working fluids. Criteria for the preliminary screening process are discussed below.

Vapor Pressure

Heat exchanger cost per unit heat transfer area is an increasing function of the operating pressure. That the working fluid pressures in both the boiler and the condenser be as low as possible is desired. But, a pressure lower than atmospheric pressure generally should be avoided. Any fluid system operating below atmospheric pressure is likely to have maintenance trouble.

Shape of the Vapor Saturation Curve

The states of working fluid at both the turbine inlet and outlet are dependent on the slope of the saturated vapor locus on a temperature-entropy diagram. A working fluid whose saturated vapor curve is vertical would be more likely to give a high efficiency cycle and to require less expensive equipment.

Kihara and Fukunaga⁽¹⁰⁾ defined a dimensionless parameter, the I-factor, to measure the slope of vapor saturation curve on a T-S diagram as follows:

$$I = 1 - \left(\frac{ds}{dT} \frac{T}{C_p} \right)_{\text{sat. vap.}} = 1 - T \left(\frac{ds}{dH} \right)_{\text{sat. vap.}} \quad (1)$$

The I-factor is unity for a fluid with vertical saturated vapor locus on a T-S diagram. If an isentropic expansion is assumed in the turbine, there would be no superheat at either end of the turbine for a fluid with $I=1$. If $I<1$, the turbine exhaust will be superheated. The superheat rejected in the condenser is, in part, lost work which was available to the cycle. For a fluid with $I>1$, the turbine exhaust could be in the two phase region. The wetness of the exhaust causes not only the maintenance problem created by liquid impingement erosion but also a decrease in the turbine efficiency.

Critical Temperature

From the standpoint of pumping power requirements, it is undesirable to have the whole cycle operating in the superheated vapor phase. The critical temperature of the working fluid therefore must be higher than the assumed condensing temperature (100°F).

Latent Heat of Vaporization

For specified boiling temperature and brine flow rate, a fluid with a small heat of vaporization leads to a low brine exit temperature. For fixed brine flow rate, the lower the brine exit temperature, the more energy is extracted from the brine.

Specific Volume of Saturated Vapor

The size of turbine is determined by the specific volume of the working fluid at the turbine exhaust. Usually, in a low temperature Rankine cycle, the turbine exhaust condition is near to the dew point at the condensing temperature. Therefore, a fluid with a small saturated vapor specific volume at the condensing temperature leads to a small, and hence inexpensive turbine.

Stability and Safety Considerations

A practical working fluid should be stable, essentially non-fouling and noncorrosive. The temperature at which the fluid starts thermal decomposition must be higher than the maximum operating temperature of the cycle.

Since the cycle working fluid is in a closed system, the requirements regarding flammability and toxicity are less rigid. However, fluids that are rated as severely toxic should be eliminated.

R. Subramanian (14) has developed a fluid screening process which uses the above mentioned criteria to select working fluids for the single boiler geothermal binary cycle. 1,2-Butadiene, cis-2-butene, 1,3-butadiene, etc. are suggested as good working fluids on the basis of economics. However, if resource utilization is of more concern than economics, isobutane may be the first choice because of the small latent heat of vaporization of isobutane at the assumed boiling temperature (210°F). Table 1 lists some important thermodynamic characterization parameters for several Rankine cycle working fluids.

TABLE 1

POTENTIAL WORKING FLUIDS FOR MULTIPLE BOILER RANKINE CYCLES

Fluid	$T_c, ^\circ\text{F}$	P_c, psia	$P_{v100^\circ\text{F}}$	I-factor	$V_{100^\circ\text{F}}$	$\Delta H_{110^\circ\text{F}}^V$	$\Delta H_{210^\circ\text{F}}^V$	ΔH_t^*
R-12	233.6	597.0	131.9	0.91	0.308	51.81	30.44	6.6
R-11	390.0	639.4	23.6	1.05	1.764	74.44	65.64	10.0
R-114	294.3	473.2	46.4	0.65	0.695	52.07	39.56	8.0
R-113	419.0	495.4	10.5	0.78	2.980	63.71	56.72	9.6
R-142b	279.0	598.2	71.5	1.24	0.724	80.91	67.16	9.9
Propane	206.3	617.4	188.3	1.32	0.559	132.86	61.19 [‡]	17.7
1,2-Butadiene	340.0	652.7	36.0	0.99	2.880	180.64	150.36	25.9
1,3-Butadiene	305.0	627.7	59.6	1.00	1.700	158.93	123.55	23.4
1-Butene	295.6	583.5	60.9	0.97	1.569	148.53	114.74	21.8
Cis-2-Butene	324.4	610.0	46.1	1.04	2.125	160.42	131.07	23.4
Trans-2-Butene	311.8	595.4	49.8	0.93	2.000	157.33	124.66	23.4
n-Butane	305.7	551.7	51.4	0.75	1.810	147.84	113.89	23.0
Isobutane	274.9	529.1	72.0	0.81	1.260	133.75	99.48	19.9
Cyclopentane	460.0	654.1	10.0	1.02	8.300	169.35	154.16	23.0
1-Pentene	376.9	590.0	19.1	0.76	5.370	150.26	128.46	23.0
n-Pentane	385.7	489.6	15.6	0.70	5.100	152.25	129.47	24.2
Isopentane	369.0	490.0	20.4	0.77	3.850	143.05	121.43	22.1

+ $V_{100^\circ\text{F}}$: specific volume of saturated vapor at 100°F , Cu ft/lb; $\Delta H_{110^\circ\text{F}}^V$, $\Delta H_{210^\circ\text{F}}^V$: latent heat of vaporization at 110° and 210°F , Btu/lb.

* ΔH_t : specific enthalpy change of isentropic expansion from 210°F to 110°F , Btu/lb.

‡ Latent heat of vaporization at 190°F .

No pure fluid can fulfill all the requirements of an ideal working fluid for the Rankine cycle. However, some drawbacks of certain pure fluids can be overcome by adding one or more other fluids to form a mixture. For example, if a fluid with an I-factor less than one is mixed with a fluid with an I-factor larger than one, the result generally will be a mixture with an I-factor closer to one. Likewise, the vapor pressure of one fluid generally will be lowered by adding a fluid with a lower vapor pressure. One significant characteristic of a mixture is its nonisothermal boiling behavior. A mixture will give a higher turbine inlet temperature than a pure fluid for the same bubble point temperature in the boiler. A higher turbine inlet temperature usually results in greater turbine work and hence a higher cycle efficiency.

THERMAL EFFICIENCY AND RESOURCE UTILIZATION OF CASCADE CYCLES

The thermal efficiency of a Rankine cycle, η_N , is defined by the relation,

$$\eta_N = \frac{\dot{W}_N}{\dot{Q}_H} \times 100\%$$

where $\dot{W}_N$ is the net cycle power and $\dot{Q}_H$ is the total heat input rate in the boiler. The thermal efficiency is only a measure of how efficiently the transferred heat is converted into work; it does not indicate how much available energy in the resource is extracted and converted into useful work. Another quantity, the net cycle work per pound of brine, is used here as the measure of resource utilization for the Rankine cycle. A specified brine flow rate is used in this study as the basis for calculation. Thus, to increase

the net work output by using a cascade cycle is our main objective.

The following operating conditions need to be considered in order to find the resource utilization optimum of a cascade cycle:

- (1) Vapor-liquid split at the LP (Low Pressure) boiler outlet.
- (2) Turbine inlet temperature and pressure in each cycle.
- (3) The ranges of the cooling water temperature rise in the condensers.
- (4) Pinch point temperature differences in both boilers and condensers.

From previous research⁽⁴⁾, it is known that the operating conditions for a resource utilization optimized Rankine cycle are close to those of a cost optimized Rankine cycle. Therefore, the cost optimized single boiler Rankine cycle⁽³⁾, is used as a base case to compare with the cascade cycles considered. Thus, the brine flow rate and the cooling water temperature rise for the cost optimized 25 MW single boiler Rankine cycle are taken as the basis for the cascade cycle calculations.

Theoretically, only when the temperature difference between the heating and cooling fluids in a heat exchanger is infinitesimal is the heat transfer reversible. A high thermal efficiency results from reversible heat transfer. However, in order to operate the heat exchanger reversibility, an infinitely large heat exchanger would be required, which cost an infinite amount of money. Therefore, a 10°F minimum pinch point temperature difference is set for both boilers and condensers in order that the heat exchangers be reasonably scaled and priced.

A sensitivity analysis is performed to determine the optimal working fluid vapor-liquid split and turbine inlet conditions for

Figure 2 Variation of Resource Utilization and Thermal Efficiency of a Dual Boiler Cycle with Vapor-Liquid Split

the cascade cycle which gives the highest resource utilization for a given brine temperature. Figure 2 shows the sensitivities of thermal efficiency and resource utilization for a dual boiler cycle to different working fluid vapor-liquid splits at the LP boiler outlet. The brine temperature is 250°F and the HP (high pressure) and LP turbine inlet temperatures are 200° and 175°F , respectively. The thermal efficiency of the dual boiler cycle decreases as the working fluid vapor mass fraction increases. Since the high pressure cycle is thermodynamically more efficient than the low pressure cycle, the thermal efficiency decreases as more working fluid is channeled to the low pressure cycle. Zero vapor mass fraction implies a single boiler cycle with all the working fluid routed to the HP turbine; and this vapor-liquid split therefore has the highest thermal efficiency. The variation of resource utilization with the vapor-liquid split is shown in Figure 2 by the two curves meeting at a point. The first curve, which starts from the vapor mass fraction of zero, is obtained when the working fluid flow rate is controlled by the HP boiler minimum pinch point temperature difference. As the vapor mass fraction increases, the amount of working fluid flowing through low pressure turbine increases and the pinch point temperature difference in the LP boiler decreases. The second curve, which intersects with the first one, shows the resource utilization of the dual boiler cycle when the LP boiler minimum pinch point temperature difference controls the working fluid flow rate. The intersection point occurs when the pinch point temperature differences in both boilers reach the minimum (10°F).

Dual boiler cycles with brine temperatures of 250° , 300° and 350°F were studied for different pairs of HP and LP turbine inlet

conditions. Complete vapor-liquid split sensitivity curves can be found in Reference 8. Figures 3 and 4 show the resource utilization of dual boiler cycles with different turbine inlet conditions for 250° and 300°F brine, respectively. The points on these curves were obtained from the highest resource utilization from each vapor-liquid sensitivity curve. The optimal pair of turbine inlet temperatures are 200° and 155°F for 250°F brine, and 270° and 185°F for 300°F brine.

The resource utilization and thermal efficiency of a dual boiler cycle with 350°F brine is shown in Figure 5. The highest values of both resource utilization and thermal efficiency occur at zero vapor mass fraction. This indicates that, for a brine temperature of 350°F, the single boiler cycle is more efficient than the dual boiler cycle in both resource utilization and thermal efficiency.

The determination of optimal operating conditions for a triple boiler cycle is more complicated than for a dual boiler cycle. Only one set of turbine inlet temperatures is studied here. A comparison of the performance of single, dual and triple boiler cycles is presented in Table 2 for 250°F brine, and in Table 3 for 300°F brine. The thermal efficiencies of dual and triple boiler cycles are within the same range as those of single boiler cycles. However, since the heat extracted from the brine by either a dual or triple boiler cycle is greater than for a single boiler cycle, the resource utilization of a cascade cycle is much higher than that of a single boiler cycle. The resource utilization of a dual boiler cycle is about 20% higher than that of a single boiler cycle. The improvement in resource utilization from a single to a triple

Figure 3 Resource Utilization of a Dual Boiler Cycle vs. LP Turbine Inlet Temperature for Different HP Turbine Inlet Temperatures

Figure 4 Resource Utilization of a Dual Boiler Cycle vs. LP Turbine Inlet Temperature for Different HP Turbine Inlet Temperatures

Figure 5 Variation of Resource Utilization and Thermal Efficiency of a Dual Boiler Cycle with Vapor-Liquid Split

TABLE 2
COMPARISON OF CASCADE CYCLES WITH A
SINGLE BOILER CYCLE FOR 250°F BRINE

Brine Temperature = 250°F

Working Fluid = Isobutane

Brine Flow Rate = 1.5×10^7 lb/hr

Parameters	Single Boiler Cycle	Dual Boiler Cycle	Triple Boiler Cycle
Turbine Inlet Temperature, °F	180.0	200.0 160.0	220.0 180.0 150.0
Condensing Temperature, °F	108.0	108.0	108.0
Brine Exit Temperature, °F	165.1	149.2	146.3
Working Fluid Flow Rate, 10^6 lb/hr	8.24	9.74	10.01
Total Boiler Duty, 10^8 Btu/hr	12.74	15.12	15.56
Total Condenser Duty, 10^8 Btu/hr	11.69	13.85	14.24
Gross Power, MW	32.42	39.86	42.03
Net Power, MW	30.52	37.29	39.15
Net Thermal Efficiency, %	8.18	8.42	8.59
Resource Utilization, Btu/lb Brine	6.94	8.49	8.91
Total Plant Cost, \$/kw	905.05	955.84	1000.86
Brine System Cost, \$/kw	954.39	781.12	744.01
Total System Cost, \$/kw	1859.44	1736.96	1744.87

TABLE 3
COMPARISON OF CASCADE CYCLES WITH A
SINGLE BOILER CYCLE FOR 300°F BRINE

Brine Temperature = 300°F

Working Fluid = Isobutane

Brine Flow Rate = 0.7040×10^7 lb/hr

Parameters	Single Boiler Cycle	Dual Boiler Cycle	Triple Boiler Cycle
Turbine Inlet Temperature, °F	220.0	270.0	270.0
		190.0	210.0
			150.0
Condensing Temperature, °F	108.0	108.0	108.0
Brine Exit Temperature, °F	170.1	153.9	137.3
Working Fluid Flow Rate, 10^6 lb/hr	5.59	6.28	7.07
Total Boiler Duty, 10^8 Btu/hr	9.15	10.28	11.45
Total Condenser Duty, 10^8 Btu/hr	8.14	9.10	10.20
Gross Power, MW	31.87	38.94	41.11
Net Power, MW	29.44	34.95	36.81
Net Thermal Efficiency, %	10.99	11.60	10.97
Resource Utilization, Btu/lb Brine	14.27	16.94	17.85
Total Plant Cost, \$/kw	755.85	848.68	925.34
Brine System Cost, \$/kw	464.97	391.67	371.88
Total System Cost, \$/kw	1220.82	1240.35	1297.22

boiler cycle is around 25%.

CAPITAL COST ESTIMATION FOR CASCADE CYCLES

The total cost of a Rankine cycle can be estimated from the cost of the major process equipment in the cycle. This study uses the factored estimate method for a geothermal binary cycle power plant first described by Miloa and Tester⁽²⁾ and later modified^(3,15). This method requires the determination of the major equipment cost as well as geothermal well cost. Other direct or indirect costs for the power plant are estimated as percentages of the major equipment and well costs. All cost estimations in this research are based on 1976 U.S. dollars.

The major component costs of single, dual and triple boiler cycles with 250° and 300°F brine are shown in Figures 6 and 7, respectively. The total plant costs, the brine system costs, and the total system costs are listed in Tables 2 and 3.

The boiler cost and the condenser cost account for the greatest part of the total plant cost. It is possible to say, therefore, that the total plant cost is mainly controlled by the costs of these two pieces of equipment. As can be found in Tables 2 and 3, the boiler duties for the dual boiler cycles are much larger than for the single boiler cycle. In addition, the average temperature differences between the brine and the working fluid in the dual and triple boiler cycles are less than in the single boiler cycle. Thus, the boiler heat transfer area for the dual boiler cycle is much greater than for the single boiler cycle, which results in a higher boiler cost.

Figure 6 Major Component Costs of Single, Dual, and Triple Boiler Rankine Cycles for 250°F Brine with Isobutane as the Working Fluid

Figure 7 Major Component Costs of Single, Dual, and Triple Boiler Rankine Cycles for 300°F Brine with Isobutane as the Working Fluid

The condenser cost, on the other hand, decreases with an increase in the number of boilers. This is because the net power output increases more rapidly than the condenser duty. Although the total cost of the condenser increases, the cost per KW power output decreases. For 300°F brine, however, the condenser cost for the triple boiler cycle is higher than that for the dual boiler cycle because the increase in total condenser cost when changing from the dual to the triple boiler cycle is greater than the increase in net power output resulting from the change.

The costs of other pieces of equipment, though varying with the number of boilers, do not change the total plant cost significantly. Turbine-generator costs and pump costs increase with the change from single to triple boiler cycles since the turbine power and the pumping power increase. The change in the cost of the cooling system follows the same pattern as the change in the cost of the condenser.

The total plant cost goes up as the number of boilers increases. The rate of increase in the total plant cost in dollars per KW power output is around 5% per boiler for 250°F brine and 10% per boiler for 300°F brine. The cost of the resource system (e.g., wells for geothermal power) is a quite substantial factor to consider in the design of a power generation system. As the cost of the resource system is added to the plant cost, the picture of the total system cost changes. When the brine temperature is 250°F, the total system cost for the dual boiler cycle is the least expensive of the three cycles considered. Although the total system cost for the triple boiler cycle is still the highest with 300°F brine, the difference in total system cost between each cycle is

less than the difference in plant cost for each cycle. It should also be noted that for 250°F brine, the brine system cost is very close to the plant cost. The high brine system cost explains the importance of maximizing the resource utilization for low temperature energy resources.

MIXTURES AS WORKING FLUIDS

Five mixture dual boiler cycles are compared with a pure isobutane cycle at a brine temperature of 300°F in Table 4. Excluding the propane-isopentane cycle, the thermal efficiency of all dual boiler cycles is around 11%. For cycles that have the same value of thermal efficiency, the brine exit temperature can be used to determine whether the resource utilization of the cycle is high or low. The brine exit temperature for the propane-isobutane cycle is 132°F, the lowest of the six cycles; this cycle thus has the highest resource utilization. The brine exit temperature of the propane-isopentane cycle is close to that of the propane-isobutane cycle. However, the low thermal efficiency of this cycle hurts its resource utilization. The low thermal efficiency of the propane-isopentane cycle is due to the large slope of its boiling temperature profile. In the boiler, the large slope works to produce a low brine exit temperature and high turbine inlet temperature; in the condenser, however, it creates irreversibility in condensation, which leads to lower thermal efficiency.

Propane was chosen to mix with isobutane, isopentane, and cis-2-butene because of its low critical temperature. The addition of propane decreases the latent heat of vaporization of the mixture in the boiler. In addition, the I-factor of propane is greater

TABLE 4
COMPARISON OF PURE ISOBUTANE AND MIXTURE DUAL BOILER CYCLES

Brine Temp.: 300°F

Brine Flow Rate: 0.70399×10^7 lb/hr

Working Fluid	Isobutane	0.5 Propane 0.5 Isobutane	0.5 Isobutane 0.5 Isopentane	0.97 Isobutane 0.03 n-Hexane	0.6 Propane 0.4 Cis-2-Butene	0.75 Propane 0.25 Isopentane
Vapor Fraction at Separator	0.375	0.041	0.506	0.435	0.349	0.209
Resource Utilization, Btu/lb Brine	16.94	18.32	17.71	18.02	17.96	16.05
Net Power, MW	34.95	37.79	36.53	37.17	37.05	33.11
Gross Power, MW	38.94	43.39	38.07	40.16	41.84	36.91
Cycle Pump Power, MW	3.99	5.60	1.54	2.99	4.79	3.80
Net Thermal Efficiency, %	11.60	10.91	11.31	11.72	10.97	9.70
Brine Exit Temperature, °F	153.9	132.0	143.3	146.2	136.3	134.6
Total W.F. Flow Rate, 10^6 lb/hr	6.277	7.280	5.830	6.220	6.500	6.370
Total Boiler Duty, 10^8 Btu/hr	10.285	11.827	11.021	10.828	11.521	11.652
Total Condenser Duty, 10^8 Btu/hr	9.095	10.536	9.781	9.562	10.272	10.530
High Pressure Cycle						
Turbine Inlet Temperature, °F	270.0	230.0	255.0	255.0	260.0	250.0
Pressure, psia	475.0	480.0	230.0	376.0	576.0	445.9
Turbine Enthalpy Change, Btu/lb W.F.	26.74	20.61	27.77	27.23	25.75	21.61
W.F. Flow Rate, 10^6 lb/hr	3.923	6.982	2.880	3.514	4.232	5.039
Boiler Latent Heat, Btu/lb W.F.	35.65	71.32	106.95	79.54	— *	106.19
Boiler Pinch Point T, °F	10.0	10.0	14.3	13.5	10.0	11.7
Condensing Temperature, °F	108.0	93.9	90.0	90.0	90.0	90.0
Low Pressure Cycle						
Turbine Inlet Temperature, °F	190.0	180.4	192.5	187.0	181.2	175.7
Pressure, psia	220.0	325.5	134.7	202.0	310.0	315.0
Turbine Enthalpy Change, Btu/lb W.F.	15.22	14.10	16.93	15.29	14.91	12.84
W.F. Flow Rate, 10^6 lb/hr	2.354	0.298	2.950	2.706	2.269	1.331
Boiler Enthalpy Change, Btu/lb W.F.	91.17	60.82	119.81	103.49	95.68	77.15
Boiler Pinch Point T, °F	10.3	10.5	9.8	10.1	9.7	10.2
Condensing Temperature, °F	108.0	95.0	90.0	102.4	90.0	90.0
Plant Cost, \$/KW	848.68	1322.39	895.60	839.91	1187.82	1060.63
Brine System Cost, \$/KW	391.67	362.23	374.73	368.27	369.47	413.43
Total System Cost, \$/KW	1240.35	1684.62	1270.33	1208.18	1557.29	1474.06

* supercritical cycle

than one. When propane is mixed with a fluid with an I-factor less than one, the I-factor of the mixture will be closer to unity. Unfortunately, the relatively high vapor pressure of propane makes the mixture cycle require very expensive heat exchangers and pumps, as shown in Figure 8. Although the small specific volume of saturated vapor for propane leads to a small, and hence less expensive turbine, the total plant costs for the three mixture cycles that use propane are much higher than the pure isobutane cycle, due to the expensive heat exchangers.

Unlike the propane-isopentane mixture, mixtures of 97 mol% isobutane and 3 mol% n-hexane and 50 mol% isobutane and 50 mol% isopentane give condensation temperature profiles that are almost parallel to the cooling water profile in the condenser (for the case of a 18°F cooling water rise used herein). When the working fluid temperature profile parallels the cooling water temperature profile in the condenser, the irreversibility in the condenser is reduced. The resource utilization for these two mixture cycles is, therefore, higher than for the pure isobutane cycle.

The vapor pressures of isopentane and n-hexane are quite low compared to that of isobutane. When a mixture of isobutane and one of these two fluids is employed as the working fluid, the operating pressure of the cycle becomes lower. The use of these mixtures therefore results in lower boiler and cycle pump costs. However, the specific volume of the saturated vapor for these two fluids is relatively large, so a larger and more expensive turbine is required.

Figure 8 Major Component Costs of Pure Isobutane and Mixture Dual Boiler Cycles

Almost all the condensers in the mixture cycles have smaller logarithmic mean temperature differences and smaller heat transfer coefficients than the isobutane cycle. In addition, the total condenser duties of all five mixture cycles are larger than the isobutane cycle. The condenser heat transfer areas for mixture cycles, therefore, are all larger than the heat transfer area for the isobutane cycle. This is why the condenser cost of the pure isobutane cycle is less than for any mixture cycle.

The total plant costs for isobutane-isopentane and isobutane-n-hexane cycles are close to the plant cost for the pure isobutane cycle. The resource utilizations for these two mixture cycles are about 5% higher than for the pure isobutane cycle. It is possible that these two mixture cycles would be cheaper than the isobutane cycle if the cost optimization were performed.

CONCLUSIONS

- (1) At a brine temperature of 350°F , the isobutane single boiler is more efficient in resource utilization and thermal efficiency than the isobutane cascade cycle.
- (2) At brine temperatures of 300°F and below, the thermal efficiency of the cascade cycle is about the same as that of the single boiler cycle. However, the resource utilization of the dual boiler cycle is about 20% higher than that of the single boiler cycle. A triple boiler cycle offers a 5% improvement over the dual boiler cycle in resource utilization.
- (3) The plant cost for the multiple boiler cycle increases as the number of boilers increases. But, the brine system cost de-

creases as the number of boilers increases. The brine system cost is dominant in the total system cost for low temperature brine (250°F). As a result, the total system cost for the dual boiler cycle is less than that of either the single boiler or the triple boiler cycle.

- (4) With proper selection of constituent fluids and with appropriate composition, binary mixtures are better than pure fluids as working fluids in the cascade cycle. Binary mixtures offer higher resource utilization and less capital investment.

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APPENDIX II

SELECTION OF WORKING FLUIDS FOR THE SINGLE BOILER GEOTHERMAL BINARY CYCLE

SELECTION OF WORKING FLUIDS FOR THE
SINGLE BOILER GEOTHERMAL BINARY CYCLE

by

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ABSTRACT

A procedure for working fluid selection for geothermal binary cycles has been developed. One hundred fluids were screened to find fluids with the best properties for single boiler Rankine cycles using heat available at 300-500°F. The first phase of the two phase screening procedure permits a quick evaluation, with potential working fluids identified for more detailed analysis. The second phase screening procedure allows one to rank potential working fluids on the basis of economics merely from knowledge of a few characterization parameters. These parameters are vapor pressure, specific volume and I-Factor for the saturated vapor at 100°F. From this study, it is concluded that the best working fluids are the fluids which have moderate vapor pressure and specific volume of saturated vapor at 100°F and an I-factor at 100°F close to 1.0. These fluids include 1-2-butadiene, 1-3-butadiene, cis-2 butene and trans-2 butene.

INTRODUCTION

The conversion of the thermal energy in liquid dominated geothermal reservoirs to electrical energy using the Rankine (binary) cycle has been under study in recent years. The geothermal temperatures encountered are usually low, in the range of 300° F to 500° F, so the thermal efficiency of the cycle is low. Therefore, the selection of a working fluid which will yield good performance is important.

There are hundreds of fluids which can be considered as working fluids for the geothermal binary cycle. Working fluids can be ranked on the basis of (1) thermodynamic performance, (2) resource utilization, and (3) economics. For this study, the selection of working fluids is based on economics.

The first phase of the working fluid screening procedure consists of a preliminary screening process, from which the potential working fluids are determined. The potential working fluids are then compared using simplified thermodynamic cycle calculations, equipment size calculations, and economic calculations in phase II of the working fluid selection procedure. Using the results of the above comparisons, the potential working fluids are ranked on the basis of total capital cost in \$/kw. The potential working fluids also can be ranked merely from knowledge of a few characterization parameters; this ranking agrees with the ranking based on the simplified cycle calculations.

This two phase methodology for working fluid selection was developed in such a manner that practicing engineers can rank

working fluids for a given geothermal binary cycle using simple hand calculated cycle parameters.

PRELIMINARY SCREENING PROCEDURE

(PHASE I)

Several fluid properties are considered to be important for the selection of working fluids for the geothermal binary cycle. If an ideal working fluid for geothermal binary cycles existed, it could be characterized reasonably well by the following requirements.⁽¹⁾

- (1) The working fluid dew point pressure at the point of initial condensation (say 110°F) in the condensor should be as low as possible (to minimize condensor cost per unit of heat transfer surface area, which is an increasing function of pressure) but greater than atmospheric pressure (to avoid leakage of air into the system).
- (2) The working fluid saturated vapor locus on a temperature-entropy diagram should be nearly vertical, to avoid excessive superheat in heat exchangers and condensation at the turbine exit. Superheat is undesirable because heat transfer coefficients in the superheat region are smaller than in the boiling and condensing regions. Condensation in the turbine is undesirable for axial flow turbines because of the loss of approximately one percent in stage efficiency for each percent of liquid formed (manufacturers claim negligible loss due to condensation in radial inflow turbines). Condensation in the turbine is also undesir-

able because of the potential maintenance problems created by liquid impingement erosion.

- (3) The working fluid specific enthalpy change in the turbine should be large, to maximize cycle thermodynamic efficiency and minimize working fluid flow rate.
- (4) The turbine inlet pressure should be reasonably low, to minimize brine heat exchanger cost per unit of heat transfer surface area, which is an increasing function of pressure.
- (5) The working fluid specific volume at the turbine exit should be small, to keep turbine size small (the turbine diameter is proportional to the square root of the exhaust volumetric flow rate).
- (6) Working fluid liquid and vapor thermal conductivity coefficients should be large, to minimize heat transfer surface area. Correlations of convective heat transfer coefficients show that high values of thermal conductivity and low values of viscosity are desirable.
- (7) Working fluid liquid and vapor viscosities should be low, to minimize frictional pressure drops and maximize convective heat transfer coefficients.
- (8) The working fluid should be stable, essentially nonfouling, noncorrosive, nontoxic and nonflammable. At the very least, underwriter approved safe handling methods must exist.

Assuming a sink temperature of 80°F and a pinch point difference of 20°F , the minimum condensing temperature of the working

fluid is limited to 100°F. So all fluid properties related to the turbine exit condition are evaluated at 100°F. To initiate the preliminary screening process the characterization parameter such as the molecular weight and the critical constants as well as certain thermodynamic property data were gathered and tabulated for the 100 working fluids considered in Table 1.0. In Table 2.0, the I-Factor is the following quantity defined by Kihara and Fukunaga⁽²⁾,

$$I = 1 - \frac{T_{\text{COND}}/C_p}{(dT/ds) \text{ Sat. vapor at } 100^\circ\text{F}} \quad (1)$$

where T_{COND} is the condensing temperature, C_p is the saturated vapor heat capacity and (dT/ds) is the slope of the saturated vapor temperature-entropy locus. For a fluid with a vertical saturated vapor locus on a temperature-entropy diagram, the I-Factor is unity. If $I < 1$, the turbine exhaust will be superheated, while for $I > 1$, the turbine exhaust may be wet and the turbine inlet superheated.⁽¹⁾

The costs of the heat exchangers and the cycle pumps increase with increasing pressure, and hence low vapor pressure is a criterion for working fluid selection.

Since the minimum condensing temperature of the working fluid is limited to 100°F, working fluids with critical temperature, T_c , less than 100°F will always remain in the superheated phase, which is not desired, and hence they can be rejected as comparatively poor working fluids.

Also, fluids which are highly toxic, corrosive and miscible with water can be rejected.

In the preliminary screening process (Phase I) as shown in Figure 1.0 candidate working fluids are rejected if (1) the fluid vapor pressure at 100° F is outside of the range 5 psia to 250 psia and/or (2) the fluid I-Factor at 100 F is outside of the range 0.5 to 2.0 and/or (3) the fluid critical temperature is less than 100° F and/or (4) the fluid is highly toxic, corrosive and/or miscible with water.

The above limits for vapor pressure and I-Factor are arbitrary values at this stage; however a better approximation for the vapor pressure range has resulted from the present study.

Out of the one hundred fluids analyzed in the preliminary screening process, twenty-six of them passed through and are listed in Table 2.0.

For the 100 fluids listed in Table 1.0, the number in the parenthesis indicates if the fluid was rejected or not and if rejected for what reason the fluid was rejected. The numbers and the restrictions are as follows:

- (0) Passed through preliminary screening restrictions.
- (1) 5 psia < vapor pressure at 100° F > 250 psia.
- (2) 0.5 < I-Factor > 2.0.
- (3) Critical temperature < 100° F.
- (4) Highly toxic, corrosive and/or miscible with water.
- (5) Insufficient data. Could not be analyzed in Phase II.

Table 1.0 Working Fluids Considered in
the Preliminary Screening Process

Water (1)	Ethyl Formate (0),(5)
Ammonia (2)	Methyl Acetate (0),(5)
Sulfur Dioxide (2)	Ethylene Dichloride (1)
R-13B1 (1)	Propane (0)
R-13 (1)	1-Propanol (1)
R-12 (0)	Furan (0),(5)
R-11 (0)	1-2,Butadiene (0)
R-10 (1)	1-3,Butadiene (0)
R-14 (3)	Vinyl Acetate (1)
Carbon Disulfide (0),(5)	Methyl Acrylate (1)
R-22 (0)	Butyronitrile (1)
R-20 (1)	I-Butene (0)
Methylene Chloride (0),(5)	Cis-2-Butene (0)
R-152a (0)	Trans-2-Butene (0)
Methanol (1)	Isobutylene (0),(5)
R-500 (0)	2-Butanone (1)
R-503 (3)	Tetrahydrofuran (0),(5)
R-504 (1)	Vinyl Ethyl Ether (0),(5)
R-502 (1)	N-Butyraldehyde (1)
R-114 (0)	Isobutyraldehyde (0),(5)
R-113 (0)	Ethyl Acetate (1)
Perchloroethylene (1)	N-Butane (0)
Trichloroethylene (1)	Isobutane (0)
Cis,1-2,Dichloroethylene (0),(5)	Ethyl Ether (0),(5)
Trans,1-2,Dichloroethylene (0),(5)	Butanol (1)
Acetyl Chloride (0),(5)	Isobutyl Alcohol (1)
R-142b (0)	Butylamine (1)
R-140a (1)	Pyridine (1)
Acetonitrile (1)	Ethyl Acrylate (1)
Chloral (1)	Cyclopentane (0)
R-150a (0),(5)	I-Pentene (0)
Propyl Chloride (0),(5)	N-Pentane (0)
Ethylene Oxide (0),(5)	Isopentane (0)
Acitric Acid (0),(5)	Bromobenzene (1)
Ethyl Chloride (0),(5)	Chlorobenzene (1)
Ethane (1)	Fluorobenzene (1)
Ethanol (1)	Benzene (1)
Ethanethiol (0)	I-Hexene (0)
Dimethylsulfide (0)	Methyl Isobutyl Ketone (1)
Ethylamine (0),(5)	Isohexane (0),(5)
R-216 (0)	N-Hexane (0)
Propadiene (0)	Propyl Ether (1)
Propionitrile (1)	Triethylamine (1)
Epichlorohydrin (1)	Toluene (1)
Propylene (0)	I-Heptene (1)
1-2,Dichloropropane (1)	N-Heptane (1)
Propionaldehyde (0),(5)	Ethyl Benzene (1)
Acetone (0),(5)	Isooctane (1)
Allyl Alcohol (1)	N-Octane (1)
Propylene Oxide (4)	Propylbenzene (1)

Figure 1.0 Preliminary Screening Procedure (Phase I)

TABLE 2.0

SELECTED POTENTIAL WORKING FLUIDS AND THEIR THERMODYNAMIC CHARACTERIZATION PARAMETERS

Fluid	Formula	Molecular Weight	Critical Temperature °F	Critical Pressure psia	Critical Density lb mole cu ft	Acentric Factor	Vapor Pressure at 100°F psia	$\frac{ds}{dT}$, Sat Vap at 100°F BTU/lb-°F	I-Factor	Specific Volume of Sat Vapor at 100°F Cu ft/lb	Normal Boiling Point °F
1. R-12	CCl_2F_2	120.93	233.6	597	0.287	0.176	131.9	-0.0000664	0.91	0.308	-22
2. R-11	CCl_3F	137.34	390	639.4	0.252	0.184	23.6	-0.0000119	1.05	1.7645	74.6
3. R-22	CHClF_2	86.47	205	74.7	0.378	0.225	212.6	-0.0003162	1.12	0.257	-41.7
4. R-152a	CH_3CHF_2	66.05	236.3	652	0.0345	0.27	124.6	-0.0002529	1.159	0.61087	-13
5. R-114	$\text{C}_2\text{Cl}_2\text{F}_4$	170.94	294.3	473.22	0.213	0.2495	46.4	+0.0000917	0.65	0.695	38.6
6. R-113	$\text{C}_2\text{Cl}_3\text{F}_3$	187.4	419	495.4	0.192	0.252	10.48	+0.00075	0.78	2.98	117.3
7. R-142b	$\text{C}_2\text{H}_3\text{ClF}_2$	100.5	279	598.2	0.27	-	71.5	-0.000048	1.24	0.724	14
8. Dimethylsulfide	$\text{C}_2\text{H}_6\text{S}$	62.13	445.4	802.6	0.311	0.19	15	-0.00014	1.10	6.348	98.9
9. Ethanethiol	$\text{C}_2\text{H}_6\text{S}$	62.13	438.2	796.7	0.302	0.19	18	-0.00009	1.18	5.915	96
10. R-216	$\text{C}_3\text{Cl}_2\text{F}_6$	220.93	355.9	399.5	0.1623	0.31	15.7	-0.0001202	1.42	1.6214	96.24
11. Propadiene	C_3H_4	40.06	249	793.8	0.391	0.273	156.3	-0.000222	1.55	0.51	-30.3
12. Propylene	C_3H_6	42.08	200	670.3	0.345	0.14	227.5	-0.000259	1.0	0.465	54.3
13. Propane	C_3H_8	44	206.3	617.4	0.308	0.152	188.3	-0.000108	1.32	0.559	-44
14. 1-2, Butadiene	C_4H_6	54.092	340	652.7	0.285	0.255	36	+0.003088	0.99	2.88	51.2
15. 1-3, Butadiene	C_4H_6	54.092	305	627.7	0.282	0.195	59.57	+0.0000247	1.0	1.7	23.7
16. I-Butene	C_4H_8	56.108	295.6	583.5	0.26	0.186	60.9	+0.000025	0.97	1.569	20.4
17. Cis-2-Butene	C_4H_8	56.108	324.4	610	0.266	0.2023	46.1	-0.000041	1.04	2.125	38.4
18. Trans-2-Butene	C_4H_8	56.108	311.86	595.35	0.262	0.214	49.8	+0.000014	0.93	2	33.2
19. N-Butane	C_4H_{10}	58.124	305.7	551.7	0.244	0.193	51.36	+0.000113	0.75	1.81	30.9
20. Isobutane	C_4H_{10}	58.124	274.9	529.1	0.237	0.177	72	+0.00013	0.81	1.26	10.3
21. Cyclopentane	C_5H_{10}	70.135	460	654.1	0.24	0.192	10.04	-0.000058	1.02	8.3	120.3
22. I-Pentene	C_5H_{10}	70.135	376.9	590	0.208	0.243	19.115	+0.000093	0.76	5.37	85.6
23. N-Pentane	C_5H_{12}	72.15	385.7	489.6	0.205	0.251	15.6	+0.0000231	0.7	5.1	96.6
24. Isopentane	C_5H_{12}	72.15	369	490.9	0.204	0.227	20.4	+0.00025	0.77	3.85	81.8
25. I-Hexene	C_6H_{12}	84.16	450	460.1	0.178	0.285	5.8	+0.000163	0.7	12	145.9
26. N-Hexane	C_6H_{14}	86.178	453.7	430.9	0.169	0.3	5.0	-0.000031	0.73	13.73	155.4

RANKING OF THE POTENTIAL WORKING FLUIDS (PHASE II)

After selecting the potential working fluids from the preliminary screening process (Phase I), it would be necessary for the practicing engineer to estimate the total capital investment in \$/kw to rank the potential working fluids. In this study two methods are used to rank the potential working fluids without complicated computer or hand calculations. In the first method cycle parameters are calculated using "Simplified Thermodynamic Cycle Calculations" and then the equipment costs are determined utilizing the correlation of capital costs of equipment units as functions of cycle parameters and a few characterization parameters. In the second method the fluids are ranked merely from knowledge of a few characterization parameters such as vapor pressure, I-Factor and specific volume for the saturated vapor at 100°F. The rankings by both of the methods agree very well.

SIMPLIFIED THERMODYNAMIC CYCLE CALCULATION

The method used for this simplified thermodynamic cycle calculation is to perform ideal Rankine cycle calculations with some of the cycle parameters being fixed. The definition of the ideal Rankine cycle used here is a Rankine cycle in which working fluid expansion and compression are reversible and there are no viscous pressure drops.

The parameters fixed are (1) the brine system (brine resource temperature, brine flow rate and brine temperature drop), (2) condenser logarithmic mean temperature difference, $LMTD_{cond}$, (3) turbine inlet temperature, (4) cooling water inlet and exit temperatures, (5) the brine heat exchanger tube velocity, (6) the heat exchanger shell side equivalent diameter for heat transfer coefficient calculations, (7) the brine forced convective heat transfer coefficient, (8) the conductivity coefficient, (9) and the brine and the working fluid side fouling factors.

The brine system was held fixed so concentration would be on the behavior of the working fluids.

The cost of the condenser can be expressed approximately using the relation

$$\begin{aligned} \text{Condenser cost} &= f \left(\frac{\text{heat transfer}}{\text{surface area}} \right) (\text{pressure}) \\ &= f \left[\frac{Q_C}{U (LMTD)_{cond}} (\text{pressure}) \right] \end{aligned} \quad (2)$$

Hence the cost of the condenser can be made dependent principally on working fluid behavior by fixing the $(LMTD)_{cond}$.

As the turbine inlet temperature is increased, the enthalpy change in the turbine increases, which results in a higher power output. However, at the same time there is an increase in the operating pressure and a decrease in the LMTD, which usually results in a cost increase for the brine heat exchanger. It has been found that the optimum turbine inlet temperature is comparable for different pure working fluids (for a common georesource temperature). Therefore, the turbine inlet temperature was held fixed

in the present calculations.

The brine system (flowrate, inlet and exit temperatures), condenser LMTD and turbine inlet temperature were fixed at realistic values determined from cost optimization of a 25 MW isobutane geothermal binary cycle single boiler power system.

The other fixed parameters such as tube velocity, equivalent diameter, fouling factors, etc. do not affect the ranking of the working fluids significantly and hence they were also fixed at realistic values determined from cost optimization of a 25 MW isobutane cycle.

Also, the overall weighted average heat transfer coefficient, U , is assumed to be approximately represented by the working fluid forced convective heat transfer coefficient, h , because in the right hand side of equation (3),

$$\frac{1}{U} = \frac{1}{\frac{1}{h} + \frac{1}{h_H} + \frac{1}{R_{WF}} + \frac{1}{R_n} + R_W} \quad (3)$$

all the terms except h are held fixed. So U is assumed to be approximately represented by h since differences in the magnitude of U are due to variations in h .

To perform the simplified thermodynamic cycle calculation the condensing temperature is taken to be 115°F , so the turbine exit pressure is the vapor pressure of the working fluid at 115°F .

Then if $I > 1$, the turbine exit condition is assumed to be the saturated vapor at 115°F . An isentropic compression calculation from turbine exit condition to the fixed turbine inlet temperature determines the turbine inlet condition. For $I < 1$, the turbine inlet condition is assumed to be saturated vapor at the turbine inlet

temperature and an isentropic expansion calculation to the turbine exit pressure fixes the turbine exit condition. The cycle pump inlet condition is assumed to be the saturated liquid at 115°F . Hence the specific turbine and the pump work can be calculated as the respective working fluid specific enthalpy changes. The brine heat exchanger duty is equal to brine flow rate, $(\dot{M}_H)$, times the brine temperature drop, $(\Delta T)_H$. The cycle thermodynamic efficiency, η_N , can be calculated as the net cycle work divided by the brine heat exchanger duty. The procedure for performing these calculations is presented in Appendix A.

RELATIONSHIP BETWEEN WORKING FLUID THERMODYNAMIC PROPERTIES AND MAJOR EQUIPMENT COSTS

In Phase I of the methodology for working fluid selection, a preliminary screening procedure was used to screen the potential working fluids. In Phase II a simple procedure was used to calculate important cycle parameters. In this section the costs of the major equipment units in the geothermal binary cycle are expressed empirically with reasonable correlations as functions of the thermodynamic properties of the working fluid along with certain cycle parameters. This correlation procedure allows one to determine the costs of the major equipment units without the use of complicated computer or hand calculations. This methodology should save a lot of time and effort for the practicing engineer as one can quickly identify the low cost cycle fluids, and thus avoid serious analysis for the comparatively poor working fluids.

To initiate this correlation procedure, the GEO4 computer simulator, which was developed at the University of Oklahoma, was used to simulate a geothermal binary cycle. Equipment sizing and costing calculations are included in GEO4 to make the program useful for design and working fluid selection, as in the work presented herein. Calculations were performed essentially as explained in the simplified thermodynamic cycle calculation procedure, but with the use of less approximations. The costs of the major equipment units in \$/kw, the total capital investment in \$/kw, operating conditions and the cycle parameters were obtained from GEO4 simulator calculations for all of the potential fluids for the three resource temperatures 300°F, 400°F and 500°F.

The correlating parameters with which the costs of major equipment units varied were sought; plots were prepared which show the costs of the equipment in \$/kw versus the corresponding correlating parameters. It should be noted that some of the fluids may decompose in prolonged service at some of the operating conditions; nevertheless, the calculations were performed in these cases to be able to develop the correlations presented herein. Decomposition should of course be taken into account in designing an actual plant.

BRINE HEAT EXCHANGER COST

The brine heat exchanger cost is a function of $\dot{Q}_H$, U , $(LMTD)_{BHE}$ and the operating pressure P_{BHE} as noted in Equation (4),

$$\text{Brine heat exchanger cost, \$ / kw} = f \left[\frac{P \dot{Q}_H}{U \text{ LMTD}} \right]_{BHE} \quad (4)$$

In the simplified thermodynamic cycle calculation, $\dot{Q}_H$ is a fixed parameter, U is represented by h and hence Equation (4) reduces approximately to,

$$\text{Brine heat exchanger cost, } \$/\text{kw} = f \left[\frac{P}{h \times \text{LMTD}} \right]_{\text{BHE}} \quad (5)$$

After study of the brine heat exchanger cost and the behavior of working fluid properties, Equation (5) resulted. The exact correlating parameter with which the brine heat exchanger cost varies is the right hand side of Equation (6) for the hydrocarbons studied and the right hand side of Equation (7) for the halocarbons studied.

$$\text{Brine heat exchanger cost, } \$/\text{kw} = f \left[\frac{P^{0.44}}{h^{0.5} \text{LMTD}} \right]_{\text{BHE}} \quad (6)$$

$$\text{Brine heat exchanger cost, } \$/\text{kw} = f \left[\frac{P}{h^{0.5} \text{LMTD}} \right]_{\text{BHE}} \quad (7)$$

Figures 2 and 3 show the brine heat exchanger cost versus the corresponding correlating parameters.

CONDENSER COST

The condenser cost is not affected by pressure for fluids with condenser operating pressures less than 40 psia. So, different correlating parameters were required for the fluids with condenser operating pressure above and below 40 psia. They are as follows,

Figure 2.0 Brine Heat Exchanger Cost For Hydrocarbons

Figure 3.0 Brine Heat Exchanger Cost For Halocarbons

Condenser Cost for Fluid
Type and Operating Pressure

Halocarbons and hydrocarbons
with operating pressure less
than 40 psia

Hydrocarbons with operating
pressure greater than 40 psia

Halocarbons with operating
pressure greater than 40 psia

Correlating Parameter

$$\frac{(\Delta H)_T}{h_{\text{cond}}}$$

$$\frac{P_{\text{cond}}^{0.44} (\Delta H)_T}{h_{\text{cond}}}$$

$$\frac{P_{\text{cond}}}{h_{\text{cond}}}$$

The above correlating parameters were selected empirically, starting with the fact that the condenser cost is a function of $\dot{Q}_C$, U , LMTD and P_{cond} as noted in Equation (8).

$$\text{Condenser cost, } \$/\text{kw} = f \left[\frac{P \dot{Q}_C}{U \times \text{LMTD}} \right]_{\text{cond}} \quad (8)$$

$\dot{Q}_C$ is equal to $(\dot{W}_N - \dot{Q}_H)$ and since in the simplified thermodynamic and hence cycle calculation $\dot{Q}_H$ is a fixed parameter and $\dot{W}_N$ is represented by $(\Delta H)_T$ times flow rate Equation (8) reduces approximately to the following

$$\text{Condenser cost, } \$/\text{kw} = f \left[\frac{P}{h \text{ LMTD}} (\Delta H)_T \right]_{\text{cond}} \quad (9)$$

Figures (4) and (5), respectively, show condenser costs for operating pressures less than and greater than 40 psia as functions of the corresponding correlating parameters.

Figure 4.0 Condenser Cost For The Working Fluids With
Condenser Operating Pressure Less Than 40 psia

Figure 5.0 Condenser Cost For The Working Fluids With Condenser Operating Pressure More Than 40 psia

TURBINE COST

The turbine cost was found to vary with the specific volume of the saturated vapor at the turbine exit as shown in Figures (6) and (7) for hydrocarbons and halocarbons, respectively.

CYCLE PUMP COST

The cycle pump cost varies with the cycle pump work for hydrocarbons and also the brine heat exchanger specific duty for halocarbons, as noted in Equations (10) and (11) and shown in Figures (8) and (9),

$$\text{Hydrocarbon cycle pump cost, } \$/\text{kw} = f \left[v^L (P_{\text{BHE}} - P_{\text{cond}}) \right] \quad (10)$$

$$\text{Halocarbon cycle pump cost, } \$/\text{kw} = f \left[\frac{(\Delta H)_P}{\Delta H_{\text{BHE}}} \right] \quad (11)$$

GENERATOR COST AND COOLING SYSTEM COST

The costs of the generator and the cooling system can be estimated by simple formulas as follows:

$$\text{Generator cost, } \$ = 22500 \left(\frac{M_{\text{we}}}{10} \right)^{0.7} \quad (12)$$

$$\text{Cooling System cost, } \$ = 3.33 (\dot{M}_{\text{CW}} \text{ in gallons/minute}) \quad (13)$$

TOTAL CAPITAL COST ESTIMATION

The total capital investment was estimated from the major equipment unit costs determined in the previous section using the

Figure 6.0 Turbine Cost For Hydrocarbons

Figure 7.0 Turbine Cost For Halocarbons

Figure 8.0 Cycle Pump Cost For Hydrocarbons

Figure 9.0 Cycle Pump Cost For Halocarbons

so called "factored estimate method". This method as used here is essentially the same as that explained by Milora and Tester⁽⁵⁾. The method requires the determination of the major equipment unit costs. The other items included in the total direct plant cost are then estimated as factors of the major equipments. The additional components of the capital investment are based on average percentages of the total direct plant cost, total direct and indirect plant cost, or total capital investment. The procedure for the estimation of the total capital investment in \$/kw, is explained in Appendix B.

The total capital investment in \$/kw is tabulated in rank order in Table (3) for twenty-six working fluids for georesource temperatures of 300°F, 400°F and 500°F. The results in Table (3) were obtained using the GEO4 simulator; however, the ranking would be essentially the same using hand calculations and the correlations herein. The determination of the ranking in Table (3) completes the process of working fluid selection within the scope of the present study; the final choice for the working fluid and cycle operating conditions should be made on the basis of more detailed cycle design calculations and vendor equipment cost quotations.

The methodology presented as Phases I and II in this study provides a viable framework for working fluid selection. In addition, due to the fact that cycle calculations were performed for a relatively large number of fluids, it was possible to note that only a few parameters seem to control the ranking of the fluids within a given class (e.g., hydrocarbons or halocarbons). These parameters are the vapor pressure, saturated vapor specific volume and the I-Factor, all at 100°F. In the next section an improved

TABLE 3.0.

TOTAL CAPITAL INVESTMENT FOR POTENTIAL WORKING FLUIDS

300° F			400° F		500° F	
Ranking	Fluid	Total Capital Invst, \$/kw	Fluid	Total Capital Invst, \$/kw	Fluid	Total Capital Invst, \$/kw
1	1-2, Butadiene	1027.26	1-2, Butadiene	566.32	Dimethyl Sulfide	416.8
2	Cis-2, Butene	1049.64	Cis-2, Butene	582.6	1-2, Butadiene	421.33
3	1-3, Butadiene	1061.17	1-3, Butadiene	595.09	Ethyl Mercaptan	426.91
4	N-Butane	1071.38	Trans-2-Butene	599.11	Cis-2, Butene	436.8
5	Trans-2-Butene	1072.58	Dimethyl Sulfide	612.18	Trans-2-Butene	454.67
6	N-Pentane	1098.04	N-Butane	616.25	1-3, Butadiene	457.96
7	I-Butene	1107.65	R-11	620.86	R-11	462.87
8	Isopentane	1118.67	I-Butene	621.9	N-Pentane	473.27
9	R-11	1134.95	Ethyl Mercaptan	625.7	N-Butane	474.61
10	Ethyl Mercaptan	1142.0	Isopentane	639.75	Isopentane	479.04
11	R-114	1153.68	N-Pentane	641.52	I-Butene	481.86
12	Dimethyl Sulfide	1156.08	R-142b	661.24	Cyclopentane	505.39
13	Isobutane	1178.56	I-Pentene	674.72	I-Pentene	513.56
14	I-Pentene	1187.48	Isobutane	687.38	R-142b	514.41
15	Propadiene	1190.10	R-114	688.79	Propadiene	527.13
16	R-216	1231.82	Propadiene	699.00	R-114	527.74
17	R-113	1285.12	R-152a	718.48	Isobutane	531.19
18	Cyclopentane	1302.77	R-216	739.12	R-113	561.56
19	R-142b	1344.90	R-113	739.43	R-216	570.63
20	R-152a	1355.36	Cyclopentane	779.24	R-152a	586.11
21	I-Hexene	1459.83	R-22	826.65	I-Hexene	635.83
22	Propane	1461.21	Propane	836.77	Propane	646.54
23	Propylene	1497.91	Propylene	840.74	R-12	653.32
24	R-12	1513.56	R-12	852.7	Propylene	653.69
25	R-22	1634.21	I-Hexene	879.00	R-22	663.00
26	N-Hexane	1787.03	N-Hexane	929.63	N-Hexane	699.20

preliminary screening procedure and a rough ranking procedure which requires only these few parameters are presented.

RANKING OF POTENTIAL WORKING FLUIDS MERELY FROM KNOWLEDGE OF A FEW CHARACTERIZATION PARAMETERS

In Phase II of the working fluid selection procedure, ideal Rankine cycle calculations were performed for the working fluids passing the Phase I preliminary screening; these fluids were ranked on the basis of total capital investment in \$/kw. In this section the working fluids will be ranked from knowledge of only a few characterization parameters.

From inspection of the behavior of the working fluids studied in Phase II, a relationship between the working fluid vapor pressure and specific volume of saturated vapor was noted. Figures (10) and (11) show the relationship between these properties evaluated at 100°F for hydrocarbons and halocarbons respectively. The fluids designated as group I are fluids with low vapor pressures and high saturated vapor specific volumes at 100°F. They have inexpensive heat exchangers and cycle pumps (due to low vapor pressure) and expensive turbines (due to high specific volume of saturated vapor). The fluids designated as group III are fluids with high vapor pressures and low saturated vapor specific volumes and hence expensive heat exchangers and cycle pumps and inexpensive turbines. Fluids designated as group II have moderate vapor pressures and saturated vapor specific volume and hence cycle total equipment costs generally are lower than both group I and group

Figure 10.0 Relationship Between The Vapor Pressure And The Specific Volume Of Saturated Vapor At 100 F For Hydrocarbons

Figure 11.0 Relationship Between The Vapor Pressure And The Specific Volume Of Saturated Vapor At 100 F For Halocarbons

III. Table 4.0 shows the 26 potential working fluids divided into 3 groups according to their behavior.

When analysing Figure (10), the curve for group I fluids begins to have a constant vapor pressure value somewhere between the saturated vapor specific volume of 6 and 12 cuft/lb. Just beyond this constant vapor pressure value (approximately 11 psia) the turbine cost becomes approximately constant, i.e., the heat exchanger cost cannot be reduced much further and hence a fluid which has a vapor pressure at 100°F, lower than this constant vapor pressure value should be rejected. When comparing two fluids, one having a lower vapor pressure and the other having a greater vapor pressure than the constant vapor pressure value, both will have approximately the same heat exchanger and pump costs but the former will have a higher turbine cost. So this constant vapor pressure value, 11 psia, can be fixed as a lower limit for the vapor pressure property at 100°F to screen the working fluids. The corresponding upper limit for the specific volume of the saturated vapor is 7 cu ft/lb. Similarly, when analysing group III fluids the curve begins to have a constant saturated vapor specific volume somewhere between vapor pressure values of 80 and 140 psia. At lower saturated vapor specific volumes the turbine cost cannot be reduced significantly and hence a limit of about 0.7 cuft/lb can be placed on the saturated vapor specific volume. The corresponding upper limit for vapor pressure is 110 psia.

Similarly, the halocarbon cycles were analysed using Figure (11), with the better working fluids falling into the ranges 11 to 100 psia for vapor pressure at 100°F and 0.8 to 2.33 cuft/lb for saturated vapor specific volume at 100°F.

TABLE 4.0

BEHAVIOR GROUPINGS FOR THE POTENTIAL
WORKING FLUIDS

HYDROCARBONS

- GROUP 1
1. N-Hexane
 2. I-Hexene
 3. Cyclopentane
 4. Dimethylsulfide
 5. N-Pentane
 6. Ethanethiol
 7. I-Pentene
 8. Isopentane

- GROUP 2
1. 1-2-Butadiene
 2. Cis-2-Butene
 3. Trans-2-Butene
 4. N-Butane
 5. 1-3-Butadiene
 6. I-Butene
 7. Isobutane

- GROUP 3
1. Propadiene
 2. Propane
 3. Propylene

HALOCARBONS

- GROUP 1
1. R-113
 2. R-216
 3. R-11

- GROUP 2
1. R-114
 2. R-142b

- GROUP 3
1. R-152a
 2. R-12
 3. R-22

These improved limits for the vapor pressure and the saturated vapor specific volume at 100°F are represented by the shaded areas in Figures(10) and (11).

These new limits for the vapor pressure and the saturated vapor specific volume can be used for an "improved preliminary screening process" for both hydrocarbons and halocarbons (the rest of the restrictions in the original preliminary screening process being the same). The 26 fluids in Table (3) were analysed using this improved preliminary screening process and 10 of the 26 were eliminated.

To rank these 16 fluids merely from knowledge of few characterization parameters, a relationship between working fluid properties and total equipment cost is required, which can be obtained from the results of Phase II analysis of this research and from Figures (10) and (11). This analysis shows that the fluids from group II have low cost cycles and hence they can be ranked best. Fluids from group I can be ranked the second best as some of them have moderately expensive turbines in addition to their inexpensive heat exchangers and pumps. The third best will be the fluids from group III.

When comparing the working fluids within a group, a third property, I-Factor is taken into consideration. Fluids with I-Factor close to 1.0 have less superheat at cycle conditions and hence have better performance. So in group II the fluids with I-Factor close to 1.0 are the best, and can be seen in Figures (12) and (13). If there are two or more fluids with equal I-Factors, they can be ranked according to their vapor pressures, the one with lower vapor pressure being better. The next best fluids are from

Figure 12.0 Ranking Methods for Potential Working Fluids Merely From Knowledge of few Properties

Figure 13.0 Screening of Hydrocarbons With Improved Limits for Vapor Pressure and Specific Volume of Saturated Vapor.

group I, and are the ones with I-Factors close to 1.0. If there are two or more fluids with equal I-Factors they can be ranked according to their saturated vapor specific volumes, the fluid having the lowest value being best. The third best fluids are the fluids from groups I and II with I-Factors far from 1.0. The fourth best fluids are the fluids from group III (if any). Figure (14) shows the ranking for the halocarbons.

The ranking done in Figures (13) and (14) agrees fairly well with the ranking done on the basis of the total capital investment in \$/kw.

CONCLUSIONS

The working fluids analysed in this research behaved in three different groups. The fluids from group II, have moderate vapor pressures and specific volumes of saturated vapor, and they are the best fluids provided they have I-Factors close to 1.0. The fluids from group I have low vapor pressures and high specific volumes of saturated vapor. They can be considered as the second best working fluids provided their specific volume of saturated vapor are low and I-Factors are close to 1.0. The third best working fluids are the ones from the above two categories which do not have I-Factors close to 1.0. The fluids from group III have high vapor pressures and low specific volumes of saturated vapor, and they can be considered as the fourth best working fluids. The fluids from all the above three groups which have exceptionally high vapor pressure or specific volume of saturated vapor can be considered as the fifth best working fluids. In this research it

Figure 14.0 Screening of Halocarbons With Improved Limits for Vapor Pressure and Specific Volume of Saturated Vapor

is concluded that group II fluids such as 1-2-butadiene, cis-2-butene, 1-3-butadiene and trans-2-butene are the best choices for working fluids for single boiler geothermal binary cycles.

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LIST OF SYMBOLS

T_{cond}	=	Saturation temperature corresponding to the condensing pressure, °F
C_p	=	Specific heat at constant pressure, Btu/lbm-°F
ds/dT	=	Slope of the saturated vapor locus, Btu/lb-°F ²
$\dot{Q}_H$	=	Brine heat exchanger duty
$\dot{Q}_C$	=	Condenser duty
U	=	Overall weighted average heat transfer coefficient, Btu/hr-°F ft ²
LMTD	=	Lograthmic mean temperature difference, °F
P	=	Pressure, psia
BHE	=	Brine heat exchanger
Cond	=	Condenser
h	=	Working fluid forced convective heat transfer coefficient, Btu/hr °F ft ²
$(\Delta H)_T$	=	Enthalpy change in the turbine, Btu/lb
$(\Delta H)_P$	=	Enthalpy change in the cycle pump, Btu/lb
v^L	=	Liquid specific volume at cycle pump inlet, lb/cuft
$(\Delta H)_{\text{BHE}}$	=	Enthalpy change in the brine heat exchanger, Btu/lb
$\dot{M}_{w_f}$	=	Working fluid flow rate, lb/hr
$\dot{M}_{cw}$	=	Cooling water flow rate, lb/hr
h_H	=	Brine side forced convertive heat transfer coefficient, Btu/hr °F ft ²
R_{w_f}	=	Working fluid side fouling factor, hr °F ft ² /Btu
R_H	=	Brine side fouling factor, hr °F ft ² /Btu
R_w	=	Conductivity coefficient, hr °F ft ² /Btu

W_T	=	Turbine work, Btu/hr
W_P	=	Pump work, Btu/hr
W_N	=	Net work, Btu/hr
ρ	=	Working fluid density, lb/cuft
V	=	Brine heat exchanger tube velocity, ft
D_e	=	The shell side equivalent diameter of the brine heat exchanger, ft
μ	=	Working fluid viscosity, lb/fthr
R	=	Reynolds number
Pr	=	Prandtl number
	=	Pressure drop, psia ft/hr ²
g	=	Gravitational constant

APPENDIX A

Procedure for the Simplified Thermodynamic Cycle Calculations

PROCEDURE FOR THE SIMPLIFIED THERMODYNAMIC CYCLE CALCULATION

The steps involved in the simplified thermodynamic cycle calculations are discussed below.

STEP 1 Fix the following cycle parameters.

- a) Brine flow rate
- b) Brine exit temperature
- c) Brine inlet temperature
- d) Turbine inlet temperature
- e) Condenser logarithmic mean temperature difference
- f) Cooling water inlet and outlet temperatures
- g) Turbine, cycle pump and other process pump efficiencies (100%)
- h) Zero viscous pressure drops within and between equipment units

STEP 2 Obtain the pressure-enthalpy diagram or suitable properties charts or tables for the fluid under consideration.

STEP 3 Identification of the state points.

- a) For the sink temperature of 80°F , a cooling water rise of 20°F and a pinch point of 15°F , the working fluid condensing temperature should be approximately 115°F . The condenser operating pressure would then be the working fluid vapor pressure at 115°F . Thus, $I > 1$, the condenser inlet temperature would be 115°F , since the turbine exit condition would be assumed

to be saturated vapor. For $I < 1$, an isentropic expansion calculation from the turbine inlet condition of saturated vapor at the turbine inlet temperature to the turbine exit pressure (taken to be the vapor pressure at 115°F) yields the condenser inlet conditions which is superheated gas, since $I < 1$.

- b) For $I < 1$, the turbine inlet and exit conditions are known from the calculations above in a). For $I > 1$, an isentropic compression calculation is performed from the condenser inlet condition to the fixed turbine inlet temperature. This point, which is superheated vapor, is the turbine inlet condition and the corresponding pressure is the brine heat exchanger operating pressure. Of course, if $I = 1$, both turbine inlet and exit conditions are saturated vapor.
- c) Saturated liquid at 115°F is the condenser exit condition. From this point an isentropic compression is made to the brine heat exchanger operating pressure, allowing a 3°F rise. This point is the cycle pump exit condition.
- d) Calculate the enthalpy changes in the cycle pump, turbine and the brine heat exchanger.

STEP 4 The calculation of the major cycle parameters is performed using the relations.

$$a) \quad \dot{M}_{WF} = \frac{\dot{M}_H C_{P_H} (\Delta T)_H}{(\Delta H)_{BHE}} \quad \text{lb/hr} \quad (14)$$

$$b) \dot{W}_T = \dot{M}_{WF} (\Delta H)_T \quad (15)$$

$$c) \dot{W}_P = \dot{M}_{WF} (\Delta H)_P \quad (16)$$

$$d) \dot{Q}_C = (\dot{W}_N) - \dot{Q}_H$$

$$= (\dot{W}_T + \dot{W}_P) - \dot{M}_H C_{P_H} (\Delta T)_H \quad (17)$$

$$e) \dot{M}_{CW} = \dot{Q}_C / C_{P_{CW}} (\Delta T)_{CW} \quad (18)$$

where the heat capacities of brine (C_{P_H}) and the cooling water ($C_{P_{CW}}$) are taken to be 1 Btu/lb °F for convenience.

STEP 5 a) Calculation of brine heat exchanger logarithmic mean temperature difference, $(LMTD)_{BHE}$, for a subcritical cycle.

Point A = Brine exit condition

Point B = Brine inlet condition

Point C = Brine heat exchanger working fluid inlet condition

Point D = Brine heat exchanger working fluid outlet condition

Point E = Brine temperature corresponding to brine heat exchanger bubble point

ΔH_1 = working fluid specific enthalpy drop in Section 1

ΔH_2 = working fluid specific enthalpy drop in Section 2

The brine temperature at point E = $T_E = T_B - \frac{\dot{M}_{WF} (\Delta H_2)}{\dot{M}_H \times C_{P_H}}$

where C_{P_H} = heat capacity of brine (assumed to be equal to 1)

Knowing all the temperatures at points A, B, C, D, E and F the temperature differences ΔT_1 , ΔT_2 and ΔT_3 can be found.

LMTD for Section 1

$$\text{LMTD 1} = \frac{\Delta T_2 - \Delta T_1}{\ln \frac{\Delta T_2}{\Delta T_1}} \quad ^\circ \text{F} \quad (20)$$

LMTD for Section 2

$$\text{LMTD 2} = \frac{\Delta T_3 - \Delta T_2}{\ln \frac{\Delta T_3}{\Delta T_2}} \quad ^\circ \text{F} \quad (21)$$

The average weighted logarithmic mean temperature difference for a subcritical cycle is = $\frac{\Delta H_1 + \Delta H_2}{\frac{\Delta H_1}{\text{LMTD 1}} + \frac{\Delta H_2}{\text{LMTD 2}}}$ (22)

b) Calculation of the $(LMTD)_{BHE}$ for a Supercritical Cycle.

Point A = Brine exit condition

Point B = Brine inlet condition

Point C = Brine heat exchanger working fluid inlet condition

Point D = Brine heat exchanger working fluid outlet condition

Point F = Assumed point = $(T_D - 10^\circ F)$

Point E = Brine temperature corresponding to point F

$\Delta H1$ = Enthalpy drop in Section 1

$$\text{temperature at E} = T_E = T_B - \frac{\dot{M}_{WF} \times \Delta H1}{\dot{M}_H \times C_{P_H}} \quad ^\circ F \quad (23)$$

Knowing temperature at E, the temperature differences $\Delta T1$ and $\Delta T2$ can be calculated.

So $LMTD$ at Section 1 = $LMTD1$

$$LMTD1 = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}} \quad (24)$$

Now assume another point, H, such that $T_H = (T_F - 10^\circ\text{F})$ and let point G be

the corresponding brine temperature

$$\text{temperature at G} = T_G = T_E - \frac{\dot{M}_{WF} \Delta H2}{\dot{M}_H C_{PH}} \quad ^\circ\text{F} \quad (25)$$

Knowing the temperature at G the temperature difference between G and H can be calculated to yield LMTD 3.

The LMTD of section 2 then is

$$\text{LMTD2} = \frac{\Delta T_2 - \Delta T_3}{\ln \frac{\Delta T_2}{\Delta T_3}} \quad (26)$$

The above steps are repeated until the temperature at "C" is reached; the overall LMTD is then the weighted average of all the calculated LMTD's.

STEP 6 a) Calculation of the working fluid forced convection heat transfer coefficient in the brine heat exchanger for a single phase region (where the working fluid exists in a single phase). The correlation used for this calculation is the "Dittus - Boelter" correlation, (3)

$$h_{WF1} = 0.023 \left(\frac{\rho_l v D_e}{\mu_l} \right)^{0.8} \left(\frac{\mu_l C_{pl}}{k_l} \right)^{0.4} \frac{k_l}{D_e} \quad (27)$$

b) calculation of the two-phase working fluid forced convection heat transfer coefficient in the brine heat exchanger is performed using Chen's correlation, (4)

$$h_{WF2} = F \cdot 0.023 \cdot Re^{0.8} \cdot Pr_l^{0.4} \cdot \frac{k_l}{D_e} + (0.001225) \left(\frac{k_l}{D_e} \right)^{0.79} \left(\frac{C_{pl}}{D_e} \right)^{0.45} \left(\frac{\rho_l}{D_e} \right)^{0.49} \left(\frac{f_c}{D_e} \right)^{0.25} \left(\frac{\Delta P_s}{D_e} \right)^{1.25} \left(\frac{\Delta T}{H_{fg} \rho_v} \right)^{0.24} \quad (28)$$

Since for this research, it is assumed that there are no pressure drops the second term in Equation (28) vanishes. The parameter "F" can be calculated from the following equation

$$1_n^F = 1.0035 - 0.5962 \ln(x_{tt}) + 0.05415 \ln(x_{tt})^2 + 0.006046 \ln(x_{tt})^3 \quad (29)$$

where

$$x_{tt} = \frac{(1-x)^{0.9}}{x} \left(\frac{\rho_g}{\rho_l} \right)^{0.5} \left(\frac{\mu_l}{\mu_g} \right)^{0.1} \quad (30)$$

STEP 7 Calculation of the working fluid forced convection heat transfer coefficient in the condenser for both single and two phase flow is performed using Jakob's correlation. (4)

$$h_{WF3} = 0.725 \left[\frac{g \rho_L (\rho_L - \rho_V) k^3 h_{fg}^1}{\mu_L (\Delta T) D_e} \right]^{0.25} \quad (32)$$

where

$$h_{fg}^1 = \begin{array}{l} \text{latent heat} \\ \text{in condenser} \end{array} + 3/8 (cp) (\Delta T)$$

$$\Delta T = \text{assumed to be } 25^\circ \text{ F}$$

As discussed previously the overall weighted average heat transfer coefficient is assumed to be equal to the overall heat transfer coefficient for the working fluid in the condenser.

APPENDIX B

Procedure for Estimation of the Total Capital Investment

PROCEDURE FOR TOTAL CAPITAL INVESTMENT ESTIMATION

The steps involved in the estimation of the total capital investment in \$/kw are as follows.

The factored estimate method can be summarized by the following cost equation

$$C_n = \sum_i C_{ei} \left(1 + \sum_i f_i \right) \left(1 + \sum_i f_I \right) + n_w C_w \left(1 + \sum_i f_w \right) \left(1 + \sum_i f_I^* \right) \quad (33)$$

where

C_e = cost of major equipment (e.g., heat exchanger, condenser, etc.)

f_i = factors for estimation of direct expenses, such as piping, control, etc.

f_I = factors for estimation of indirect expenses, such as fees, escalation, etc.

n_w = number of wells required for a particular size plant.

C_w = cost of a geothermal production and/or reinjection well.

f_w = factor which accounts for piping from the well head to the power plant.

f_I^* = indirect cost factors, e.g., costs associated with drilling exploratory holes, contingencies, etc.

C_n = total capital investment.

BRINE SYSTEM COST

The cost of a geothermal well is dependent on the depth and diameter of the well and the type of rock. To simplify the calculations herein, a cost of \$ 500000 (c_w) for a well flowing at the rate of 500,000 lb per hour of brine has been used. Also a non-integer number of wells (n_w) has been used to allow the brine system cost to be a continuous function. Table B.1 shows particulars regarding the well cost.

COST ESTIMATING FACTORS

Table 5.1 gives the values of the direct cost multiplying factors (fractions of the major equipment and well costs). Table B.2 gives the indirect cost multiplying factors (fraction of the total equipment for well and piping cost) indirectly associated with the capital investment.

Table B.1

$$\sum_i C_{f_i} = C_{HE} + C_C + C_T + C_{cp} = \text{Equipment Total}$$

Additional Costs (including labor)	Fraction of $\sum_i C_{E_i}, f_i$
Equipment installation	0.5
Instrumentation/Control	0.15
Piping/Insulation	0.75
Insulation	0.02
Auxiliaries	0.1
Buildings/Structures/Concrete	0.15
Land/Improvements	0.10
Fire Control	0.05
Electrical (Smith Yard)	0.06
Environmental	0.05
Startup	0.05
Total $(1 + \sum_i f_i)$	3.00
$n_w c_w$ = Production and Reinjection Wells (Drilled and Cased)	
Additional costs (including labor)	Fraction $n_w c_w, f_w$
Piping (well head to plant)	
$n_{wells} \leq 6$	0.16
$6 < n_{wells} \leq 18$	0.24
$18 < n_{wells} \leq 36$	0.34
$36 < n_{wells} \leq 60$	0.44
$60 < n_{wells} \leq 90$	0.49
$n_{wells} > 90$	0.5

Table B.2

Indirect Cost Factors for
Geothermal Power Plants

Equipment related (C_E)	f_I
Engineering/Legal	0.15
Contingency	0.10
Overhead/Escalation	0.15
Environmental/Safety	0.10
Working Capital	0.15
$(1 + \sum_i f_I)$	1.65
Well resource discovery or exploration-related (C_W)	f_I^*
Land acquisition (leasing, legal fees)	0.19
Drilling exploratory holes (1 out of 4 successful)	0.14
Surface exploration (geophysical-geochemical)	0.10
Contingency	0.13
$(1 + \sum_i f_I^*)$	1.56

APPENDIX III

**EXCERPTS FROM THE PH.D. THESIS
OF K.Z. IQBAL (PAGES 175-238)**

CHAPTER V

ADVANTAGES AND DISADVANTAGES OF THE USE OF MIXTURES IN GEOTHERMAL BINARY CYCLES

The use of hydrocarbon mixture working fluids in the conventional geothermal binary cycle has been advocated in research carried out at the University of Oklahoma^(7,26,27); however, only isobutane-isopentane binary mixtures were considered in these studies. This study broadens the scope of investigation to other hydrocarbon systems (pure fluids and mixtures) and the use of these systems in cascade (multi-boiler) binary cycles. Advantages and disadvantages of the use of mixtures in the geothermal binary cycles are summarized.

A fixed plant simulator, the discussion of which is provided in Appendix B, was used in a preliminary study of the effects of georesource temperature decline on the performance of the pure and mixture binary cycles. A scoping study was also carried out for the hardware(equipment) evaluation of mixtures in the geothermal binary cycle.

Major Differences Between Pure and
Mixture Fluid Cycle Parameters

The purpose of this discussion is to point out the major differences between pure and mixture fluid cycle parameters. Isobutane-isopentane mixture cycles are compared with pure isobutane and pure isopentane cycles on temperature-entropy (T-s) diagrams for supercritical cycles in Figure 44.0 and subcritical cycles in Figure 45.0. Reversible cycles are shown to simplify the analysis and diagrams; irreversible behavior is similar.

The important behavioral characteristics of mixtures compared to the pure fluids can be noted in Figures 44.0 and 45.0 as follows:

- (1) At a constant pressure, mixtures condense nonisothermally whereas pure fluids condense isothermally.
- (2) At a constant pressure, mixtures vaporize nonisothermally whereas pure fluids vaporize isothermally.
- (3) The binary mixture condensation pressure lies between the condensation pressures of the constituent pure fluids (for a given dew point temperature).

Figure 44.0 Supercritical Cycles

(a)

(b)

(c)

Figure 45.0 Subcritical Cycles

- (4) The binary mixture turbine exit superheat lies between the turbine exit superheats of the constituent pure fluids (for a given dew point temperature).
- (5) The cost optimized turbine inlet pressures for mixture binary cycles lie between the turbine inlet pressures of the constituent pure fluids (for a given georesource temperature and specified net power).⁽²⁰⁾
- (6) The cost optimized brine heat exchanger and condenser LMTD's lie between the LMTD's of the constituent pure fluids (for a given georesource temperature and specified net power).

The first two behavioral characteristics are of primary importance when process and economic considerations are ignored, i.e., when only thermodynamic optimization is considered. The latter four behavioral characteristics are of primary importance when process and economic factors are considered. However, when process and economic considerations are ignored, a large fraction of the availability in the resource brine can be converted to power, as was pointed out previously.

Advantages and Disadvantages of Mixtures

The use of mixtures as working fluids in geothermal binary cycles provides certain advantages over cycles using

pure component working fluids. In most of the studies carried out at the University of Oklahoma, the comparisons of mixture and pure fluid cycles have ignored mass transfer effects. Ignoring mass transfer effects the heat transfer coefficients, obtained through the use of standard heat transfer correlations, are virtually equal for similar mixture and pure fluid cycles in these studies. The purpose of this discussion is to provide some understanding of the expected mass transfer effects on mixture binary cycles, and to show the advantages of mixtures in specific terms.

Basis of Comparison

For comparing various working fluid binary cycles, a common basis is needed to draw meaningful conclusions. In the cost optimization studies, carried out for single boiler binary cycles, the total system cost expressed in dollars per kilowatt net plant electrical output (25.0 MW for this study) was chosen as the basis. Other design basis engineering parameters are given in Appendix A. In the preliminary study of cascade (or multi-boiler) binary cycles, the basis of comparison is the net thermodynamic cycle work per unit mass of brine (for a specified brine flow rate, $1.04 \times 10^6 \text{ lb}_m/\text{hr}$).

Single Boiler Conventional Binary Cycle

The GEO4 simulator was used in all the calculations carried out for the single boiler conventional binary cycle. In this study, only binary mixtures of isobutane and isopentane are considered, for comparison with pure isobutane and pure isopentane cycles. To simplify the analysis, only one equimolar mixture of isobutane and isopentane is used here. The temperature of the resource brine chosen for the cost optimization study was 300°F and net plant power was specified to be 25.0 MW. A comparison between mixture and pure fluid cycle state points is given in Figure 46.0 on a superimposed temperature-enthalpy diagram for isobutane and the equimolar isobutane-isopentane mixture cycles for the case of the 300°F georesource temperature. The turbine inlet temperature and enthalpy for the mixture are both considerably higher than for isobutane. However, the mixture also has greater superheat and a higher enthalpy than isobutane at the turbine exit. Because the gain at the turbine inlet exceeds the loss at the turbine exit, the mixture cycle yields more gross turbine work per unit mass of working fluid than the isobutane cycle.

For the equimolar mixture of isobutane and isopentane, the turbine exit superheat would be greater than for isobutane and less than for isopentane. Thus, the overall

Figure 46.0 Superimposed Temperature-Enthalpy Diagram of Isobutane and Mixture (50% isobutane-50% isopentane) Cycle

condenser LMTD for the binary mixture would be between the pure fluid cycle condenser LMTD's. With respect to the brine heat exchanger, the near optimal LMTD for isopentane is lower than for isobutane. This is because the turbine inlet pressure to achieve a given turbine inlet temperature is smaller for isopentane than isobutane (by a factor of about one third), leading to lower brine heat exchanger cost per unit area and a smaller LMTD for the isopentane cycle. This lower cost per unit heat transfer surface area for the brine heat exchanger also allows a larger brine exit temperature for the economic optimum for the isopentane cycle. For the equimolar mixture of isobutane and isopentane, the near optimal brine heat exchanger LMTD's and brine exit temperatures fall between the pure fluid cycle values. It is interesting to note that the bubble point temperature of the working fluid in the BHE is virtually independent of working fluid composition (within a few degrees F) for a given georesource temperature (Figure 46.0). The fact that the isobutane-isopentane mixture vaporization curve is nonisothermal then yields a larger enthalpy at the turbine inlet than would be obtained for pure isobutane (Figure 46.0).

Thus, the larger enthalpy change across the turbine for the mixture cycle results in a larger net thermodynamic cycle specific work and higher net thermodynamic cycle efficiency, η_c , compared to pure isobutane or pure isopentane.

The increase in η_c implies a lower working fluid flow rate per unit brine flow rate for the mixture cycle. By lowering the working fluid to geothermal brine flow ratio, process equipment sizes and parasitic power requirements are reduced.

The cost advantages of mixtures considered here can be explained by referring to the effect of mixture properties on binary cycle operating conditions as noted in Table 16. By increasing the working fluid molecular weight (adding isopentane to isobutane) the dew point pressure is decreased below that of isobutane. This means that the condenser cost can be reduced compared to isobutane. The lower brine heat exchanger operating pressure compared to isobutane also permits a slightly lower required condenser operating pressure compared to isobutane to achieve a given net power output for the mixture cycles.

The quantity of cooling water required is smaller for the mixture cycle than the isopentane cycle because the turbine exit superheat for the mixture is smaller than for isopentane; mixture cycle cooling water requirements are smaller than the isobutane cycle because of larger cooling water rise in the condenser for the mixture cycle. The turbine-generator cost for the mixture cycle is lower than the isopentane cycle primarily because of the greater turbine enthalpy change and lower volumetric flow rate for the mixture cycle.

Table 16

COMPARISON OF PURE FLUID AND MIXTURE FLUID CYCLE PARAMETERS
FOR THE GEORESOURCE TEMPERATURE OF 300°F

Process Parameters	Fluid	Mixture:		
	Isobutane	Isobutane - 50% Isopentane - 50%	Isopentane	
Cycle				
Net Power, NW	25.00	25.00	25.00	
Gross Power, MW	31.08	29.78	29.92	
Net Plant Work, Btu/lb Brine	11.35	10.54	9.68	
Net Thermo. Efficiency, %	10.80	11.02	10.28	
Power Plant Cost, \$/kw	864	775	819	
Total System Cost, \$/kw	1449	1405	1504	
Brine Delivery				
Brine Flow, lb/kw	301.	323.8	352.6	
Brine Exit T, °F	182.6	193.1	192.0	
Brine Delivery Cost, \$/kw	585	629	685	
Brine Heat Exchanger (BHE)				
BHE LMTD, °F	36.7	35.3	37.8	
BHE Load ΔH, Btu/lb _m W.F.	163.6	189.2	180.2	
BHE Cost, \$/kw	264	229	191	
Condenser				
Cond. LMTD, °F	18.6	20.7	21.8	
Cond. Superheat ΔT, °F	26.0	39.8	43.2	
Cond. Superheat ΔH, Btu/lb	11.95	18.12	18.97	
Cond. Dew Point T, °F	108.0	119.3	109.0	
Cond. Dew Point P, psia	78.1	43.7	23.6	
Cond. Load ΔH, Btu/lb _m W.F.	145.8	168.3	161.6	
Cond. Cost, \$/kw	345	278	236	
Cooling System				
Cooling Water Flow, lb/kw	1773	1562	2164	
Cooling Cost, \$/kw	83	52	72	
Turbine				
Turbine Inlet P, psia	300	200	100	
Turbine Inlet T, °F	220	235	210	
Turbine ΔH, Btu/lb _m W.F.	19.37	21.90	19.05	
Working Fluid Flow, lb/kw	219.	185.7	214.4	
Exhaust Vol. Flow, ft ³ /min/kw	4.5	6.6	12.6	
Turbine-Generator Cost, \$/kw	146	177	283	
Cycle Pump				
Parasitic Power, MW	2.51	1.414	0.845	
Pump Cost, \$/kw	26	16	10	

Cascade Binary Cycle

This study was made primarily for the Idaho National Engineering Laboratory for the low temperature brine available at the proposed Raft River 5-MW geothermal pilot plant dual boiler binary cycle plant.⁽³⁾ The purpose of the study was the selection of a working fluid and suitable operating conditions for maximizing the net plant work per unit mass of brine in the Raft River geothermal dual boiling (cascade) binary cycle. Since it is believed that mixtures offer possible advantages over pure fluids for use as working fluids in cascade binary cycles, both pure fluids and mixtures are considered as working fluids in this study.

In the absence of a cascade binary cycle simulator, a simplified procedure for evaluation of the dual boiler binary cycle was developed. This procedure, outlined in Appendix C, utilizes the single boiler geothermal binary cycle simulator, GEO4, in the calculations for both pure fluids and mixtures as working fluids. The working fluids considered here include isobutane, cis-2-butene, and binary mixtures of cis-2-butene, propane and cyclopentane.

Dual boiler binary cycle process. The major elements of the dual boiler binary cycle power plant are shown in Figure 47.0. Other process units not shown in Figure 47.0

FIGURE 47.0 Dual Boiler Binary Cycle State Points

include brine wells and gathering system, cooling system and auxiliary plant equipment. The nodal points indicated in Figure 47.0 correspond to the process state points used in the calculation. The dual boiler process was described earlier, in chapter I; here the dual boiler binary cycle will be explained with reference to a $T-\dot{Q}$ diagram. To understand the dual boiler binary cycle shown in Figure 47.0, consider first a single boiler ideal binary cycle on a temperature-entropy diagram in Figure 48.0 (represented by solid lines). Since the objective here is the maximization of the net plant work per unit mass of brine, one would like to select the highest temperature and pressure conditions at turbine inlet (state point 1) without violating the pinch criterion in the heat exchanger (in order to increase the net plant work for specified brine flow rate conditions).

Consider now the revised single boiler ideal binary cycle (shown by dotted lines in Figure 48.0) where the turbine inlet temperature and pressure are increased to state point 5. But for this revised subcritical cycle, the brine flow rate would have to be increased in order to avoid the pinch occurring around state point 7'. The new brine outlet temperature will then be T'_{HO} (Figure 48.0). Neglecting the changes due to kinetic and potential energy, and assuming no viscous pressure drops, the net thermodynamic cycle specific work obtained from the working fluid will be

Figure 48.0 Temperature-Entropy Diagram of Single Boiler Binary Cycles

the difference in areas under the working fluid warming and cooling curves on a temperature-entropy diagram. The net cycle specific work obtained from the revised case and the original case will be represented by the difference in the areas 5-6-2'-3-7-7'-5 and 1-2-2'-3-4-4'-1. It is evident from Figure 48.0 that the area 5-6-2'-3-7-7'-5 is larger than the area 1-2-2'-3-4-4'-1. The difference in these areas is shown cross-hatched in Figure 48.0. Thus, the net cycle specific work, $\underline{W}_N$, increases for any increase in temperature and pressure of the working fluid at the turbine inlet. For subcritical cycles (such as the one shown in Figure 48.0), the pinch usually occurs (depending on the georesource temperature) near the bubble point temperature of the working fluid in the brine heat exchanger. For a fixed brine flow rate, the pinch will occur at the bubble point shown by state point 7' in Figure 48.0 when the pressure is increased. Therefore, the brine flow rate would have to be increased to avoid the violation of the pinch temperature criterion. This will result in a higher brine outlet temperature and, if the pressure is increased beyond the optimum, a decrease in resource utilization. Thus, maximization of the net plant work per unit mass of brine, $\underline{W}_N/\underline{M}_H$ is a trade-off between the higher turbine inlet pressure (for increasing the net cycle work) and lower brine flow rate (for increasing resource utilization).

For specified brine flow rate and brine outlet temperature (T_{HO}), the net cycle work also can be increased by utilizing cascade (or multi-boiler) binary cycles. To simplify this discussion, only the dual boiler binary cycle will be explained here with reference to the working fluid temperature profiles in the dual boilers. Figure 49.0 shows on a $T-\dot{Q}$ diagram the dual boiler binary cycle with reference to the brine heat exchanger. This diagram will also be used in reference to Figure 47.0 to keep the discussion short. The working fluid is partially vaporized by brine up to state point 6 in a low pressure boiler 2. The partially vaporized working fluid (pure fluid for this discussion) is then taken to a separator (Figure 47.0) where the working fluid vapor at state point 6 is taken from the top of the vessel and expanded in low pressure turbine 2 in Figure 47.0. The liquid working from the separator at state point 6 is then pumped to a higher pressure and passed through a high pressure boiler 1 (Figure 47.0) where the working fluid is vaporized and converted to saturated vapor or superheated gas, by heat transfer with the incoming brine from the production wells. This process of warming and vaporization is shown by the working fluid state points (temperature profile) 6-7-5 in Figure 49.0. The high pressure gas phase working fluid (at state point 5 in Figure 49.0) is then expanded through turbine 1 (Figure 47.0) for power generation.

Figure 49.0 T-Q Diagram of Cis-2-Butene

The nonisothermal vaporization and condensation behavior of mixtures (at constant pressure) can be utilized to advantage here by making the working fluid temperature profiles (in both boilers) come closer to being parallel to the brine profile and the working fluid temperature profiles in condensers come closer to being parallel to the cooling water profiles to obtain more area under the warming and cooling curves for the mixture cycle compared to the pure fluid cycle. This behavior of mixtures will be further discussed in the comparison of pure fluid and mixture dual boiler binary cycle parameters. However, the point to keep in mind is that more work is obtained per unit mass of brine in going from the single boiler binary cycle to the dual boiler binary cycle. Theoretically, an infinite number of boilers would make the working fluid profile exactly parallel to the brine profile, even for a pure fluid. A very sufficiently wide boiling range mixture can do the same job, i.e., make the working fluid and brine profiles parallel with a finite number of boilers. However, the wide boiling mixture will have an even wider condensing range, which would be disadvantageous to a single or perhaps even dual boiler mixture cycle. Cascade cycles using more than two boilers will not be discussed here. Instead, the advantages of mixtures in the dual boiler binary cycles will be addressed.

Comparison of pure hydrocarbon and mixture working fluid for the dual boiler binary cycle. The objective function for comparing pure hydrocarbon and mixture working fluids was chosen as the net thermodynamic cycle work (turbine work-cycle pump feed work) per unit mass of brine. This choice was made because cost calculations for the dual boiler cycle would not be made easily. However, as noted earlier, for the lower georesource temperatures, the cost optimum and resource utilization optimum nearly coincide. The cycle operating conditions were determined using the simplified hand calculation procedure outlined in Appendix C. The design parameters used to conduct this sensitivity study also are listed in Appendix C.

The parameters which can be chosen for optimization in the dual boiler process include the following:

- (1) Working Fluid Starting Composition
- (2) Vapor/Liquid Split in Boiler 1
- (3) Turbine Inlet Temperature and Pressure in Low and High Pressure Cycles
- (4) Brine Outlet Temperature
- (5) Cooling Range in Condensers 1 and 2
- (6) Pinch Point Temperature Differences

A few calculations were performed to optimize the vapor-liquid split in boiler 2 for one binary mixture of propane-cis-2-butene. The semi-optimum vapor/liquid split

obtained for this mixture was then utilized for other binary mixtures. A mixture of 60 mole per cent cis-2-butene and 40 mole per cent cyclopentane was chosen for the first calculation, because both fluids have I-factors close to unity. The vapor-liquid split for this mixture was chosen so as to give approximately an equimolar mixture of cis-2-buten and cyclopentane in the high pressure cycle. The selection of a 50-50 mixture was based on the fact that such mixtures give the maximum slope on the temperature-enthalpy vaporization curve and thus higher turbine inlet temperature. However, because of the greater slope of the saturated liquid locus, for this mixture compared to cis-2-butene, the latent heat of vaporization at a given temperature was greater than that of pure cis-2-butene resulting in a higher brine outlet temperature for this mixture (than cis-2-butene). To reduce the slope of the saturated liquid locus, propane and cis-2-butene mixtures were chosen for further study.

No attempt was made in this preliminary study to optimize the parameters 4, 5 and 6 mentioned above.

Table 17 gives a summary of the most significant parameters for the comparison of pure and mixture dual boiler cycles. Table 18 provides a more detailed comparison of various other cycle parameters. Pure cis-2-butene was selected as the basis of comparison. Two mixtures (i.e.

Table 17

**SUMMARY OF THE DUAL BOILER BINARY CYCLE
PRELIMINARY CALCULATIONS**

Working Fluid Parameters	Cis-2-Butene	Isobutane	Mixture I*			Mixture II	Mixture III	Mixture IV
	Vapor Fraction [†] (40%)	Vapor Fraction [†] (40%)	Vapor Fraction [†] (29%)	Vapor Fraction [†] (40%)	Vapor Fraction [†] (51%)	Vapor Fraction [†] (41%)	Vapor Fraction [†] (41%)	Vapor Fraction [†] (38.6%)
Net Plant Work, Btu/lb _m brine	18.49	20.16	21.85	22.46	21.87	22.27	20.85	20.17
Ratio of Net Plant Work**	1.00	1.09	1.18	1.21	1.18	1.20	1.13	1.09
Net Power, MW	6.046	6.595	6.966	6.952	7.055	7.349	6.784	6.594
Gross Power, NW	6.315	7.005	7.470	7.516	7.582	8.008	7.187	6.685
Brine Flow, (10 ⁶)lb _m /hr	1.116	1.116	1.088	1.056	1.101	1.126	1.110	1.116
Brine Exit T, °F	137.	138.5	132.5	130.	132.5	132.5	139.	144.5

* Mixture I Initial Composition: Propane, 60%; Cis-2-Butene, 40%.
 Mixture II Initial Composition: Propane, 70%; Cis-2-Butene, 30%.
 Mixture III Initial Composition: Propane, 50%; Cis-2-Butene, 50%.
 Mixture IV Initial Composition: Cis-2-Butene, 60%; Cyclopentane, 40%.

** Net Plant Work in Btu/lb_m brine.

[†] Vapor Fraction is the Vapor Split used in Boiler 2.

Table 18

COMPARISON OF PURE AND MIXTURE FLUID DUAL BOILER
BINARY CYCLE PRELIMINARY CALCULATIONS

Parameters \ Working Fluid	Cis-2-Butene	Isobutene	Mixture I*			Mixture II	Mixture III	Mixture IV
	Vapor Fraction (40%)	Vapor Fraction (40%)	Vapor Fraction (29%)	Vapor Fraction (40%)	Vapor Fraction (51%)	Vapor Fraction (41%)	Vapor Fraction (41%)	Vapor Fraction (38.6%)
Net Plant Work, Btu/lb _m brine	18.49	20.16	21.85	22.46	21.87	22.27	20.85	20.17
Net Power, MW	6.046	6.595	6.966	6.952	7.055	7.349	6.784	6.594
Gross Power, MW	6.315	7.005	7.470	7.516	7.582	8.008	7.187	6.685
Net Thermo. Efficiency, %	12.08	13.31	13.87	14.04	13.88	14.14	13.81	13.86
Brine Flow, (10 ⁶) lb _m /hr	1.116	1.116	1.088	1.056	1.101	1.126	1.110	1.116
Brine Exit T, °F	137.	138.5	132.5	130.	132.5	132.5	139.	144.5
Total Boiler Duty, (10 ⁷) Btu/hr	17.078	16.912	17.143	16.9049	17.342	17.7384	16.7676	16.2332
Boiler 1 Duty, (10 ⁷) Btu/hr	8.9052	7.4976	8.4484	6.9029	6.0701	7.4762	7.2588	8.1455
Boiler 2 Duty, (10 ⁷) Btu/hr	8.1733	9.4140	8.6945	10.002	11.2717	10.2621	9.5088	8.0876
Total Cond. Duty, (10 ⁷) Btu/hr	14.984	14.661	14.776	14.538	14.935	15.410	14.4972	13.9705
Condenser 1 Duty, (10 ⁷) Btu/hr	9.015	8.8484	10.5907	8.676	7.4398	9.2732	8.7827	8.7049
Condenser 2 Duty, (10 ⁷) Btu/hr	5.968	5.8122	4.1850	5.8618	7.4957	6.1368	5.7145	5.2655
Turbine 1 Inlet Pressure, psia	220.4	372.0	470.0	560.0	572.2	620.0	438.7	132.0
Turbine 1 Inlet Temp., °F	218.0	240.0	242.0	260.0	275.0	275.0	262.0	240.0
Turbine 2 Inlet Pressure, psia	104.3	190.0	312.0	312.0	312.0	310.0	275.0	81.4
Turbine 2 Inlet Temp., °F	156.0	177.0	180.7	183.0	185.0	182.6	185.8	175.4
Turbine 1 ΔH, Btu/lb _m W.F.	29.08	27.57	30.69	32.73	35.63	35.41	34.38	35.22
Turbine 2 ΔH, Btu/lb _m W.F.	15.09	17.07	20.25	20.60	20.94	19.49	20.29	20.82
Total W.F. Flow, (10 ⁵) lb _m /hr	9.1783	10.23	9.2142	9.2142	9.2142	9.450	8.5691	7.6915
W.F. Flow in Turbine 1, (10 ⁵) lb _m /hr	5.5070	6.138	6.5476	5.4981	4.4819	5.5995	5.0690	4.7236
W.F. Flow in Turbine 2, (10 ⁵) lb _m /hr	3.6713	4.092	2.6667	3.7161	4.7324	3.8505	3.500	2.9680

- Mixture I Initial Composition: Propane, 60%; Cis-2-Butene, 40%.
 Mixture II Initial Composition: Propane, 70%; Cis-2-Butene, 30%.
 Mixture III Initial Composition: Propane, 50%; Cis-2-Butene, 50%.
 Mixture IV Initial Composition: Cis-2-Butene, 60%; Cyclopentane, 40%.

mixtures I and II) of propane and cis-2-butene provide 20% or more net plant work compared to pure cis-2-butene and almost 11% more than pure isobutane. The overall improvement in net plant work of all other mixtures over cis-2-butene ranges between 9 and 18%. Mixture IV gives almost identical results to isobutane. However, the operating pressures for the heat exchanger and condenser are much lower (almost 1/3) than those for the isobutane cycle. This would mean lower costs for heat transfer units. However, somewhat higher turbine costs can be expected for this mixture compared to isobutane. Although cost calculations were not performed, it is evident that whether the objective is maximum utilization of the resource or cost of major equipment, mixtures do offer advantages over the pure fluids considered. The net power obtained from mixture cycles is also equal to or significantly greater than the pure fluids considered in this analysis, as can be noted in Table 17.

Figure 50.0 illustrates on a T-Q diagram the brine and working fluid profiles of pure cis-2-butene and a propane-cis-2-butene mixture. The cycle lost work is minimized by the use of mixtures because of their non-isothermal vaporization profiles in the brine heat exchangers (at a constant pressure).

It is interesting to note from Table 17 that the brine outlet temperature for mixture I (vapor fraction of 40%) is the lowest for all mixtures and pure fluids

Figure 50.0 T- $\dot{Q}$ Diagram of Pure and Mixture Fluid

considered in this study, and at the same time it gives the best performance. Since minimizing the geothermal brine outlet temperature simultaneously aids both power conversion goals, resource utilization efficiency and minimum cost, the brine outlet temperature is an important parameter in cycle optimization studies. Another factor which should be noted is the fact that it is possible to adjust the cooling range for a mixture, thus reduction the cooling water flow requirements, whereas it is not possible to adjust the cooling range for a pure fluid.

Another significant result to note from Table 17 is the compositions of mixture I and II. Both mixtures give almost the same amount of improvement over the pure fluids. Mixture I is a subcritical cycle whereas mixture II is a supercritical cycle, so that there is a range of operating conditions and working fluid compositions which give the best performance. Such a range of operating conditions is not possible for a pure fluid. In other words, the performance of these mixtures are relatively insensitive to changes in composition between 60 and 70 mole per cent propane in cis-2-butene. This fact can be of immense value in adjusting the working fluid composition to match changes in the geothermal resource temperature over the plant lifetime. Other parameters given in Tables 17 and 18 are self explanatory and so will not be discussed here.

Comparison of pure hydrocarbon and mixture working fluids for multiple boiler binary cycle. The purpose of this discussion is to show the relationship of the number of boilers (in the multiple boiler binary cycle) versus the net plant work per unit mass of brine, and versus net thermodynamic cycle efficiency. The working fluids for comparison in this study are pure isobutane and a mixture of 60% propane and 40% cis-2-butene (referred to as mixture I, with vapor fraction of 0.4 in Table 17).

Tables 19 and 20 give a summary of the results for isobutane and mixture I multiple boiler cycles. Only isobutane triple boiler calculations were performed. From Tables 19 and 20 the following points can be noted:

- (1) For the single boiler binary cycle, mixture I yields approximately 18% greater net plant work than isobutane. The net thermodynamic cycle efficiency is about 13% higher for the mixture cycle. The single boiler binary cycle calculations were not optimized. However, the conclusions should not change for optimized cases.
- (2) For the dual boiler cycle, mixture I provides approximately 11% more net plant work than isobutane. The increase in the net plant work from the single boiler cycle to the dual boiler cycle is about 18% for the mixture I and 25% for isobutane.

Table 19

NET PLANT WORK PER POUND OF BRINE

Working Fluid \ No. of Boilers	No. of Boilers		
	1	2	3
Mixture I*	19.04	22.46	---
Isobutane	16.15	20.16	22.33

Table 20

NET THERMODYNAMIC CYCLE EFFICIENCY, PERCENT

Working Fluid \ No. of Boilers	No. of Boilers		
	1	2	3
Mixture I*	11.74	14.0	---
Isobutane	10.36	13.3	13.13

* Mixture I initial composition:

Propane = 60%

cis-2-Butene = 40%

- (3) For the triple boiler cycle, only isobutane was considered. The net plant work for this cycle is 22.3 Btu/lb_m of brine, which is 10% greater than the isobutane dual boiler cycle, but still less than the dual boiler mixture cycle. The slight decrease in η_c for triple boiler cycle is due to the fact that brine heat exchanger duty increases as the number of boilers are increased. However, the isobutane cycle is not optimized and therefore η_c is inconsistent with increase in W_N .

Figure 51.0 and 52.0 show the results in Tables 19 and 20 graphically. It can be seen from Figure 51.0 that the net plant work gained from a triple boiler binary cycle is very close to the maximum work which can be obtained from an infinite boiler binary cycle. Also, the increase in net plant work is greater when the number of boilers is increased from 1 to 2. From the slope of the curve in Figure 51.0 it can be anticipated that mixture I will yield a greater net plant work than isobutane for the triple boiler cycle. However, the percentage of excess work will decrease as the number of boilers is increased until mixture and pure fluid cycles become equal. A point to note is the fact that as the number of boilers is increased, the number of optimization variables also will increase. Mixture cycles will always have more variables to be optimized than pure

Figure 51.0 Net Plant Work Versus Number of Boilers for Multiple Boiler Binary Cycles

Figure 52.0 Net Thermodynamic Cycle Efficiency versus Number of Boilers for Multiple boiler Binary Cycle

fluid cascade cycles. Therefore, the mixture cycles beyond the dual boiler cycle will become quite complicated for computer simulation and/or optimization.

In summary, the net plant work and thermodynamic efficiency for the mixture dual boiler cycle are greater than for the isobutane triple boiler cycle.

Disadvantages of the use of mixtures in single boiler and multiple boiler binary cycles. The comparisons presented for mixture and pure fluid cycles have so far ignored mass transfer effects. Standard heat transfer correlations have yielded heat transfer coefficients which are within a few percent for similar mixture and pure fluid cycles in these case studies.⁽²⁰⁾ Thus, only mass transfer effects and the related fact that vapor and liquid in two-phase flow are of different composition would be expected to detract from the advantages of mixture cycles over pure fluid cycles noted in these case studies (which ignored mass transfer effects).

Mass transfer effects which have the major influences on cycle performance are those mass transfer effects which occur in the two-phase regions of the brine heat exchanger and condenser. Two types of mass transfer occur, (1) bulk mass transfer occurs for both mixtures and pure fluids in evaporation and condensation and is not considered here. Diffusive mass transfer occurs only for mixtures in evaporation

and condensation. Because for mixtures, the coexisting vapor and liquid (at equilibrium) are of different compositions, concentration gradients are established in both the vapor and liquid phases (as the system attempts to achieve equilibrium). Because the major driving force for the diffusive mass flux of a component in a single phase mixture is the component concentration, it follows that mass transfer effects will be greatest when composition differences between liquid and vapor are greatest. Thus, mass transfer effects would be expected to be larger for mixtures of widely different fluids, such as ammonia-water mixtures than for mixtures of similar fluids, such as propane normal butane mixtures.

In a very preliminary study carried out at the University of Oklahoma regarding the diffusive mass transfer effects in geothermal binary cycles,⁽²²⁾ it was pointed out that there is a probability that for many working fluid mixtures, mixture boilers will be slightly smaller than boilers for the corresponding pure fluids (for equivalent heat transfer coefficients and LMTD's). On the other hand, it was concluded that because of the resistance to diffusive mass transfer in the vapor phase (of working fluid in condenser), mixture condensers will be larger than in the absence of diffusive mass transfer. However, the net disadvantages of the diffusive effects of mixtures in geothermal binary cycle should be offset by the advantages

mixtures exhibit with respect to their thermodynamic behavior. Besides these considerations, it should be noted that the commercial grades of isobutane and other pure working fluids actually are mixtures. Thus, mixture effects will occur even in "pure fluid" cycles.

Since working fluid heat transfer coefficients in the condenser are smaller than for pure fluids,⁽²⁰⁾ the negative effects of mass transfer will further increase the mixture fluid condenser sizes.

Effects of the Georesource Temperature

Decline on Pure Fluid and Mixture

Conventional Binary Cycle Performance

The effects on the net plant power, net plant work per unit mass of brine, net thermodynamic cycle efficiency and cooling water requirements, caused by an anticipated decline in the geothermal brine temperature (also design temperature) over the lifetime of power plants, are described with reference to pure isobutane and mixture (an equimolar mixture of isobutane and isopentane referred here as mixture I) binary cycles. Both pure isobutane and mixture cycles were cost optimized (\$/kw) for the 350°F georesource and are used here as the base cases.

Geothermal reservoirs are finite sources of thermal energy. The rate of decline of the available brine temperature with time depends on the withdrawal rate of brine and

the thermal energy which it carries. If the withdrawal rate of the brine exceeds the replacement (i.e., brine reinjection) rate, the temperature of the brine would be expected to decline with time. Thus, geothermal power plants of larger capacities would cause greater reservoir temperature decline and require greater unit design modifications. Hankin and others⁽²⁹⁾ utilized a reservoir computer model to study the effects of reservoir temperature decline on the two-stage flashed steam energy conversion process for various plant capacities. Hankin determined that for their base case plant, designed to produce 200 MW_e under assumed constant brine temperature conditions, the net plant output dropped to 68 percent of the initial capacity in 30 years if subjected to time-wise brine temperature decline. This declining power output resulted in a decrease in lifetime electrical energy output of almost 14 percent and an increase in levelized bus-bar electric energy cost (in mills/kwh) of over 12 percent compared to constant brine temperature operation.

It can be anticipated that a decline in the available temperature of geothermal brine as a function of time would either cause the power output of the plant to also decline, or the flow rate of brine would have to be increased in order to maintain the initial rated plant capacity. The objective of this preliminary study is to present a simplified analysis of the reservoir temperature decline (for fixed brine flow rate condition) on the performance of pure and

mixture binary cycles, and then to check the conclusions using the fixed plant simulator developed by the author (see Appendix B for details).

Consider first brine heat exchanger duty, $\dot{Q}_H$, which for the base case can be written as

$$\dot{Q}_H = U_H A_H (\text{LMTD}_H) \quad (94)$$

For decreased brine temperature, case 1

$$\dot{Q}_{H1} = U_{H1} A_H (\text{LMTD}_H)_1 \quad (95)$$

Because of decline in T_{HI} (brine temperature),

$$(\text{LMTD}_H)_1 < \text{LMTD}_H$$

and

$$U_H \simeq U_{H1}$$

it follows then

$$\dot{Q}_{H1} < \dot{Q}_H \quad (96)$$

Thus, brine heat exchanger duty decreases for a decrease in T_{HI} . We also know that for the base case

$$\dot{Q}_H = \dot{M}_H \bar{C}_P (T_{HI} - T_{HO}) \quad (97)$$

and for case 1

$$\dot{Q}_{HI} = \dot{M}_H \bar{C}_P (T'_{HI} - T'_{HO}) \quad (98)$$

therefore

$$\frac{\dot{Q}_{HI}}{\dot{Q}_H} = \frac{T'_{HI} - T'_{HO}}{T_{HI} - T_{HO}}$$

since

$$\frac{\dot{Q}_{HI}}{\dot{Q}_H} < 1$$

it follows that

$$\frac{T'_{HI} - T'_{HO}}{T_{HI} - T_{HO}} < 1$$

or

$$(T'_{HI} - T'_{HO}) < (T_{HI} - T_{HO})$$

But

$$T'_{HI} < T_{HI}$$

therefore

$$T'_{HO} > T_{HO} \quad (99)$$

The brine exit temperature, therefore, increases for a decrease in resource temperature. The condenser duty, $-\dot{Q}_C$, also decreases because of decrease in $\dot{Q}_H$. To verify this,

we have for the base case

$$\dot{Q}_H = \dot{M}_W \Delta H_{\text{BHE}} \quad (100)$$

and for case 1

$$\dot{Q}_{H1} = \dot{M}'_{W1} \Delta H'_{\text{BHE}} \quad (101)$$

since

$$\Delta H_{\text{BHE}} \approx \Delta H'_{\text{BHE}}$$

where ΔH_{BHE} and $\Delta H'_{\text{BHE}}$ are, respectively, working fluid specific enthalpy change in the brine heat exchanger for the base case and case 1.

We have

$$\frac{\dot{M}_{W1}}{\dot{M}_W} = \frac{\dot{Q}_{H1}}{\dot{Q}_H}$$

or

$$\frac{\dot{M}_{W1}}{\dot{M}_W} < 1$$

Thus,

$$\dot{M}_{W1} < \dot{M}_W$$

The condenser duty for case 1 is

$$-\dot{Q}_{c1} = \dot{M}_{W1} \Delta H_c$$

Because $\dot{M}_{W1} < \dot{M}_W$ and $\Delta H_{c1} \approx \Delta H_c$

$$-\dot{Q}_{c1} < -\dot{Q}_c \quad (102)$$

where $-\dot{Q}_c$ is the condenser duty for the base case. This analysis shows that the condenser duty decreases for a decrease in the resource temperature. A related effect of condenser duty can be seen on the condenser $LMTD_{c1}$ as follows:

$$-\dot{Q}_{c1} = U_{c1} A_c LMTD_{c1} \quad (103)$$

Since

$$U_{c1} = U_c$$

and

$$-\dot{Q}_{c1} < -\dot{Q}_c$$

it follows that

$$LMTD_{c1} < LMTD_c \quad (104)$$

For the pure fluid, the condenser $LMTD$ would decrease only if the approach temperature, $DTCWO$, would decrease. The decrease in $DTCWO$ means a decrease in the dew point temperature and pressure of the condenser. The decrease in the dew

point temperature (for fixed turbine inlet T , P) means an increase in the turbine specific work and an increase in the net thermodynamic cycle efficiency (as was noted in Chapter II, earlier). Thus, a decrease in the resource temperature implies an increase in the net thermodynamic cycle efficiency. The working fluid flow rate decreases more rapidly than the increase in the turbine specific work, resulting in a decrease in the net plant power, $\dot{W}_{N1}$. The net plant work per unit mass of brine, $\dot{W}_{N1}/\dot{M}_H$ would therefore decrease, for a decrease in the resource temperature. The decrease in the condenser duty for case 1 implies less cooling water requirements for cooling purposes. Tables 21 and 22 show the effects of georesource temperature decline on the performance of pure isobutane and mixture I binary cycles. The base cases for the 350° resource were revised (as explained in Chapter IV, in the brine flow rate section) for both pure isobutane and mixture I cycles. Therefore, the results presented in Tables 21 and 22 are in slight error. However, the overall conclusions drawn on relative basis would still hold. It is evident from Tables 21 and 22 that $\dot{W}_N/\dot{M}_H$ decreases for both pure fluid (isobutane) and mixture I cycles when the resource temperature declines from 350°F to 330°F. The net power decreases by approximately 13% and 17% for isobutane and mixture cycles, respectively.

Table 21

EFFECT OF GEORESOURSE TEMPERATURE DECLINE ON THE
PERFORMANCE OF ISOBUTANE BINARY CYCLE**

Georesource T, °F Process Parameters	Isobutane		Mixture II*
	350	330	330
Cycle			
Net Power, MW	24.63	21.43	19.45
Gross Power, MW	30.70	26.29	22.07
Net Plant Work, Btu/lb _m brine	18.21	15.85	14.38
Net Thermo. Efficiency, %	12.10	12.52	12.65
Turbine			
Turbine Inlet P, psia	450	450	200
Turbine Inlet T, °F	260	260	254
Turbine ΔH, Btu/lb _m	21.90	22.90	26.26
Working Fluid Flow, 10 ⁶ lb _m /hr	4.7838	3.919	2.8677
Brine Delivery			
Brine Flow, 10 ⁶ lb _m /hr	4.6153	4.6153	4.6153
Brine Exit T, °F	187.4	194.4	208.8
Brine Heat Exchanger (BHE)			
BHE LMTD, °F	47.4	41.45	45.05
BHE Load ΔH, Btu/lb _m W.F.	159.92	162.56	198.93
BHE Duty, 10 ⁸ Btu/hr	7.6504	6.3716	5.7048
Brine Velocity, ft/sec	7.0	7.0	7.0
Condenser			
Cond. LMTD, °F	23.67	19.18	19.51
Cond. Load ΔH, Btu/lb _m W.F.	140.58	142.20	173.76
Cond. Duty, 10 ⁸ Btu/hr	6.7250	5.574	4.9829
Cond. Dew Point T, °F	115.5	110.8	114.6
Cond. Dew Point P, psia	86.9	81.2	31.7
Cond. Superheat ΔT, °F	17.1	18.4	52.6
Cooling Water Velocity, ft/sec	6.94	5.75	5.14
Cooling System			
Cooling Water Flow, 10 ⁷ lb _m /hr	3.3597	2.780	2.4894
Cooling Water Pump Power, MW	1.55	1.12	0.94
Cycle Pump			
Cycle Pump Power, MW	3.586	2.92	0.917
Cycle Pump ΔH, Btu/lb _m	2.56	2.54	1.09

* Mixture II Composition: Isobutane, 25%; Isopentane, 75%

** Fixed Heat Transfer Surface Areas: BHE Area, 68260 ft²;
Condenser Area, 177,820 ft².

Table 22
EFFECT OF GEORESOURCE TEMPERATURE DECLINE ON THE
PERFORMANCE OF MIXTURE BINARY CYCLE[†]

Process Parameters	Georesource T, °F	Mixture I *		Mixture II **
		350	330	330
Cycle				
Net Power, MW		24.41	20.21	20.65
Gross Power, MW		28.50	23.25	23.60
Net Plant Work, Btu/lb Brine		17.16	14.2	14.51
Net Thermo. Efficiency, ^m %		13.02	13.54	12.66
Turbine				
Turbine Inlet P, psia		300	300	200
Turbine Inlet T, °F		273	273	254
Turbine ΔH, Btu/lb _m		26.79	28.83	26.37
Working Fluid Flow, 10 ⁶ lb _m /hr		3.5511	2.752	3.0535
Brine Delivery				
Brine Flow, 10 ⁶ lb _m /hr		4.854	4.854	4.854
Brine Exit T, °F		208.6	218.5	206.8
Brine Heat Exchanger				
BHE LMTD, °F		46.7	40.31	44.24
BHE Load ΔH, ⁸ Btu/lb W.F.		197.05	200.83	199.67
BHE Duty, 10 ⁸ Btu/hr ^m		7.0189	5.528	6.0972
Brine Velocity, ft/sec		7.0	7.0	6.98
Condenser				
Cond. LMTD, °F		23.2	17.34	18.29
Cond. Load ΔH, ⁸ Btu/lb W.F.		171.92	173.63	174.40
Cond. Duty, 10 ⁸ Btu/hr ^m		6.1051	4.779	5.3255
Cond. Dew Point T, °F		121.6	115.7	114.2
Cond. Dew Point P, psia		45.4	41.2	31.5
Cond. Superheat ΔT, °F		48.4	50.4	52.71
Cooling Water Velocity, ft/sec		6.74	5.28	5.88
Cooling System				
Cooling Water Flow, 10 ⁷ lb _m /hr		3.050	2.752	2.6605
Cooling Water Pump Power, MW		1.47	0.963	1.155
Cycle Pump				
Cycle Pump Power, MW		1.72	1.322	0.98
Cycle Pump ΔH, Btu/lb _m		1.656	1.64	1.096

[†]Fixed Heat Transfer Surface Areas: BHE Area, 74062 ft²; Condenser Area, 197310 ft²

^{*}Mixture I Composition: Isobutane, 50%; Isopentane, 50%

^{**}Mixture II Composition: Isobutane, 25%; Isopentane, 75%

It was argued earlier, in Chapter IV, that a mixture composition can be changed over the plant lifetime to match the changing characteristics of the resource. The main objective of this section is to provide the understanding and discussion of the procedure used here for the selection of optimum working fluid mixture composition for a declining georesource temperature. Figure 53.0 (taken from reference 20) shows a plot of near optimum average molecular weight, MW, (for the special case of the paraffin hydrocarbon working fluids and cost formulas utilized) as a function of georesource temperature. Figure 54.0 shows other near optimal characterization parameters (critical temperature, T_c , critical density, ρ_c , and acentric factor, ω) as a function of georesource temperature. These quantities are calculated using the formulas

$$MW = \sum Z_i (MW)_i$$

$$T_c = \sum Z_i T_{c_i}$$

$$\rho_c = \sum Z_i \rho_{c_i}$$

$$\omega = \sum Z_i \omega_i$$

where $(MW)_i$, T_{c_i} , ρ_{c_i} , ω_i and Z_i are, respectively, molecular weight, critical temperature, critical density, acentric factor and mole fraction of the i th component and summations range over all components in the mixture (for a pure working fluid, there is only one term in the sum).

Figure 53.0 Molecular Weight as a Function of Georesource Temperature

Figure 54.0 Critical Temperature, Critical Density and Accentric Factor as a Function of Georesource Temperature

Table 23.0 shows the values of these characterization parameters for the optimal working fluids determined in an earlier study. (20)

For a 20°F decline in the resource temperature, i.e., for the 330°F georesource, the optimum working fluid molecular weight was read from Figure 53.0 as 68.6. Other characterization parameters were read from Figure 54.0 as follows: T_c , 345.0; ρ_c , 0.211; ω , 0.21. These characterization parameter values correspond closely to a binary mixture of isobutane (25%) and isopentane (75%) in Table 23.0. This mixture will be referred to as mixture II. After selecting the near optimal mixture composition for the 330°F georesource, the near optimal turbine inlet pressure was determined to be 200 psia for mixture II from Figure 55.0; which shows near optimal turbine inlet pressure as a function of molecular weight and georesource temperature. Figure 56.0 is another way of representing Figure 55.0. The curve of near optimal molecular weight working fluids for the 330°F georesource was constructed using interpolated values of optimum turbine inlet pressures from Figure 55.0. Thus, Figures 53, 54, 55 and 56 suggest that at 330°F georesource, mixture II should give the best performance compared to other working fluids considered in these plots.

To verify the above prediction, mixture II was simulated using fixed heat transfer surface areas of mixture I and isobutane cycles and the results are shown in Tables 21

Table 23

NEAR OPTIMAL WORKING FLUID PARAMETERS FOR
GEORESOURCES TEMPERATURE RANGE 300°F-500°F

Georesource Temperature (°F)	Compound	Molecular Weight	Pseudo Critical Density (lb mole/ft ³)	Pseudo Critical Temperature (°F)	Pseudo Accentric Factor
300	iC ₄ H ₁₀ = 50% iC ₅ H ₁₂ = 50%	65.133	0.220	321.98	0.2045
350	iC ₄ H ₁₀ = 25% iC ₅ H ₁₂ = 75%	68.639	0.21135	345.49	0.21525
400	iC ₄ H ₁₀ = 15% iC ₅ H ₁₂ = 85%	70.042	0.20789	354.89	0.22165
500	iC ₅ H ₁₂	72.146	0.2027	369.0	0.226

Figure 55.0

Near Optimal Turbine Inlet Pressure Versus Molecular Weight and Georesource Temperature

Figure 56.0

Near Optimal Turbine Inlet Pressure as a Function of Georesource Temperature and Molecular Weight

and 22. Mixture II yields approximately 2% (or 0.41 MW) more net plant power than for mixture I (when using mixture I heat transfer areas and brine flow rate) for the 330°F georesource. When isobutane cycle brine flow rate and heat transfer areas were utilized, mixture II yielded approximately 4.2% less net power than for isobutane cycle for the 330°F georesource. Also, the net plant work per unit mass of brine increased for mixture II cycle compared to mixture I cycle, but decreased compared to isobutane cycle.

These preliminary results demonstrate the fact that mixture composition and cycle operating conditions can be varied over the lifetime of the plant to an advantage to match the changing resource characteristics. Another point to note is that power plant equipment which is originally designed for a mixture cycle would benefit other higher molecular weight mixtures and cycle operating conditions. On the other hand, the power plant equipment which is originally designed for a pure fluid may not benefit (higher molecular weight) mixture cycles.

Although Figures 53.0, 54.0, 55.0 and 56.0 can be used for working fluid selection and parameters such as turbine inlet pressure, caution should be exercised in such use of these results. The consideration of other classes of working fluids (such as halocarbons) will introduce additional factors (such as dipole moment effects) and the consideration of different equipment types and/or brine system

and equipment cost formulas will cause translation and warping of the plots of various parameters studied. Nevertheless, the study presented here provides perspective regarding the behavior and performance of binary mixtures compared to pure fluids in the binary cycle for a declining georesource temperature.

Figure 57.0 (shown on the next page and the analysis required to construct it) demonstrates that a decrease in the resource temperature always yields less net plant work per unit mass of brine, $\dot{W}_N/\dot{M}_H$ and that the net plant power decreases. This implies that a provision should be made for the georesource temperature decline in the design of geothermal power plant.

Equipment Needs Peculiar to Geothermal Binary Cycles Using Mixture Working Fluids

The use of mixture working fluids in geothermal binary cycles has been proposed in research at the University of Oklahoma^(7,20,26,27). To date, only horizontal, countercurrent, single pass shell and tube exchangers and condensers have been considered in these studies; with the working fluid on the shell side and brine and cooling water on the tube side of heat exchanger and condenser, respectively. In this report, other types of heat exchange equipment are considered which may enhance the performance of mixture working fluids in geothermal binary cycles. Other

Fixed A_H , A_C , $\dot{M}_H$

Figure 57.0 Effect of Georesource Temperature Decline (for Fixed A_H , A_C , $\dot{M}_H$) on Pure and Mixture Binary Cycles

factors taken into consideration are: types of flow, working fluid location, tube arrangement.

Brine Heat Exchanger

The conclusions from studies to date can be summarized as follows:

- (1) Only one shell pass and one tube pass, horizontal, counter-current heat exchangers have been considered. Because of non-isothermal heat transfer between the working fluid and the hot brine, counter-current flow will have a distinct thermal advantage over the co-current flow because in co-current flow, the hot fluid (brine in this case) cannot be cooled below the cold fluid (working fluid in this case) outlet temperature (30).
- (2) If brine fouling can be controlled, or if there is little fouling, then the working fluid should be on the tube side and the brine on the shell side. The assumption of working fluid phase equilibrium at each point in the two-phase section of the heat exchanger may be adequate if the working fluid is on the tube side. This would be a good assumption if the slip velocity, $V_s = V_g - V_l$, were zero; i.e., both gas and liquid phases travelling at the same velocity.

V_s is dependent on the flow regime, turbulence, fluid properties and the length of the conduit (31). The gas velocity, V_g , is usually higher than the liquid velocity, V_l , as the fluid progresses in the two-phase section. The composition of the fluid therefore changes as it flows through the tube. Although the assumption that phase equilibrium exists is not exact, it is the best one that can be made without very tedious calculations. A survey by DeGance does not favor the flow-regime based correlations because of the additional errors associated with the calculation of flow regimes (32).

- (3) The liquid/vapor ratio for mixture working fluids can be calculated for tube side flow, if the heat exchanger two-phase section is divided into large numbers of subsections with respect to pressure or temperature. The pressure or temperature interval midpoint may then be used to get the value of L/V by assuming phase equilibrium in each subsection. DeGance recommends Dukler's constant slip method for horizontal flow pressure drop calculations (33). Constant slip would mean that the slip velocity, V_s , is zero; or in other words, the velocities of gas and liquid phase are identical at each point in

the heat exchanger. The heat transfer coefficient correlation for two-phase flow used in the GEO4 simulator developed at the University of Oklahoma is probably adequate for the design purposes.

- (4) Tubes are arranged in triangular, square or rotated square pitch. Triangular tube layouts result in better shell side coefficients and provide more surface area in a given shell diameter (30). The GEO4 simulator uses triangular tube pitch and therefore no change is needed.
- (5) Higher circulation rates are possible on the tube side compared to the shell side, and the tube side flow distribution is more uniform. Mixture working fluid heat transfer coefficients are higher than pure fluids when the working fluid is on the shell side. Therefore, even higher heat transfer coefficients may be obtained for mixture working fluids flowing on the tube side because of higher circulation rates.
- (6) It has been shown that sand fluidized by the geothermal brine on the shell side prevents scaling and increases the brine side heat transfer coefficient over conventional shell and tube heat exchangers (34). Both horizontal and vertical-tube bundle arrangements are being

considered (35). In preliminary evaluations, the vertical tube model appears to have more uniform flow distribution and slightly higher heat transfer coefficients than horizontal models (at the same cross-sectional velocity); the converse is true when no bed is involved. However, the heat transfer coefficient of the horizontal-tube model can be brought closer to the vertical-tube model by some design improvements. Since fluidized bed heat exchangers could reduce the size of the heat exchanger by as much as 50%, the development of such heat exchangers would definitely enhance the chances of working fluids being used on the tube side. Under such circumstances, the use of fluidized-bed vertical-tube heat exchangers must be given serious consideration.

- (7) The use of direct contact heat exchangers for mixture cycles is attractive because a countercurrent flow situation can be achieved. In the Elgin type of brine heat exchanger, the working fluid is heated as it moves vertically upward in countercurrent flow with the brine. Because the height of the Elgin column is much greater in the liquid-liquid range than the boiling range, the fact that boiling is initiated

earlier by the mixture is a probable advantage (decreases size of heat exchanger).

Condenser

In earlier studies on the use of mixtures in geothermal binary cycles the heat transfer coefficients of the mixtures were calculated to be lower than the pure fluids in horizontal single pass shell and tube condensers. The working fluid mixtures were assumed to be condensing on the shell side with cooling water flowing counter-currently on the tube side. However, since the condensate drops from the tubes as it is formed in the horizontal condenser, a true counter-current flow behavior would not be achieved. In order to achieve true counter-current flow behavior, vertical vapor-in-tube down-draft condensers could be employed. Such condensers have several advantages over horizontal condensers, which will be discussed later in this section.

Many of the same advantages are offered by the vertical vapor-in-shell condensers that use baffles designed to permit condensate to remain on the tube (36). The condensing two-phase heat transfer coefficient can be increased by introducing turbulence, or in other words, by increasing Reynolds and Prandtl numbers. This may also be done by employing low-finned tubes or fluted tubes.

The mixture working fluid heat transfer coefficients can also be increased by employing vertical vapor-in-tube condensers. The vertical condenser could be of the conventional shell and tube exchanger type or the spool-wound exchanger shell type. A spool-wound exchanger consists of a layer of spirally wound tubes around a core with a fairly small pitch (37). The working fluid mixture would then condense downward within the tubes, while the cooling water would flow within the shell and between the tubes. This would require technology transfer from the LNG (liquefied natural gas) industry where spool-wound exchangers are used. Reference 37 indicates that mixtures may provide better heat transfer compared to pure fluids for identical pressure drop ratios of liquid and gas phases.

In spool-wound tube exchangers, the tube side working fluid mixture composition in liquid and gas phase at each point in the exchanger subsection can be assumed to be constant if the exchanger is subdivided into a large number of subsections. The vapor phase working fluid enters the exchanger at high velocity. As the mixture fluid condenses, the liquid phase (condensate) flows along with the uncondensed gas phase. Initially the gas and liquid phase velocities are different—the gas phase velocity being on the higher side. However, under the conditions of equilibrium (because of the increased turbulence and increased vapor liquid contact in spool-wound tubes) a constant slip may be assumed. The

Dukler correlation applies for vertical tubes, and can be modified for the spool-wound tubes (36,37).

In the conventional shell-and-tube vertical condenser, the condensate subcooling (for tube side cooling) is more efficiently accomplished due to falling-film heat transfer. Horizontal tube side cooling uses only a small portion of the available area. Appreciable horizontal shell side subcooling can be achieved only by flooding part of the shell.

The situation with respect to the direct contact condenser has not been considered in sufficient detail to make definite statements, but the fact that counter-current flow can be achieved offers an obvious advantage for mixtures compared to pure fluids in direct contact condensers.

CHAPTER VI

CONCLUSIONS AND RECOMMENDATIONS

Conclusions

Geothermal binary cycles using paraffin hydrocarbons as working fluids have been analyzed using principles of basic thermodynamics and in a relatively simple way. The complex interrelationships of thermodynamic, equipment and unit operational, and cost parameters have been developed and presented in a short form.

When the design objective is the maximization of the net plant work per unit mass of brine, mixtures provide more work than pure fluids. Although resource temperature decline will always result in a decrease in net plant power, mixture composition and behavior can be changed for better performance over the plant lifetime to match the changes in the geothermal resource.

At lower georesource temperatures, the cost optimum and resource utilization optimum nearly coincide, whereas at higher georesource temperatures they differ. The thermodynamic optimum always differs from the cost and resource utilization optima.

Recommendations

The methodology developed in this research cycle using paraffin hydrocarbons may be used as a first step for analyzing other types of energy conversion processes as well as geothermal binary cycles. A preliminary but detailed thermodynamic analysis should be performed for any energy conversion process before any computer simulation. This relatively inexpensive task of thermodynamic analysis can save a lot of headaches which may not be understood by computer simulation alone.

Provision must be made for the georesource temperature decline in the design of geothermal power plants.

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