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REVIEW AND TENTATIVE SELECTION OF A WORKING FLUID FOR USE WITH A MEDIUM TEMPERATURE (300°F) GEOTHERMAL RESOURCE

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ABSTRACT

A review of conventional refrigerants was made to determine the best fluids to be used as the working fluid in an electrical power plant application where energy will be derived from a medium temperature geothermal heat source. Major considerations were operating pressures, operational flexibility, thermal efficiency, line sizes and potential for best utilization of the geothermal heat source. As a result of this work, isobutane was selected as the best working fluid for a 300^{o}F geothermal fluid application. This fluid has been identified by other investigators for use with higher temperature application. Its' acceptability for the lower temperatures indicates the general utility of this fluid for power plant applications over a significant temperature range.

In an effort to obtain maximum utilization of the geothermal fluid, a dual boiling cycle was selected as the reference cycle for preliminary study and a basis for future comparative analysis.

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NOMENCLATURE

A - Area

Cp_b - Specific heat, geothermal fluid

Cp_f - Specific heat, working fluid

Cp_{cw} - Specific heat, cooling water

h - Headloss

h_e - Turbine exit enthalpy

h_{fcond} - Saturated liquid enthalpy, condenser

Δh_t - Turbine enthalpy drop

Δhp - Feed pump head rise

M - Molecular weight

P_t - Turbine inlet pressure

$Pcond$ - Condenser pressure

ΔP_f - Pressure losses

P_{net} - Net power after major system losses

q_{net} - Net power per pound of working fluid

q_{pw} - Feed pump work per pound of working fluid

q_{cw} - Work required for condenser cooling per pound of working fluid

q_t - Turbine work per pound of working fluid

q_r - Heat rejected to the condenser per pound of working fluid

Q_a - Total, heat added to the working fluid

V_e - Exit velocity turbine last stage

W_b - Weight flow rate, geothermal fluid

W_f - Weight flow rate, working fluid

Tcw_o - Condenser cooling water outlet temperature

Tcw_i - Condenser cooling water inlet temperature

NOMENCLATURE

T_{cond} - Condensing temperature of the working fluid

ΔT_b - Brine temperature drop

ΔT_{sc} - Condenser hot well subcooling

N_{tg} - Turbine generator efficiency

N_{eff} - Effectiveness parameter

ν - Specific volume

γ - Ratio of specific heats, $\gamma = {}^{Cp}/\, Cv$

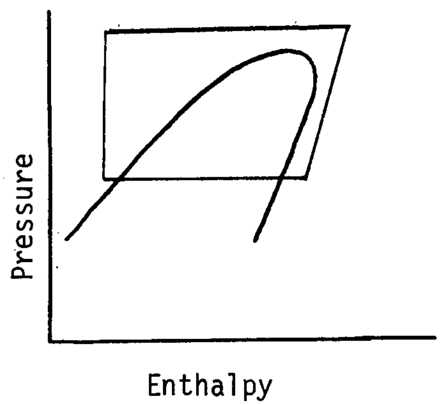
1.0 INTRODUCTION

Geothermal energy can be utilized for the production of electrical power by conventional means in either of two ways to drive a turbine-generator. The simplist in many respects and perhaps the most economical in certain situations is to flash the geothermal fluid and use the steam to drive a steam turbine. Flash systems may have one or more flashes, however, they currently have been limited to pressures above atmospheric pressure to prevent oxygen from entering the system. Thus, a brine temperature less than $\sim$205-212^{o}F is not possible. This limitation is particularly serious when considering use of a medium temperature geothermal fluid for the production of power.

Heat exchange between the geothermal brine and a secondary "working" fluid offers the potential to increase the brine utilization significantly. The secondary fluid is used in a conventional Rankine cycle. Selection of a working fluid is a major problem because of the many fluids available (mainly organic compounds) and various ways they may be used, that is, the use of a boiling or super-critical state or as mixtures. In addition to pure thermodynamic considerations are the more subjective considerations such as weighting turbine size, heat transfer area,flammability and toxicity, and the ability to obtain adequate thermodynamic properties.

A preliminary review of cycle performance of various fluids was previously reported in Reference 1 where boiling, superheating, regeneration and super-critical cycles were considered.

The purpose of this paper is to present a further review of the more common refrigerants (References 2 and 3), for which thermodynamic properties are readily available, with respect to their use in an electrical power plant and to present the basis for the preliminary selection of a cycle for use with a medium temperature geothermal resource. The particular emphasis of this report are factors which affect equipment size, cost and operational flexability.



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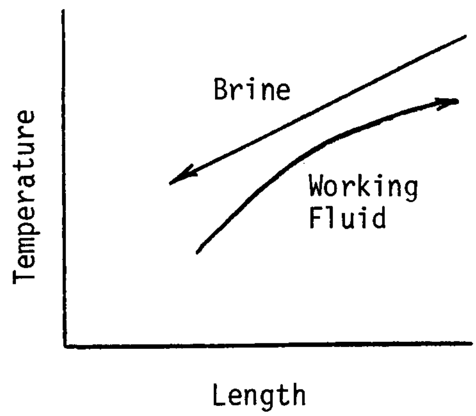

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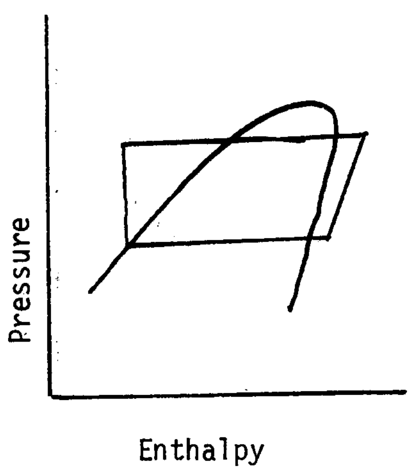

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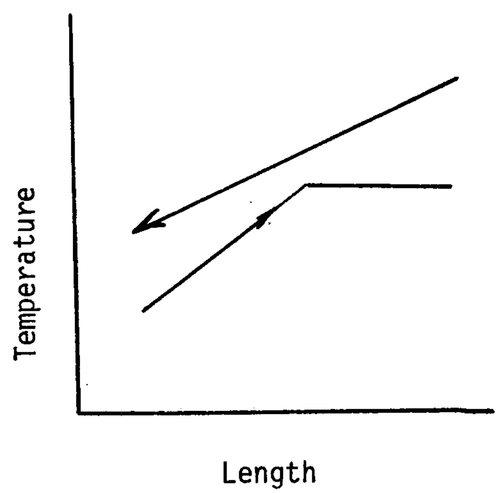


1(d)

Figure 1

2.0 A REVIEW OF CYCLE PERFORMANCE

A wide range of fluid state points are possible due to the many potential working fluids. Conventional boiling and superheating can be used to establish turbine inlet conditions, or supercritical conditions can be established. In addition, multiple stage boiling is possible. Figure 1(a) shows a typical super-critical cycle in a pressure - enthalpy diagram. Figure 1(b) shows the corresponding heat exchanger conditions. The bending of the working fluid curve in Figure 1(b) is due to the highly variable specific heat of the fluid just above the critical point. As a result of this variation, a pinch point occurs between the two end points. The extreme of this situation is the boiling cycle (Figure 1(c)) where the pinch point generally occurs at the point boiling begins (as shown in Figure 1(d)). The closer a super-critical cycle approaches the critical temperature more extreme is the variation shown in Figure 1(b) until there is little advantage to the super-critical cycle.

A key factor in the selection of a working fluid is the actual brine utilization for the production of power. The following is a simplified approach to establishing the brine utilization considering only the major parasitic losses within the power plant. Losses in the brine transmission systems, supply well and re-injection well pumping are not included since they are minimized when the brine flow rate is minimized.

Let

(1) $P_{net} = W_f \; q_{net}$, where

(2) $q_{net} = N_{tg} \; q_t - q_{pw} - q_{cw}$

The heat added to the working fluid is

(3) $Q_a = W_f \; (q_r + q_t - q_{pw})$

Since an equal amount of energy must be given up by the brine

(4) $Q_a = C_{pb} \; W_b \; \Delta T_b$

Equating (3) and (4) and substituting into equation (1)

(5) $$\frac{P_{net}}{W_b} = Cp_b \; \Delta T_b \quad \frac{q_{net}}{q_r + q_t - q_{pw}}$$

Substituting equation (2) into equation (5) for q_{net} and dividing by q_t yields

(6) $$\frac{P_{net}}{W_b} = C_{pb}\Delta T_b \left(\frac{N_{tg} - {}^{q_{pw}}/_{q_t} - {}^{q_{cw}}/_{q_t}}{1 + {}^{q_r}/_{q_t} - \frac{q_{pw}}{q_t}} \right)$$

Let

(7) $$N_{eff} = \frac{N_{tg} - {}^{q_{pw}}/_{q_t} - {}^{q_{cw}}/_{q_t}}{1 + {}^{q_r}/_{q_t} - {}^{q_{pw}}/_{q_t}}$$

Then

(8) $$Q_{net}/W_b = C_{pb}\Delta T_b \, N_{eff}$$

The bracketed term of equation (6) has been defined as an effectiveness N_{eff} and provides a means of measuring the potential of one fluid against another for a given set of conditions. N_{eff} is merely a term in which the system losses (pump work, cooling work and heat rejection) are compared with the ideal work in the turbine.

It is obvious from equation (8) that utilization is also a function of the brine temperature drop and in reality it is the product $\Delta T \; N_{eff}$ that must be maximized to provide the highest utilization. The ΔT_b is sometimes limited by pinch point problems on the heat exchangers. However, for the purpose of comparing various fluids to identify those which offer the greatest potential N_{eff} can be used as a guide.

To provide estimates of the pump work, the following equation was used.

(9) $$q_{pw} = \frac{0.185 \, \nu_f}{\eta_p} \left(P_t - P_{cond} + \Delta P_f\right)$$

Where ΔP_f was assumed constant at 40 psi and P_t and P_{cond} varied according saturation pressure for the turbine inlet and condensing temperatures.

The power expended for condenser cooling was approximated for a wet cooling tower. This reduces to

$$q_{cw} = \frac{0.263\,(h_e - h_{fcond} + C_{pf}\,\Delta T_{sc})}{C_{Pcw}\,(T_{cwo} - T_{cww})}$$

Hotwell subcooling of 5^oF and a cooling water temperature rise of 25^oF was assumed for all calculations.

Values of N_{eff} have been computed for several turbine inlet temperatures and fluids which survive the initial screening. The results will be discussed in a later section.

3.0 PRELIMINARY COMPARISON OF FLUID PERFORMANCE

A comprehensive evaluation of working fluids and associated cycles would require cycle optimization for each fluid and a comparative cost estimate of each optimized cycle. In lieu of such an extensive effort, certain properties and parameters were compared that would permit a quantative screening of fluids.

3.1 Basis for Comparison

The fluids were compared to identify those most suitable for use with a 300^{0}F heat source. Table I lists the fluids and the more important parameters. The table does not include those refrigerants which obviously, because of low critical temperature would not be suitable in a Rankine cycle. For the screening process the cycle was assumed to operate between a maximum temperature of 280^{0}F and a condenser temperature of 110^{0}F. Each of the parameters listed are briefly discussed below.

3.1.1 Critical Temperature and Pressure

The critical temperature determines whether or not a cycle may be operated in the super-critical region or if the cycle is restricted to boiling. Generally, fluids with a low critical temperature (as is necessary for super-critical operation with a 300^{0}F heat source) require a relatively high operating pressure at the desired operating temperature. An important consideration for super-critical operation is, for a given turbine inlet temperature, whether there is an adequate margin to the critical temperature. A small margin would result in a system very sensitive to a change in source temperature and would result in heat exchanger limitations approaching those of a conventional boiling system.

A fluid with a relatively low critical temperature used in a boiling cycle would tend to have a higher boiler pressure than another fluid with a higher critical temperature. The condensing pressure will also tend to be higher. Thus, in a general way, the critical temperature implies a factor on heat exchanger cost because of pressure.

3.1.2 Condenser Pressure

Condenser pressures were tabulated at 110°F. A preferred fluid should have a condenser pressure above atmospheric pressure to ensure that air does not leak into the system and to permit non-condensables that may result from working fluid breakdown or are residual from system fill to be vented from the system. However, the pressure should not be extremely high since this would increase the cost of the condenser. The cost increases about linearly with pressure above 150 psi at a rate of about 10% per 100 psi up to about 600 psi.

3.1.3 Turbine Inlet Pressure

The turbine inlet pressure is determined by the maximum fluid temperature. For the purpose of this initial screening, a maximum fluid temperature of 280°F was used to establish this pressure. This temperature is sufficiently high to permit some fluids to operate in the super-critical state, however, it may somewhat distort results for fluids in boiling cycles, particularly if operated at substantially lower temperatures. For example, the turbine parameter (see below) would be distorted, however, the distortion would be favorable (due to the higher enthalpy drop associated with a higher temperature). Therefore, a fluid that does not appear satisfactory under these conditions would be less satisfactory under more critical evaluation.

As with the condenser, high pressure causes an increase in cost of the heat exchanger equipment. The range of pressure variations is quite large leading to significant feed pump work for certain fluids.

3.1.4 Turbine Enthalpy Drop and Specific Volume

The turbine enthalpy drop is indicative of the flow requirements of the working fluid. Other factors being equal, a large enthalpy drop is desirable since it reduces the amount of working fluid mass flow.

TABLE I
COMPARISON OF COMMON REFRIGERANTS FOR MEDIUM TEMPERATURE POWER CYCLE APPLICATIONS

Refrig. No.	Mol. Wt.	Crit. Temp. ^oF	Crit.Press. Psia	Press. at 280^oF,psia	Cond.Press. at 110^oF	Δh Turbine Btu/#	$\Delta h_t/h_{fg}$	Pumpwork Relative $\frac{\Delta h_p}{\Delta h_t}$	Relative Piping Size for Same Power Loss: Liquid 10^{+4}	Vapor Turbine Outlet $X10^{+3}$	Vapor Turbine Inlet $X10^{+3}$	Relative Turbine Size: $\frac{v_g}{\Delta h_t}\sqrt{\frac{M}{\delta}}$	Fraction of R-113	Rating	Comments
11	137	388	640	247	27.8	17	0.23	0.026	1.6	21.4	2.4	0.97	0.56	D	Turbine,Turbine exhaust large
12	121	233	597	700	151	12	0.22	0.105	3.1	6.5	1.1	0.23	0.13	B	Generally good,pumpwork slightly high
21	103	353	730	400	47.4	14	0.15	0.56	2.3	22	2.5	0.78	0.43	D	Turbine & exhaust line large efficiency low
22	86	205	722	1000	241	15	0.21	0.13	2.5	3.8	3.5	0.13	0.07	C	Generally good,pumpwork high,operating pressures high
31/114	94	287	749	695	95	19	0.21	0.078	1.6	7.2	1.2	0.28	0.15	A	
32	52	176	454	1400	400	19	0.19	0.13	1.6	1.5	-	0.05	0.03	C	Pumpwork high,operating pressures high
40	50	289	969	580	135	28	0.21	0.054	1.9	2.9	2.6	0.19	0.10	A	Good,toxicity may be problem
113	187	417	499	146	12.76	17.6	0.28	0.015	1.4	33	3.4	1.8	1	D	Turbine & line sizes large
114	171	294	473	400	53.8	12	0.23	0.061	2.8	14	1.3	0.62	0.34	C	Turbine & line sizes somewhat large
115	155	176	451	2000+	209	8	0.24	0.56	5.8	0.88	0.88	0.19	0.11	D	Pumpwork & pressures excessive
142b	100	279	598	500	68.7	16	0.20	0.08	2.4	5.7	1.3	0.06	0.03	C	Efficiency low,otherwise fair
152a	66	236	652	700	144	22	0.18	0.0882	1.8	5.1	0.97	0.15	0.085	B	Efficiency low,otherwise good
216	221	356	400	192	19	13	0.26	0.039	2.3	2.9	2.8	0.5	0.83	D	Large Turbine
290	44	206	617	1020	214	24	0.19	0.214	2.9	4.2	0.68	0.13	0.07	C	Efficiency low,operating pressures & pumpwork high
318	200	239	401	520	79	9	0.22	0.102	4.2	12.1	0.74	1.5	0.81	D	Turbine,turbine exhaust line large
500	99	222	642	1000	179	13	0.19	0.172	3.1	5.7	1.2	0.20	0.11	D	High boiler pressure,high pumpwork
502	112	194	619	∿1000+	260	12	0.25	0.158	3.4	3.7	--	0.12	0.71	D	Turbine large,pumpwork
504	79	151	696	∿2000	427	16	0.28	0.32	2.7	1.9	--	0.06	0.034	D	Pumpwork excessive
600	58	305	550	437	60.3	34	0.23	0.061	1.5	7.8	0.86	0.33	0.18	A	Generally good
600a	58	274	529	520	83.2	30	0.23	0.082	1.8	6.6	0.76	0.26	0.14	A	Generally good
717	17	271	1657	1000+	247	65	0.16	0.060	0.53	2.3	--	0.07	0.037	C	Low efficiency,high pressure
1270	42	197	670	1200	252	24	0.19	0.24	0.88	3.5	0.81	0.07	0.037	C	Excessive pumpwork

The extensive heat transfer surface required by the power plant requires a substantial amount of piping to connect multiple heat exchangers. This piping represents a substantial cost, which is, in part, a function of its size. If it is assumed that independent of the working fluid any plant will have about the same fraction of power losses permitted in plant piping then, a parameter can be deduced to compare the relative size of piping for different fluids.

$$\text{Power Loss} = W_f h = \frac{P}{\Delta h_t} h$$

From continuity and the fact that the flowrate is inversly proportional to the enthalpy drop

$$\text{Power Loss} = K \frac{P^3}{A^2} \frac{\nu^2}{\Delta h_t^3}$$

Thus for the same allowable power loss between plants with various working fluids

$$A_p = K \frac{\nu}{\Delta h_t^{3/2}}$$

Values of the parameter $\nu/\Delta h_t^{3/2}$ are given in Table I for a liquid and vapor at condenser conditions and a vapor at turbine inlet conditions.

3.1.5 Relative Thermal Efficiency

An indication of the relative thermal efficiency can be achieved by comparing the ratio of turbine enthalpy drop to the latent heat at the condensing pressure. This ratio is listed in Table I and is significant since it implies the relative heat exchanger size. The higher this ratio, the smaller will be the heat transfer area, other factors being equal. The reciprocal implies the relative power loss for cooling the condenser since it is proportional to the heat rejected. Use of the latent heat assumes that the amount of superheat in the turbine exhaust is small and this is adequate for a first order approximation.

3.1.6 Relative Pump Work

A significant fraction of cycle work can be expended as feed pump work. A relative value can be obtained by forming a ratio of the pump head rise in feet to the turbine enthalpy drop. That is, since

Pumping Power = $W_f \Delta hp$

and $W_f = \dfrac{\text{Power}}{\Delta h_t}$

then pumping power $\propto \dfrac{\Delta hp}{778 \Delta h_t}$

and the net power after pumping = $1 - \dfrac{\Delta hp}{778\ \Delta h_t}$

3.1.7 Relative Turbine Size

Turbine size is a significant factor in selecting a working fluid not only from a cost viewpoint, but because a significant range of plant sizes must be accommodated. For example, a fluid requiring a 10 ft diameter turbine rotor for a 10 MW plant would not be suitable for a 50 MW plant without using five separate turbines.

A simple means of relating relative turbine size between fluids was established based on turbine exit conditions.

The exit area

$$A_e = \frac{Q}{V_e} = \frac{W_f \nu_e}{V_e}$$

Since W_f = Power/Δh_t

and the exit velocity is assumed to be a fraction of the sonic velocity at exit conditions.

$$V_e = (\text{Const})\sqrt{\gamma g \frac{1544}{M} \text{ Tcond}}$$

Since the comparison is based on similar exit temperature conditions and the gross cycle power is relatively constant. A first order approximation of relative turbine size is obtained by the parameter.

$$A_e \propto \frac{\nu g}{\Delta h_t} \sqrt{\frac{M}{\gamma}}$$

This parameter is largest for R-113. Crude estimates place the maximum rotor diameter at about 8 ft for a 10 MW single flow machine. Using the R-113 value as a base, a relative value to R-113 has also been included in Table I.

3.2 Comparative Ratings

Table I is the basis for the initial screening of the various fluids which were rated on a scale of A through D. Fluids with a pressure in excess of 1000 psi at 280^{o}F, condenser pressures above 300 psi and turbine sizes that were not at least one third the size of a R-113 turbine, or with pump work which exceeded about 10% of the ideal turbine work were rated low. In general, fluids requiring large turbines also required relatively large turbine exhaust piping although some exceptions exist, for example R-142 and R-216. Pumping power varied considerably, however generally, fluids with the higher operating pressures also had the higher pump head requirements.

In summary, the fluids shown in Table II appear to be superior. Flammability limits, Underwriter Laboratories toxicity group rating and upper Threshold Limit Values [4] are given for these fluids. The higher the Underwriters Laboratory group number the less toxic the fluid. The Threshold Limit Values are upper values and normal design requirements would be at least one tenth of these values. Obviously it is desirable to select the least toxic fluid unless improved performance obviously would pay for the required safety systems and increased maintenance costs. This rationalization can also be applied to flammable vs non-flammable fluids.

Calculations of effectiveness (equation 7) were made for various operating temperatures to compare the performance potential of each of the fluids tested in Table 3. A tabulation of the results is given in Table 3 in order of increasing effectiveness at 280^{o}F. From this tabulation, isobutane (R-600a) appears to be the best working fluid; followed closely by normal butane (R-600). It is evident that both isobutane and normal butane have about equal performance followed closely by R31/114 (about 5% below) and R-152(a). R-40 and R-12 run about 25% lower than the butanes.

Isobutane appears to be the best overall fluid. Although it is flammable, it's limits are narrower than all the other fluids except normal butane. A disadvantage is that it also has the lowest limit. It is low in toxicity, ranking next to R-12 and R-152(a). R 31/114 although not flammable, is closer to methyl chloride in toxicity.

Based on this evaluation, it is concluded that isobutane has the best overall qualities. Furthermore, detailed cycle evaluations may prove the other fluids as good, however, it is doubtful if they will be substantially better. Therefore initial studies of the organic cycle power plant operating with a moderate temperature geothermal source will utilize isobutane as the reference fluid. This fluid also offers the advantage of being used in mixtures with other hydrocarbons which can potentially increase the overall cycle efficiency and/or reduce cooling water requirements.

TABLE II

FLUIDS SELECTED ON INITIAL SCREENING

Fluid No.	Fluid Name	Relative Rating	Flammable % by Volume	Toxicity Group No. (1)	Health Limit (2) PPM (Volume/Volume)
R-12	Dichlorodifluoromethane	B	None	6	1000
R-31/114	Azeotrope Chlorofluoromethane and Dichlorotetrafluoroethane	A	None	∿4-5	-
R-40	Methyl Chloride	A	8.1-17.2	4	100
R-152(a)	Difluoroethane	B	5.1-17	6	-
R-600	Normal Butane	A	1.65-6.5	5	600
R-600(a)	Isobutane	A	1.8-8.4	5b	600-

(1) Underwriters Laboratory Group Rating

(2) From reference 5. Isobutane value estimated. Values of R-31/114 and R-40 not available.

TABLE III

POTENTIAL RELATIVE EFFECTIVENESS OF SELECTED FLUIDS

Fluid	Temperature at Turbine Inlet, °F 280	260	240	220	200	180
R-12	.098	.093	.083	.073	.067	.063
R-40	.10	.09	.078	.062	.057	.045
R-152(a)	.113	.105	.093	.084	.075	.061
R-31/114	.114	.107	.096	.085	.06	.057
R-600	.12	.11	.10	.088	.077	.065
R-600(a)	.13	.011	.10	.092	.079	.067

4.0 TENTATIVE FLUID CYCLE SELECTION

The better fluids all imply a boiling cycle since their critical temperatures are relatively close to that of the heat source ($300^{0}F$). Any system selected should be capable of operation at a degraded temperature condition and super-critical cycles are not particularly good under these conditions.

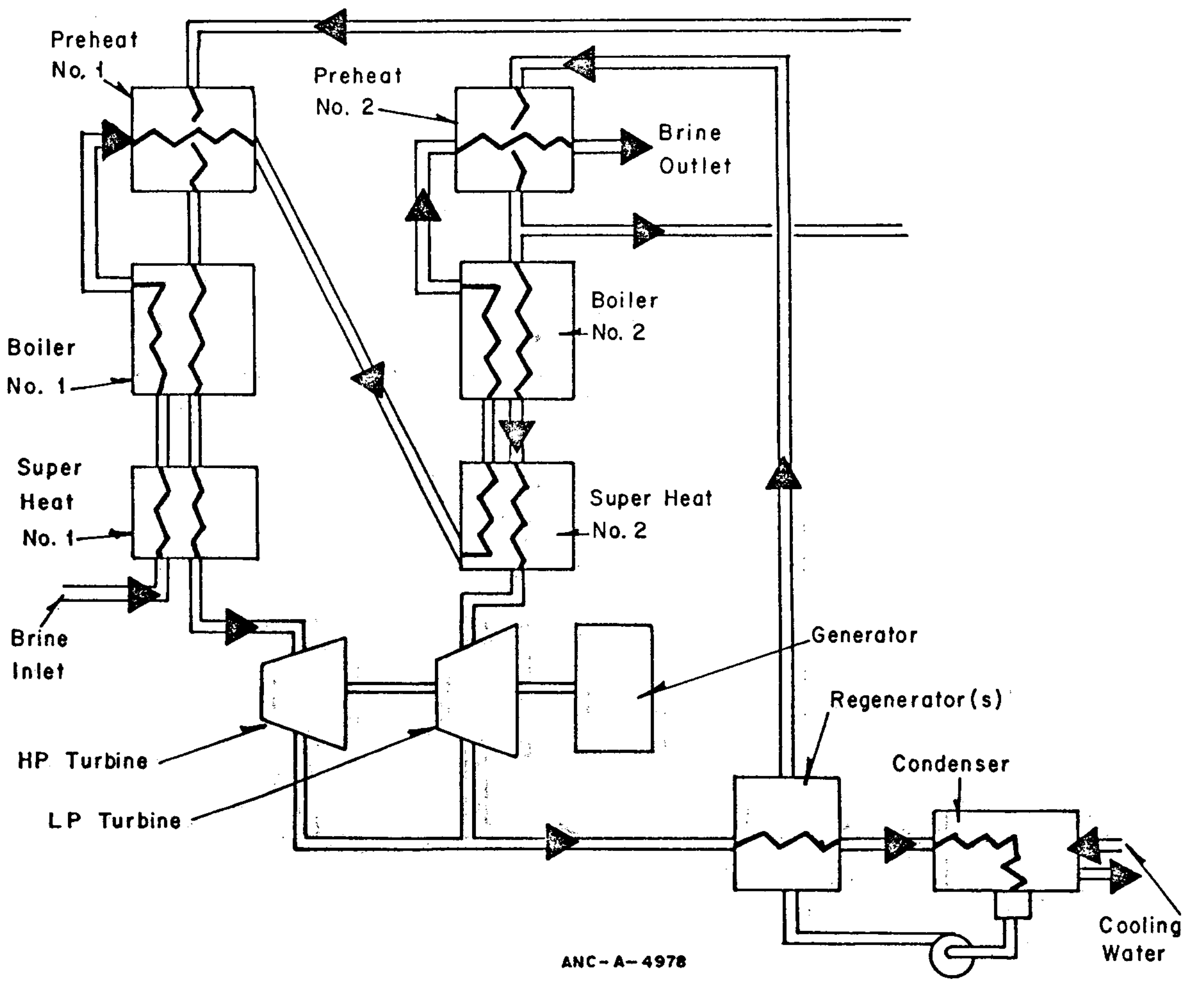
A single boiling isobutane cycle, without super-heating, will achieve its maximum utilization at a turbine inlet temperature of about $220^{0}F$. At this condition, the utilization is about 3.9 MW per million pounds/hour of geothermal fluid (not including brine pumping losses) and the exit brine temperature is about $150^{0}F$.

A dual boiling system as shown in Figure 2 offers the potential of increasing the brine utilization over that of a single boiling cycle. Preliminary calculations indicate that with turbine inlet conditions of $250^{0}F$ and $190^{0}F$ with about 60% of the total working fluid going to the high pressure boiler that a utilization of about 4.5 MW per million pounds/hour of geothermal fluid is possible with a relatively small increase in the required heat transfer area.

The dual boiling cycle shown on Figure 2 includes a regenerator in the turbine exhaust and superheaters,however they would only be added if detailed analysis indicated that they are warranted. The additional complexity introduced by a dual boiling system is not as significant as it may appear. Estimates of surface area requirements for a single boiler system indicate that total heat transfer areas of about 2.5×10^{5} ft^{2} are required for a 10 MW power plant. Since such large heat transfer surfaces are required multiple units will be required, thus, dual boiling may not add a significant complication. It is estimated that the additional heat transfer area required is about 20% greater than that of a single boiling cycle.

It is expected that the increase in performance will pay for the increased cost of a dual boiling system, however, more detailed studies will be required to determine actual benefits.

Based upon the considerations discussed in this report, a dual boiling cycle utilizing isobutane as a working fluid will be the reference system for further study for comparative performance analysis with other possible systems and for preparation of a detailed plant cost estimate.



Preheat No. 1
Preheat No. 2
Brine Outlet
Boiler No. 1
Boiler No. 2
Super Heat No. 1
Super Heat No. 2
Brine Inlet
Generator
Regenerator(s)
Condenser
HP Turbine
LP Turbine
Cooling Water
ANC-A-4978

Figure 2

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