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SUBJECT: Choice of Working Fluid and Operating Conditions for Energy Conversion with Geothermal Heat Sources*

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ABSTRACT

A study of electric power generating cycles which can be used in conjunction with a low-temperature geothermal energy (sensible heat) source is presented. Investigators at Los Alamos Scientific Laboratory have proposed a steam/isobutane dual cycle capable of generating 50.7 Mw of electric power. This cycle is optimized using regenerative feed water heating to yield a power output of 52.1 Mw or an improvement of 2.8%. Other power cycle designs and working fluids are investigated. Working fluids including various Freons, steam, isobutane, ammonia, SO₂, CO₂, and several low molecular weight alkanes and alkenes are considered. The power cycle designs which are investigated include single and dual Rankine cycles (some of which involve regenerative feed heating), as well as topping/bottoming Rankine cycles, and Brayton gas cycles.

The investigation shows that use of a single Freon-11 cycle results in a 57.7 Mw power output representing an improvement of 13.8% over the LASL proposal. This cycle represents the best performance of all the combinations investigated, although several other cycles exhibit performance superior to that of the LASL proposal. It is recommended that the Freon-11 single cycle be considered on an economic basis as a replacement for the steam/isobutane cycle in the proposed LASL system. Furthermore, it is recommended that several new Freon working fluids be investigated for which thermodynamic data were not available at this time.

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1. SUMMARY

This report presents the results of an investigation of a low temperature electric power generating cycle to be used in conjunction with a low temperature geothermal energy source.

Recent interest in unexploited energy sources has shown that a significant amount of energy may be extracted from hot rock deposits beneath the earth's surface, in some cases at depths of only 5000 ft. Los Alamos Scientific Laboratory (LASL) has proposed a system which would convert this geothermal energy into electric power by extracting the heat with pressurized water and using this water as the energy input to a power generating cycle. In LASL's proposal the power cycle is a saturated steam/supercritical isobutane dual cycle capable of generating 50.7 Mw of electric power.

The purpose of this investigation was to try to optimize the proposed LASL cycle with respect to energy output and efficiency and to investigate the possibility of using other working fluids and other power cycle designs. All proposals considered were based on a geothermal water source of 683 lb/sec entering at 2000 psia and 536°F. Specifically the working fluids which have been considered include various Freons, including Freon-11, 21, 115, and 502, isobutane, steam, ammonia, CO₂, SO₂, and several low molecular weight alkanes and alkenes including ethane and ethylene. The power cycles which have been investigated include single and dual Rankine cycles, topping/bottoming Rankine cycles, and Brayton gas cycles. In addition, regenerative feed heating has been applied to some cycles.

It has been found that the proposed LASL cycle could be improved about 2.8% to yield a net output of 52.1 Mw by use of regenerative feed water heating. The saturated steam cycle is inherently inefficient due to the phase change encountered in boiling water to saturated steam, and so it has been found that the cycle cannot be significantly improved.

Use of other working fluids and different cycle schemes have resulted in significantly improved performance. The best case, a Freon-11 single cycle, resulted in 57.7 Mw power output or a 13.8% increase over the proposed LASL steam/isobutane cycle. The isobutane topping/bottoming cycle, the SO₂ cycle, and the Freon-11/isobutane dual cycle were also found to have power outputs higher than the LASL proposed.

Criteria were developed for evaluation of working fluids prior to their use in power cycle calculations. These criteria include low specific volume of the working fluid in the compressed liquid region and a high specific volume in the superheated and supercritical regions. Also, extensive use was made of "temperature-enthalpy transferred," or T-Q, diagrams in evaluating the various cycle/fluid combinations.

It is recommended that the proposed LASL steam/isobutane dual cycle and a Freon-11 single cycle be compared on an economic basis. It is further recommended that the Freon-11 single cycle be more thoroughly optimized and that additional working fluids, for which thermodynamic data were not available at the time of this study, be considered.

2. INTRODUCTION

2.1 Geothermal Energy

Several forms of geothermal energy including subterranean hot water, hot rock, and steam are currently under investigation as possible sources of heat for power generating plants. We are concerned with a hot rock geothermal source present in southwestern USA.

2.2 Hot Rock Deposits

In general this type of geothermal energy can be found at depths of about 16,000 ft from the earth's surface and at temperatures above 553°F (290°C). A western area of about 95,000 square miles (formed by 13 states) in the vicinity of New Mexico comprises the more active geothermal centers. The Los Alamos Scientific Laboratory has proposed the construction of a pressurized water loop to extract the thermal energy for production of electricity (1). Drilling will be used to reach the hot rock region and hydrofracturing techniques will be employed to increase the area for heat transfer at the hot rock depth, thus creating a heat reservoir (see Fig. 1). Pressurized water is to be pumped into this heat reservoir through an inlet drill-hole and then extracted from an outlet drill-hole at 536°F. The type of fracturing proposed produces a vertically-oriented heat reservoir which will increase in depth due to thermally induced stresses on the rocks resulting from the heat removal process. LASL is conducting studies in the region of the Rio Grande Valley (New Mexico) to investigate the feasibility and economics of the hydrofracturing method. The hot rock deposits lie 5000 ft below the surface at this particular location and it has been proposed that a small power plant be constructed.

2.3 Proposed LASL System

The system proposed by LASL for extracting geothermal energy involves pumping water at 2000 psia into the hot rock layer. The compressed water is heated and returns to the surface (536°F) where it exchanges heat with a power generating cycle. The cooled water (131°F) is then returned to the ground thus completing the ground water cycle.

The power generating cycle with which this project is concerned is shown schematically in Fig. 2. The most important characteristic of the LASL proposed cycle is that it involves the use of two independent cycles each extracting part of the energy from the ground water stream. The first cycle uses saturated steam at 220 psia as the working fluid and absorbs approximately half of the ground water heat. Due to the temperature profiles within the exchanger which result from transferring sensible heat to a latent heat sink, this half of the cycle is relatively inefficient

compared to an ideal Carnot system exchanging heat with the same geothermal stream (see Table 1).

The second cycle uses supercritical isobutane at 588 psia. In contrast to the steam cycle this cycle has a high relative efficiency (Table 1).

Table 1. Output and Efficiency of LASL Steam/Isobutane Proposal

Cycle	Heat Input (Mw)	Heat Rejected (Mw)	Work Output (Mw)	Actual Efficiency (%)	Carnot Efficiency (%)	Relative Efficiency (%)
Steam	153.0	118.6	33.0	21.6	37.8	57.1
Isobutane	153.8	132.1	17.7	11.5	16.0	71.9
Overall	306.8	250.7	50.7	16.5	26.7	61.8

Rankine power cycles involve four characteristic steps. As shown in Fig. 3 for the ideal saturated Rankine cycle the cycle begins at (1) with saturated liquid. The liquid is compressed isentropically to (2). The compressed liquid then passes through the primary heat exchanger where it boils at constant pressure to (3). The third step is an isentropic turbine expansion to (4) which results in the production of work. The final step is isobaric condensation of the gas to (1), thus completing the cycle. This cycle is shown on temperature-entropy and pressure-enthalpy diagrams (Fig. 3), both of which appear throughout the report. Figure 4 shows the same process occurring in an ideal Rankine cycle operating in the supercritical region. The working fluid is pumped isentropically from (1) to (2), heated isobarically to (3), expanded isentropically to (4), and condensed isobarically to (1).

The above description of ideal Rankine cycles has assumed that all steps are reversible, and in the case of compression or expansion, are adiabatic. This is clearly not true in actual operation, and so it is necessary to determine how non-idealities affect the Rankine cycle. The main non-idealities include:

1. Irreversibilities in both the pump (compression) and turbine (expansion) steps which result in an increase in entropy.
2. Condensation in the turbine for cycles which expand to subsaturation pressures (such as the LASL proposed steam cycle) results in decreased turbine efficiency and increased entropy.
3. Nonadiabatic pump and turbine operation result in decreased entropy.
4. Flow through the condenser and boiler results in pressure drops in these units.

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IDEAL SATURATED RANKINE CYCLE

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FIG.
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IDEAL SUPERCRITICAL RANKINE CYCLE

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The nature of these non-idealities may be seen in Fig. 5; the net effect of the non-idealities is a decrease in power obtained from the cycle. As will be discussed in Sect. 3.4, only non-idealities (1) and (2) have been taken into account as part of the calculations since the pressure drop and non-adiabatic corrections were considered relatively insignificant.

Performance figures for the proposed cycle are included in Table 1. The net power output of the cycle is 50.7 Mw at an efficiency of 16.5%. To achieve a power output of 50.7 Mw, the figures presented by Brown (1) were modified. Recalculation of the isobutane turbine output coupled with a better estimate of the steam turbine efficiency (Sect. 4.2.1) resulted in a much lower power output than expected (~45 Mw). As a result the ground water flow rate was increased to 683 lb/sec (306.8 Mw).

3. PROCEDURE

The two options for improving electrical power output from this geothermal source are modifications of the LASL proposal or investigation of different power cycle systems.

3.1 Regenerative Feed Water Heating

Initial efforts were directed toward improving the power output and efficiency of the proposed LASL scheme. The most obvious inefficiency of this LASL scheme is the large temperature difference (223°F) at the ground water exit of the steam cycle heat exchanger. Heat from the geothermal stream is being accepted by the water initially at a very low temperature. An increase in the feed water temperature above 120°F would result in a higher cycle efficiency. Feed water heating is widely used in existing power plants. The concept is illustrated in Fig. 6. Partially expanded steam at a given pressure is bled from the turbine and used to heat the heat exchanger feed water to an appropriate temperature. Heating of the feedwater stream in the LASL proposed cycle resulted in a higher power output. Calculations were performed using ORCENT, a computer program used at ORNL for designing power plants. A discussion of results is presented in Sect. 4.2.2, and a detailed explanation of the program's operation may be found in Appendix 7.3.

3.2 Alternate Power Cycle Schemes

As a means of improving the total power output obtainable from this geothermal source, other cycle schemes were considered and are summarized below.

Only supercritical Rankine cycles were considered since they allow for maximum heat transfer at the highest working fluid temperatures. No large

NOTE: Numbers in parentheses refer to inefficiencies and non-idealities described in Sect. 2.3.

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NON-IDEALITIES IN RANKINE CYCLES

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T_R = temperature of turbine bleed stream

T_F = feed water temperature = $T_R - 5^\circ\text{F}$

T_O = temperature of turbine bleed stream at feed
heater exit = $T_C + 10^\circ\text{F}$

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REGENERATIVE FEED WATER
HEATING FOR LASL STEAM CYCLE

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FIG.
6

temperature gradients due to working fluid vaporization are encountered in the heat exchangers and conditions are easily chosen such that turbines are free of moisture at all points in the expansion step.

(1) Binary Cycle. Replacing either one or both of the two fluids in the proposed cycle with different working fluids was investigated. For the top part of a binary cycle, Freon-11 (trichlorofluoromethane) and Freon-21 (dichlorofluoromethane) were studied, while for the bottom part Freon-115 (chloropentafluoroethane), Freon-502 [an azeotrope of Freon-22 (chlorodifluoromethane) and Freon-115], and isobutane were considered.

(2) Single Cycle. Using the geothermal energy completely in a single cycle was investigated for several working fluids (Freon-11, ammonia, and SO_2). Single cycles are inherently simpler to operate than binary cycles and equipment costs could be lower. A single supercritical Rankine Cycle is illustrated in Fig. 4.

(3) Topping-Bottoming Cycle. A topping-bottoming cycle uses one working fluid (the topping cycle) to accept all of the heat from the geothermal source as it is partially expanded to a superheated state. The working fluid is then cooled to a liquid by exchanging heat with a second working fluid which then undergoes an expansion step (Fig. 7). Isobutane was the only working fluid considered for use as a topping cycle fluid while isobutane was also used as the bottoming cycle fluid.

Working fluids which are gaseous throughout the temperature-pressure range of operation can be supplied with heat from a geothermal energy source and operated as Brayton cycles (see Fig. 8). With the working fluids considered (CO_2 , ethylene, and ethane), Brayton cycle operation was possible only as the top part of a binary cycle (geothermal stream from 536 to 343°F).

Table 2 presents a list of the important equipment operating parameters assumed as well as a summary of the geothermal source characteristics. Efficiencies for pumps and turbines are included to account for irreversibilities in their operation. Cooling and/or condensation is to be accomplished by dry cooling towers with an effective cooling capability to 120°F.

Table 2. Equipment Operating Parameters and Geothermal Source Characteristics

Turbine efficiency	= 85%
Pump efficiency	= 75%
Minimum difference in heat exchanger	= 10°F
Temperature of condenser/cooler	= 120°F
<u>Geothermal Stream</u>	
Flow rate	= 683 lb/sec
Power (thermal)	= 307 Mw
Inlet temperature	= 536°F, outlet temperature $\geq$ 131°F
Pressure	= 2000 psia
Maximum Carnot efficiency obtainable	= 26.7% (see Appendix 7.2)

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TOPPING-BOTTOMING RANKINE CYCLE

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IDEAL BRAYTON CYCLE

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FIG.
8

3.3 Working Fluid Selection

The first step in any cycle consideration is the choice of a working fluid. Certain fluid properties are especially desirable for high power output and maximum efficiency (see Table 3).

Table 3. Desirable Properties for Working Fluids

-
1. High C_p
 2. Low v at pump inlet
 3. $\left(\frac{\partial C_p}{\partial T}\right)_{\text{working fluid}} \approx \left(\frac{\partial C_p}{\partial T}\right)_{\text{geothermal stream}}$
 4. $P_C < 3000$ psia
 5. Non-corrosive

Additionally, for Rankine supercritical cycles,

6. $120^\circ\text{F} < T_C < 525^\circ\text{F}$
7. Low ΔH_{vap} at $T_{\text{condenser}}$ relative to ΔH in evaporator

Additionally, for Brayton gas cycles,

- 6'. $T_C < 120^\circ\text{F}$
 - 7'. Low P_C
-

Both high heat capacity (C_p) and small specific volume at the pump inlet are desirable to minimize the pump work required. A heat capacity which remains approximately constant over the temperature range of interest (similar to the C_p of the geothermal stream) will minimize temperature differences in the heat exchanger and thus permit maximum transfer of heat at the higher temperatures. A pressure limit of 3000 psia was used as a property of construction materials (ordinary low carbon steel was assumed for equipment construction). In addition, the fluids were required to be non-corrosive and thermally stable.

For supercritical Rankine cycle operation the working fluid must by definition have a critical temperature between the condenser temperature of 120°F and the maximum operating temperature of about 526°F. A small enthalpy of condensation results in minimum heat throw-away in the condensation stage. For Brayton cycles in which a condensed phase is not involved, a low critical temperature and pressure are desirable to avoid liquid formation at any point in the cycle.

3.4 Procedure for Cycle Evaluation

For all cycles in which the working fluid exchanges heat with the geothermal stream, i.e., all cycles considered except bottoming cycles, a very simple design procedure was employed. First a working fluid was selected based on the properties listed in Table 3. Cycle design proceeded as follows:

1. The operating pressure was chosen (greater than the critical pressure but less than 3000 psia). In general, it has been found that pressures just above the critical give optimum cycle performance (see Sect. 4.1 for discussion).

2. Starting from a temperature of 120°F (working fluid inlet heat exchanger temperature) a plot of h versus T [line (a) in Fig. 9] for a constant pressure heating process was constructed. Thermodynamic properties were obtained from plots or tables (P-H, T-S) (4, 5, 6, 7, 8).

3. A straight line is drawn above the first one starting at 536°F with the restriction that it must at all points lie at least 10°F above the first line [line (b) in Fig. 9]. The heat exchanger temperature profile has been constructed by assuming constant heat capacity for the geothermal stream.

4. Since the flow rate and enthalpy change of the geothermal stream are known, the total heat transferred to the working fluid can be computed by:

$$Q_s = \dot{m}_s (h_{536s} - h_{T_{out}s}) \quad (1)$$

The enthalpy between 120°F and the exit temperature of the working fluid is known and therefore its flow rate may be calculated.

$$\dot{m}_{wf} = \frac{-Q_s}{(h_{120wf} - h_{T_{out}wf})} \quad (2)$$

The straight line (b) (Fig. 9) can be constructed in a number of ways so that the optimal flow rate, outlet temperature, and pressure of the working fluid can be found. If the working fluid under consideration is

to emerge as a saturated vapor from the turbine, the outlet temperature is generally chosen such that an isentropic expansion results in saturated vapor at 120°F (condenser temperature). This constraint applies to all but topping and Brayton cycles.

Consideration of the P-H and/or T-S diagrams for the working fluid will give a good indication of what operating temperatures and pressures are best. The T-Q diagram assures that there will be no pinch point in the heat exchanger by illustrating the heat exchanger temperature profile. The information gained from the diagram, namely outlet temperature and flow rate of the working fluid, can then be used to calculate net work output from the cycle and its efficiency. A sample calculation is presented in Appendix 7.5.

For Brayton cycles a similar procedure may be followed, recalling that the most important factor governing turbine outlet conditions is the specific volume of the working fluid since pump work is considerable. A sample calculation is presented in Appendix 7.5.

4. RESULTS AND DISCUSSION

4.1 Effect of Operating Temperature and Pressure on Power Cycle Efficiency and Output

4.1.1 Ground Water Temperature

The temperature of the ground water stream is specified as 536°F at the inlet to the power cycle. There is some flexibility with the ground water return temperature; however, and in the case of dual cycle systems, the intermediate water temperature is unspecified. To see the effect of the latter two temperatures on cycle efficiency, it is necessary to recall that Carnot efficiency for such a cycle is defined as:

$$\eta_c = \frac{T_H - T_C}{T_H} \quad (3)$$

where T_H is the temperature of the heat source (the ground water temperature in this case) and T_C is the temperature of the heat sink, corresponding to the low temperature of the working fluid (120°F).

As the ground water stream exchanges heat with the power cycle, its temperature drops with a corresponding decrease in η_c obtainable. This necessitates the evaluation of an integral Carnot efficiency, $\overline{\eta_c}$ which takes into account this variation of T_H (see Appendix 7.2). This integral efficiency will depend on the range of T_H from $T_{H,in}$ to $T_{H,out}$ and on the value of T_C . Figure 10 indicates the variation of $\overline{\eta_c}$ as a function of T_{HH} , T_{HC} being fixed at 131°F. Figure 11 indicates the variation of $\overline{\eta_c}$ as a function of

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INTEGRAL CARNOT EFFICIENCY OF A SYSTEM
WITH T_H VARYING FROM 536°F TO $T_{H_{out}}$
AND $T_C = 120^\circ\text{F}$

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T_{HC} , T_{HH} being fixed at 536°F. From these two figures it is clear that η_c increases with increasing T_{HH} or T_{HC} which means that increasing the range and absolute value of T_H results in increased efficiency.

Simultaneously, although η_c increases with increasing T_{HC} (assuming T_{Hin} is fixed), the net work output of the power cycle will decrease since the total amount of heat transferred per pound of ground water decreases:

$$\int dW = \int \dot{m} dh = \int_{T_{Hout}}^{T_{Hin}} \dot{m} C_p(T) dT \quad (4)$$

or

$$W = \text{power out} = \dot{m} \overline{C_p} (T_{Hin} - T_{Hout}) \quad (5)$$

From Eq. (5) it is clear that as T_{Hout} increases, $(T_{Hin} - T_{Hout})$ decreases, resulting in a net decrease in power output. The interaction between η_c and net power output necessitates an optimization procedure to achieve the best balance.

4.1.2 Working Fluid Temperature

The temperature range of the working fluid has the same general effect on power cycle output and efficiency as does ground water temperature, as discussed in Sect. 4.1.1. The limits of working fluid temperature are somewhat lower, about 120 - 520°F, to permit a temperature gradient for heat transfer.

The temperature range of a particular working fluid is determined by its thermodynamic properties. Due to the 3000 psia pressure constraint, some of the fluids considered could not be used in the upper temperature range, and hence were used in the lower temperature part of a dual cycle. Others could only be used in the upper temperature range (above the critical temperature).

A practical constraint on the temperature range which was found to exist with some working fluids can be seen from the T-Q diagram presented in Fig. 12, which shows the temperature profiles of isobutane and the ground water as a function of the fraction of total heat transferred between the two streams. To maximize the efficiency of the heat exchange, the shaded area between the two temperature profiles which corresponds to inefficiencies must be minimized. The efficiency can be maximized in this example by allowing the isobutane stream to reach a maximum temperature of T^* , after which point the two temperature profiles begin to diverge significantly. A second cycle would then be needed to take advantage of the remaining high temperature portion of the ground water heat source. Once again an optimum balance must be determined.

4.1.3 Working Fluid Pressure

Working fluid operating pressure affects cycle efficiency and power output as a result of nonidealities in the pump and turbine stages (see Sect. 2.3), as well as inherent thermodynamic properties of the fluid. The working fluid condensate pump has been assumed to be 75% efficient and the turbine 85% efficient. The reversible pump work must be increased by a factor of $1/0.75$ while the reversible turbine output is decreased by a factor of 0.85. Both pump work and turbine output depend on pressure as may be seen from Eqs. (6) and (7).

$$W_{\text{pump}}^{\text{ideal}} = \dot{m} \int_{P_1}^{P_2} v dP = \dot{m} \bar{v} (P_2 - P_1) \Big|_{\text{const}} \quad (6)$$

$$W_{\text{turbine}}^{\text{ideal}} = \dot{m} \int_{P_2}^{P_1} dH = \dot{m} [h(P_1) - h(P_2)] \Big|_{\text{const}} \quad (7)$$

The net effect of pressure clearly depends on the variation of enthalpy with pressure and is not obvious from the above discussion. However, it has been found that for most of the working fluids considered, the net power output increases as the pressure increases above the critical pressure and reaches a maximum at about twice the critical pressure. Further increases in operating pressure result in decreased net power output. This effect may be seen for Freon-11 and isobutane in Figs. 13 and 14, respectively.

Although the above effects have been attributed to pump and turbine inefficiencies, the reason for the positive or negative effects of pressure changes is the variation of $h(P)$ as has been mentioned. The importance of this dependency may be seen in Fig. 15. The result of changing the operating pressure from 1000 to 3000 psia is summarized in Table 4.

Table 4. Effect of Pressure on Isobutane Cycle Output

Pressure (psia)	@ 75% Eff. Pump Power Input (Btu/lb-sec)	@ 85% Eff. Turbine Power Output (Btu/lb-sec)	Net Power Output (Btu/lb-sec)
200-1000	$\Delta h_1 / 0.75 = 9.3$	$0.85 \Delta h_3 = 16.0$	6.7
1000-3000	$(\Delta h_2 - \Delta h_1) / 0.75 = 14.7$	$0.85 (\Delta h_4 - \Delta h_3) = 14.6$	-0.1
200-3000	$\Delta h_2 / 0.75 = 24.0$	$0.85 \Delta h_4 = 30.6$	6.6

The reason for this effect is the shape of the constant entropy curves. As may be seen in Fig. 15, the curves are essentially parallel above 2000 psia, thus taking into account the previously mentioned efficiencies results in a decreased net power output.

$P_{\text{critical}} = 640 \text{ psia}$

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POWER INPUTS/OUTPUTS FOR FREON-11
SINGLE CYCLE

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FIG.
13

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POWER INPUTS/OUTPUT FOR HYPOTHETICAL
ISOBUTANE SINGLE CYCLE, NEGLECTING
CONDENSER FAN POWER

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FIG.
14

4.2 Improvement of LASL Steam Cycle

4.2.1 Moisture Removal

One of the drawbacks of the LASL steam cycle is the low turbine efficiency of 79%. The supercritical cycles considered were assumed to have 85% efficient turbines which is a reasonable assumption for turbines which are run "dry," i.e., with no vapor condensation. In the LASL cycle turbine, however, condensation of the steam does take place. Outlet steam quality is only about 0.84 vapor. The decrease in turbine efficiency with moisture content is about 1% per % of average turbine moisture.

Moisture removal is a common practice in present-day turbine design. At various positions in the low-pressure end of the turbine, the moisture is extracted by the use of grooved turbine blades and cycled to the condenser inlet. Using five stages of moisture removal in the turbine (see Table 10 in Appendix 7.3 for the characteristics of each stage), net power output from the LASL steam cycle increases by about 0.2 Mw to 33.2 Mw, an increase of about 0.6%.

4.2.2 Regenerative Feed Water Heating

Regenerative feed water heating incorporated in the steam cycle of the LASL scheme results in higher power output and better cycle efficiency. Figure 16 shows the power output from the steam cycle as a function of feed water temperature.

The feed water is heated by saturated steam extracted from the turbine at one or more of the following pressures: 50, 30, 20, 10, and 5 psia corresponding to temperatures of 281, 250.3, 228, 193.2, and 162.2°F, respectively. Five stages of moisture removal were also incorporated in the turbine design at these pressures. It was assumed, based on conventional power cycle design procedures, that resulting feed water temperatures would be 5°F lower than steam inlet temperatures. A complete discussion of regenerative feed water heating conditions is given in Appendix 7.3.

According to Fig. 16 a maximum power output of 34.3 Mw is obtainable with a single stage of feed water heating (to 188°F) for the proposed steam cycle. This compares with 33.2 Mw obtainable without feed water heating but with moisture removal (i.e., an inlet temperature of 120°F), and thus results in a 3.9% increase in power output for the top part of the cycle of 2.6% for the overall LASL scheme.

Two cases of two-stage regenerative feed water heating were also tried; these results are also indicated on Fig. 16. It appears that for either single or dual-stage feed heating, a feed temperature of 180 to 200°F will produce maximum power output. The gain in power production and thermal efficiency afforded by feed water heating would, of course, be weighed against the increased capital and operating costs.

4.3 Evaluation of Freon Working Fluids

The results of a preliminary survey of several freons as possible working fluids are presented in Table 5. Based on these results, a detailed study was made of Freon-11, -21, -115, and -502. Freon-11 was considered for a single cycle and for a dual cycle in conjunction with Freon-115. The other freons were evaluated in dual cycles. The results of these investigations are presented in Table 6 and discussed below. References for the working fluid thermodynamic properties may be found in Table 12 and Appendix 7.9.

4.3.1 Freon-11 Single Cycle

The single Freon-11 cycle was found to be the best cycle of all those evaluated. As may be seen in Table 6, the power output of the single Freon-11 cycle is 57.7 Mw, making it 13.8% more productive than the steam/isobutane cycle proposed by LASL.

Figure 17 shows the pressure-enthalpy diagram for the Freon-11 cycle at three operating pressures; the optimum was found to be about 1000 psia (see Fig. 13). Figure 18 shows the temperature-total entropy change plot for this case. Under these conditions it was found that the temperature profiles of Freon-11 and the ground water streams in the primary heat exchanger matched quite well (see Fig. 19). The effect of going to either higher or lower pressures may be seen in Table 6 and Fig. 13.

4.3.2 Freon-11 Topping/Bottoming Cycle

As a third attempt to utilize more of the heat in the ground water stream, a Freon-11 topping/bottoming cycle was considered. It was hoped that this would allow the freon temperature to exceed 480°F (which is the maximum in the single cycle), thereby using more of the ground water energy. However, this cycle configuration does not appear practical due to the inability to superheat the Freon-11 sufficiently to allow a supercritical bottoming cycle.

4.3.3 Freon-21 Single Cycle

Table 6 indicates that Freon-21 is a slightly less efficient working fluid than Freon-11. The reason for this is that ΔH_{vap} is higher and $\partial H / \partial P|_s$ is lower for Freon-21 than for Freon-11.

The cycle has, nevertheless, been evaluated, since it is comparable to Freon-11 and may have advantages not apparent at this time such as economic or thermal stability benefits. The results of this evaluation are summarized in Table 6.

Table 5. Preliminary Evaluation of Working Fluids

Freon	T _c (°F)	P _c (psia)	Rankine	Brayton	High T	Low T	Single	Dual	Latent ^a Heat @ 120F	$\frac{\partial H}{\partial P} _S$ @ 120F x 10 ³	C _p ^c (Btu/lb-°F) ²	Comments
11	388.4	640	x		x	x	x	x	73.2	31.1	0.21	
12	233.6	597	x			x		x	54.0	3.5	0.235	
13	83.9	561		x	x	x	x	x	-	-		Other gases more favorable
13B1	152.6	575	x			x		x	24.5	17.5	0.21	Insufficient data
14	-50.2	543		x	x	x	x	x	-	-		Other gases more favorable
21	353.3	749.4	x		x		x	x	92	26.6	0.256	
22	204.8	721.9	x			x		x	68	5.71	0.306	Good, but #11 is better
23	78.1	701.4		x	x	x	x	x	-	-		Other gases more favorable
113	417.4	498.9	x		x		x	x	62	27.3	0.218	$\frac{\partial H}{\partial P} _S < \frac{\partial H}{\partial P} _{sat \text{ vap}}$
114	294.3	473	x			x		x	50	9.5	0.246	$\frac{\partial H}{\partial P} _S < \frac{\partial H}{\partial P} _{sat \text{ vap}}$
115	175.9	458	x			x		x	32	9.3	0.263	
152a	236.3	652	x			x		x	106	52.5	0.418	Insufficient data
216	356	399.5	x		x		x	x	49	21.2	0.17	$\frac{\partial H}{\partial P} _S < \frac{\partial H}{\partial P} _{sat \text{ vap}}$
C318	239.6	403.6	x		x	x	x	x	39	7.9	0.27	$\frac{\partial H}{\partial P} _S < \frac{\partial H}{\partial P} _{sat \text{ vap}}$
500	221.9	641.9	x			x		x	63	17.5	0.290	
502	194	619	x			x		x	44	30	0.305	Insufficient data
503	67.1	60.7		x	x	x	x	x	-	-	0.290	Other gases more favorable
504	151.5	690.5	x			x		x	52	27.5	0.423	Insufficient data
<u>Other Fluids</u>												
Ethane	90	710		x	x	x	x	x	-	-	2.50	
Ethylene	50	742		x	x	x	x	x	-	-		
Propane	206.3	617.4	x			x		x	120	37.5	0.65	
Propylene	197.2	670.3	x			x		x	120	40	0.61	
Isobutane	275	529	x		x	x	x	x	124	33.3	0.60	
Ammonia	271.4	1657	x		x	x	x	x	454	315	1.43	Decomposes
SO ₂	315.4	1143	x		x	x	x	x	128	51.1	0.60	
CO ₂	87.9	1070		x	x	x	x		-	-	2.50	
H ₂ O	705.4	3206	x		x			x	1023	947	1.0	

^aBtu/lb^bBtu/lb-psia^cat 86°F

Table 6. Summary of Individual Rankine Cycle Performance

Working Fluid	Heating Fluid		Working Fluid				Power (Mw)				Efficiency (%)			Cycle* Type
	T _H (°F)	T _C (°F)	T _H (°F)	T _C (°F)	P (psia)	m (lb/sec)	Pump	Turbine	Condenser	Net	Actual	Carnot	Relative	
Freon-11	536	343	520	120	1200	1481	5.08	39.82	1.28	33.46	21.9	37.6	58.2	TD
	536	280	520	120	700	1576	3.09	46.61	1.68	41.84	21.0	34.6	60.7	TD
	536	240	520	120	1400	2382	9.57	51.23	2.03	39.63	17.3	32.6	53.1	TD
	536	131 ^a												
	536	170	480	120	1000	2699	7.67	67.73	2.38	57.68	20.7	28.8	71.9	S
	536	200	520	120	1200	2491	8.55	66.97	2.15	56.27	21.9	30.4	72.0	S
	536	170	425	120	700	2785	5.46	64.89	2.38	57.05	20.5	28.8	71.2	S
Steam	536	343	390	120	220	130.5	0.12	34.41	1.25	33.04	21.6	37.6	57.4	TD
Isobutane	343	131	297	120	588	900.1	3.51	22.59	1.39	17.69	11.5	16.0	71.9	BD
	340	140	290	120	600	854.9	3.45	22.22	1.32	17.45	12.0	15.5	77.4	BD
	360	140	340	130	1000	973.3	7.05	29.66	1.52	21.09	13.2	16.8	78.6	BD
	390	140	380	130	2000	1032	15.75	40.69	1.62	23.32	12.8	18.5	69.2	BD
	400	140	390	140	3000	1049	24.41	48.89	1.66	22.82	12.0	19.0	63.2	BD
	536	140 ^a												
	536	160	520	120	600	814.7	3.29	37.97	1.22	33.46	18.1	26.7	67.8	T
Freon-502	420	140	390	120	1700	690.5	8.88	28.47	1.19	18.40				B
	536	280	520	120	650	756.6	3.36	34.58	1.76	29.46	14.8	34.6	42.8	TD(R)
	240	150	220	120	700	1204	1.81	7.55	0.62	5.12	7.9	10.5	75.2	BD
Freon-115	240	140	220	120	800	1842	3.75	9.90	0.66	5.49	7.6	10.0	76.0	BD
Freon-21	536	280	520	120	1500	1501	6.94	48.43	1.62	39.87	20.0	34.6	57.8	TD
Ammonia	536	150	430	120	1764	463.5	5.07	55.93	2.34	48.52	16.5	27.7	59.6	S
	536	165	520	120	2866	437.0	8.32	60.73	2.21	50.20	17.8	28.5	62.5	S
Sulfur Dioxide	536	160	500	120	1300	1443	5.49	57.94	2.29	50.16	17.5	28.2	62.1	S
	536	160	500	120	1150	1413	4.70	61.80	2.31	54.79	19.2	28.2	68.1	S

• TD = high temperature cycle of a dual cycle system

BD = low temperature cycle of a dual cycle system

S = single cycle

T = topping cycle

B = bottoming cycle

(R) = regenerative feed heating used

^aNot calculated; T-Q diagram pessimistic

P = 1000 psia

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TEMPERATURE- Δ TOTAL ENTROPY DIAGRAM FOR
THE FREON-11 SINGLE CYCLE

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FIG.
18

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T-Q DIAGRAM FOR FREON-11 at 700
AND 1000 PSIA

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FIG.
19

4.3.4 Freon-502 and Freon-115

Both Freon-502 and -115 were considered for low temperature cycles in dual cycle applications (see Table 5). As Freon-115 was the more efficient of the two (Table 6), it will be the only fluid discussed for low temperature, dual-cycle applications.

Figures 20 and 21 show the P-H and T-S diagrams, respectively, for Freon-115 used in a typical low temperature application. From Fig. 20 it is clear why this working fluid is used only for low temperature applications: for the isentropic expansion step to end at 120°F on the saturated vapor curve, the pressure at 520°F would have to be approximately 50,000 psia (based on extrapolation of the P-H diagram) - far above the pressure constraint.

The efficiency and power calculations of this cycle are summarized in Table 6 and the cycle is compared with other cycles such as a Freon 11/115 dual cycle (Table 7). (A temperature-total entropy change diagram for this case is presented as Fig. 22.) From Table 7 it is clear that the single Freon-11 cycle is superior to the Freon 11/115 dual cycle configuration.

Table 7. Efficiency and Power Output of Various Power Cycle Combinations

Cycle Type	Component Fluids		Power Output (Mw)	Efficiency (%)
	(1)	(2)		
Single	Freon-11		57.7	20.7
Dual	Freon-11	Freon-115	47.3	17.4
Dual ¹	Sat'd steam	isobutane	50.7	16.5
Single ²	SO ₂		54.8	19.2

¹See Sect. 2.3.

²See Sect. 4.6.

4.4 Evaluation of Isobutane as a Working Fluid

Isobutane was considered for use as a working fluid in single cycles, topping/bottoming cycles, and dual cycles in conjunction with saturated steam and Freon-11. The results of this investigation show that isobutane can be used most efficiently in the topping/bottoming cycle configuration, yielding about 51.9 Mw. Since the dual cycle calculations are similar to other calculations previously discussed (Sects. 3.2 and 4.3), only the results are presented (Table 8).

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P-H DIAGRAM FOR FREON-115

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FIG.
20

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T-Q DIAGRAM FOR FREON-115

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FIG.
21

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TEMPERATURE-TOTAL ENTROPY DIAGRAM FOR
THE FREON-11/115 CYCLE

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FIG.
22

Table 8. Isobutane Cycle Performance

Cycle Type	Component Fluids		Cycle Output (Mw)	Efficiency (%)
	High Temperature	Low Temperature		
Dual	Isobutane	Freon-115	35.0	12.9
Dual	Freon-11	Isobutane	51.2	16.7
Topping/Bottoming	Isobutane	Isobutane	51.9	18.1

4.4.1 Isobutane Topping/Bottoming Cycle

To extract most of the heat from the ground water stream without going to pressures above 3000 psia requires heating the isobutane to a high specific enthalpy, but at low pressure. When the isobutane is expanded isentropically to the low operating pressure, it is still superheated (see Fig. 23). This makes it ideal for a topping/bottoming cycle. The P-H and T-Q diagrams for such a cycle configuration are presented as Figs. 23, 24, and 25, respectively. The topping cycle is illustrated in Fig. 23 as GCDF and the bottoming cycle as ABEF. It is clear from the T-Q diagram that the temperature profile of compressed liquid isobutane is well matched to the profile of the superheated vapor. Thus, the inefficiencies are minimized. Figure 26 shows a temperature-total entropy change for the cycle in question. Table 8 shows that this configuration is the most efficient of all the cycles utilizing isobutane yielding 51.9 Mw.

4.4.2 Single Isobutane Cycle

It was found that isobutane cannot be used at high temperatures and still expand to saturated vapor at the minimum temperature constraint (120°F) for operating pressures less than 3000 psia. Even if the pressure constraint is not obeyed, the cycle output decreases with increasing pressure above about 1500 psia (Fig. 14). Hence, it has been concluded that the only efficient way to use isobutane as the sole working fluid is in a topping/bottoming cycle configuration.

4.5 Evaluation of Ammonia as a Working Fluid

Ammonia was studied as the working fluid in a single supercritical Rankine cycle. Two different operating pressures were considered (1764 and 2866 psia) and the cycles are illustrated in a temperature-entropy diagram in Fig. 27.

Total power output from the ammonia single cycles was found to be 48.5 and 50.2 Mw for operating pressures of 1764 and 2866 psia, respectively.

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P-H DIAGRAM FOR ISOBUTANE
TOPPING/BOTTOMING CYCLE

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FIG.
23

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T-Q DIAGRAM FOR TOPPING CYCLE OF
ISOBUTANE TOPPING/BOTTOMING CYCLE

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FIG.
24

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T-Q DIAGRAM FOR BOTTOMING CYCLE OF
ISOBUTANE TOPPING/BOTTOMING CYCLE

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FIG.
25

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TEMPERATURE-TOTAL ENTROPY CHANGE FOR
THE ISOBUTANE TOPPING/BOTTOMING CYCLE

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T-S DIAGRAM FOR AMMONIA SHOWING
CYCLES AT TWO DIFFERENT PRESSURES

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Thus an ammonia cycle appears to be slightly less efficient than the proposed LASL system.

Consideration of the T-Q diagram (Fig. 27) shows that the heat capacity variation of the fluid is quite favorable; hence this is not the cause of the low efficiency. The problem lies in the high specific volume of the fluid at 120°F. Table 6 shows that pump work required for the 2866 psia cycle is 8.3 Mw. Another factor is the variation of enthalpy with pressure during the constant entropy expansion step which is quite low, resulting in low turbine work per pound of fluid.

4.6 Evaluation of Sulfur Dioxide as a Working Fluid

Sulfur dioxide was considered for use in a single fluid Rankine cycle at two different pressures: 1150 ($P_c = 1143$ psia) and 1300 psia. Net work output obtainable from the cycles is 54.8 and 50.2 Mw, respectively (Table 6). SO_2 thus seems to be a very efficient working fluid, second only to Freon-11 in single cycle performance. The net power output is roughly 8.1% greater than that obtainable from the proposed LASL scheme. A P-H diagram for SO_2 presenting the two cycles considered is shown as Fig. 28.

4.7 Brayton Cycles

The possibility of using a Brayton cycle operating between 536 and 131°F was investigated. This proposal represents a great reduction in total cost and ease of operation (especially for the turbine where no moisture is present). The pressure-enthalpy diagram for ethylene (see Fig. 29) represents a typical Brayton cycle exchanging heat with the geothermal stream from 536 to 131°F (cycle AGHK). It can be seen that in order to obtain an inlet gas temperature of 120°F to the heat exchanger (thus providing an 11°F ΔT in the heat exchanger), the condition of the working fluid after the cooling step would be a mixture of vapor and liquid for turbine outlet pressures less than 750 psia (cycle ABCD). If the outlet pressure from the turbine is increased above 750 psia (for example cycle AGMD), pump inlet temperatures smaller than 120°F are needed to have the 11°F approach at the inlet of the heat exchanger. For this reason only working fluids which have a low critical temperature and pressure were examined.

As a result the gas cycles were operated so as to exchange heat with the geothermal from 536 to 343°F (cycle AGHK). The results obtained for the working fluids - ethylene, ethane, and CO_2 - are presented in Table 9. The highest net work produced is 6.3 Mw using CO_2 as the working fluid. The pump work in Brayton cycles is responsible for the low power output. For CO_2 the pump work is about 88% of the turbine work while for the steam cycle it is only 0.4%. The high specific volume encountered in the pumping of gases (alternatively low value of $\partial H / \partial P$) accounts for the high pumping power.

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P-H DIAGRAM FOR SO_2

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Table 9. Results of Brayton Cycles (536-343°F)

Working Fluid	Pressure (psia)	Heat Exchanger ¹ Temperature (°F)		Flow Rate (lb/sec)	Work (Mw)			Net Power
		T _{out}	T _{in}		Turbine	Condenser	Pump	
Ethylene	2500	520	315	1075	73.22	1.56	69.14	2.52
Ethane	2500	520	310	2460	163.14	4.18	153.03	5.93
CO ₂	2000	520	320	2282	92.03	1.50	80.84	9.69

	Efficiency (%) ²	
	Cycle	Relative
Ethylene	1.6	4.3
Ethane	3.9	10.2
CO ₂	6.3	16.8

¹T_{in} = outlet pump temperature. The inlet temperature to the pump was 120°F in all cases.

² η_{Carnot} = 37.8%

Brayton cycles need to be operated in a high pressure, low specific volume and low temperature region for geothermal energy utilization. For the working fluids investigated, these conditions could not be met.

5. CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

The primary objective of this project has been to maximize the efficiency and power output of a power cycle extracting energy from a given low temperature geothermal source. The results are presented in Tables 7, 8, and 9.

The power output of the LASL proposed steam-isobutane cycle can be improved approximately 2.8% by use of regenerative feedwater heating to give an output of 52.1 Mw.

Four of the cycles investigated have higher outputs than the LASL proposal. They are: (1) a single Freon-11 cycle, (2) a single SO₂ cycle,

(3) an isobutane/isobutane topping/bottoming cycle, and (4) a Freon-11-isobutane dual cycle. For use with a sensible heat source, a supercritical Rankine cycle appears to be the optimum design. Brayton gas cycles were found to have extremely low (9.7 Mw) power outputs due to the high pump work required.

Of all the combinations investigated the Freon-11 single cycle yielded the greatest improvement (13.8%) over the LASL proposal. As a single cycle, it may also have significant capital and operating cost benefits.

5.2 Recommendations

Recommendations which can be made fall into two categories: (1) consideration of other cycles and fluids and (2) improvement of the accuracy of existing calculations.

Fluids which appear promising but for which thermodynamic data were not available include Freon-152A and Freon-502. The accuracy of existing calculations could be improved by determining pump and turbine efficiencies more accurately and by making better estimates of the condenser fan power required.

The best cycles should be analyzed on an economic basis. The optimum in many cases may be at power outputs much less than the maximum.

6. ACKNOWLEDGEMENTS

We wish to thank S.L. Milora and R.N. Lyon of ORNL for their thoughtful suggestions regarding possible cycle schemes and working fluids. We also express our appreciation to L.C. Fuller for his invaluable help with the ORCENT computer program used in some of the calculations.

7. APPENDIX

7.1 Steam Cycle Energy Balance

The operating conditions for the steam cycle (top part of the proposed LASL cycle) were computed by performing an energy balance on the heat exchanger (Fig. 30). Inlet and outlet geothermal source temperatures are known; these are 536 and 343°F, respectively. The steam enters the exchanger as a compressed liquid and leaves as saturated vapor. Therefore at some point in the exchanger, the pinch point, the liquid will begin to vaporize at constant T and P until it becomes saturated vapor at the exchanger outlet. (The steam cycle calculations were made assuming a pinch point existed. All other cycles assume a minimum 10°F temperature difference throughout the heat exchanger.) The first law can be written considering the water stream within the heat exchanger shell as the system:

$$dE_w = dQ_w - dW_s + h_{in_w} \dot{m}_{in_w} - h_{out_w} \dot{m}_{out_w} \quad (8)$$

$$dE_w = 0, \text{ steady state; } dW_s = 0, \text{ no shaft work}$$

$$\dot{m}_{in_w} = \dot{m}_{out_w} = \dot{m}_w \quad (9)$$

$$dQ_w = (h_{out} - h_{in})_w \dot{m}_w \quad (10)$$

A similar expression may be applied to the geothermal stream.

$$dQ_s = (h_{out} - h_{in})_s \dot{m}_s \quad (11)$$

With the relation (for no phase change),

$$\Delta H = \int (h_{out} - h_{in}) d\dot{m} = \int \dot{m} C_p(T) dT \quad (12)$$

we can rewrite Eq. (10) as

$$dQ_w = \int_{T_{in_w}}^{T_p} \dot{m}_w C_{p_w}(T) dT_w + \dot{m}_w \Delta H_{v(w)} \quad (13)$$

Numbers in parentheses refer to values calculated according to Eqs. (19) and (20).

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ENERGY BALANCE ON LASL STEAM CYCLE
HEAT EXCHANGER

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Similarly, Eq. (11) becomes,

$$dQ_s = \int_{T_{in_s}}^{T_{out_s}} \dot{m}_s C_{p_s}(T) dT_s \quad (14)$$

The heat terms must be equal for an exchanger with no heat losses through its shell. Equations (13) and (14) may, therefore, be integrated assuming constant heat capacities and set equal to yield:

$$\dot{m}_s C_{p_s} (T_{out_s} - T_{in_s}) = -\dot{m}_w [C_{p_w} (T_p - T_{in_w}) + \Delta H_{v(w)}] \quad (15)$$

Furthermore, we know that at the pinch point the temperature difference between the geothermal stream and working fluid is very small and may be assumed zero. Therefore a heat balance for the bottom part of the exchanger yields:

$$\dot{m}_w C_{p_w} (T_p - T_{in_w}) = \dot{m}_s C_{p_s} (T_p - T_{out_s}) \quad (16)$$

Equations (15) and (16) may now be combined to eliminate $\dot{m}_s C_{p_s} / \dot{m}_w$.

$$\frac{(T_p - T_{out_s})}{C_{p_w} (T_p - T_{in_w})} = - \frac{(T_{out_s} - T_{in_s})}{C_{p_w} (T_p - T_{in_w}) + \Delta H_{v(w)}} \quad (17)$$

The following variables are known:

$$\dot{m}_s = 683 \text{ lb/sec}$$

$$T_{in_s} = 536^\circ\text{F}$$

$$T_{out_s} = 343^\circ\text{F}$$

$$C_{p_s} = 1.1 \text{ (assumed constant)}$$

$$T_{in_w} = 120^\circ\text{F}$$

$$C_{p_w} = 1.0 \text{ (assumed constant)}$$

ΔH_v may be linearized and made into a function of T_p in the expected range of T_p (between 380 and 400°F).

$$\Delta H_V \approx 1199 - 0.93 T_p \quad (18)$$

Substituting these numbers into Eq. (17) and rearranging,

$$0.110 T_p^2 + 843.5 T_p - 345,300 = 0 \quad (19)$$

$$\therefore T_p = 390^\circ\text{F}$$

Inserting this value into Eq. (16), we can then solve for $\dot{m}_w$:

$$\dot{m}_w = \frac{683(1.1)(T_p - 343)}{(1.0)(T_p - 120)} = 130.8 \text{ lb/sec} \quad (20)$$

The discrepancy between $\dot{m}_w$ as calculated here and as presented in Fig. 2 is due to the assumption of constant heat capacities.

7.2 Carnot Efficiency

The ratio of the amount of work obtained from a cyclic process to the amount of heat input is defined as the thermal efficiency. Thus, thermal efficiency is a measure of the conversion of heat added to work produced. The Carnot efficiency represents a reversible operation in which heat is taken from a hot temperature reservoir (T_{hot} is finite), used to produce work, and rejected to a low temperature reservoir (T_C different from absolute zero). For a cycle operating between a constant low temperature and a varying high temperature (see Fig. 31), the Carnot efficiency can be found by taking incremental amounts of heat (δQ_H) to produce incremental amounts of work (δW) and rejecting a fraction of the heat (δQ_C) to the cold temperature (T_C). All contributions are added over the temperature range of the hot temperature reservoir.

An entropy balance around one of the Carnot engines of Fig. 31 yields:

$$\delta Q_C = - \frac{T_C}{T_H} \delta Q_H \quad (21)$$

An energy balance gives:

$$\delta W = \delta Q_H + \delta Q_C \quad (22)$$

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FIG.
31

Substituting Eq. (21) into Eq. (22),

$$\delta W = \delta Q_H \left(1 - \frac{T_C}{T_H}\right) \quad (23)$$

and using the relation $\delta Q_H = -mC_p(T_H)dT_H$ to eliminate δQ_H from Eq. (23), the Carnot efficiency can be expressed as:

$$\eta_c = \frac{W}{Q_H} = - \frac{\int_{T_{H_i}}^{T_{H_f}} C_p(T_H) \left(\frac{T_C}{T_H} - 1\right) dT_H}{\int_{T_{H_i}}^{T_{H_f}} C_p(T_H) dT_H} = - \frac{[T_C \Delta S - \Delta H]_{T_{H_i}}^{T_{H_f}}}{\Delta H|_{T_{H_i}}^{T_{H_f}}} \quad (25)$$

If Eq. (25) is applied to the LASL proposed cycle operating between a hot reservoir varying from $T_{H_i} = 536^\circ\text{F}$ (995°R) to $T_{H_f} = 131^\circ\text{F}$ (590°R) and a constant $T_C = 120^\circ\text{F}$ (579°R), the Carnot efficiency can be found by graphical integration or use of the steam tables (9) to be 26.7%. Thus, a maximum of 26.7% of the heat input is converted to work in reversible operation of the cycle which provides a basis for evaluating the performance of the LASL proposal and alternate cycle schemes.

7.3 Regenerative Feed Water Heating

Regenerative feed water steam cycle calculations were made by hand and then checked against values generated by the computer program ORCENT. This program incorporates the turbine design methods used by the General Electric Company (3, 4). Five stages of moisture removal were included in the turbine design; turbine characteristics at each moisture removal stage are presented in Table 10.

Table 10. Characteristics of Moisture Removal - ORCENT Results

	Pressure (psia)	Temp- erature (°F)	Moisture (%)	Enthalpy (Btu/lb)	Entropy (Btu/lb-°R)
Turbine Inlet	230.0	393.7	0.00	1200.1	
Before Moisture Removal #1	50.0	281.0	8.07	1099.5	1.5578
After Moisture Removal #1	50.0	281.0	7.91	1101.0	1.5598
Before Moisture Removal #2	30.0	250.3	9.86	1070.9	1.5681
After Moisture Removal #2	30.0	250.3	9.60	1073.4	1.5716
Before Moisture Removal #3	20.0	228.0	10.98	1050.9	1.5786
After Moisture Removal #3	20.0	228.0	10.57	1054.8	1.5843
Before Moisture Removal #4	10.0	193.2	12.68	1018.8	1.5970
After Moisture Removal #4	10.0	193.2	11.65	1028.9	1.6124
Before Moisture Removal #5	5.0	162.2	13.58	995.2	1.6256
After Moisture Removal #5	5.0	162.2	10.67	1024.3	1.6724
Turbine Outlet	1.69	120.0	13.67	973.5	1.6920

With a heated feed stream, the criterion of saturated steam at the heat exchanger exit (and turbine entrance) requires lower steam peak temperature and higher mass flow rates. This can be seen in the T-Q diagram, Fig. 32. The dotted line shows the behavior of the steam cycle with no regenerative feed-water heating. A pinch point is reached at a temperature of 390°F ($\dot{m} = 130.5$ lb/sec). Heating the feed to 188.2°F and keeping the same mass flow rate (dashed line) results in a fairly low pinch point temperature ($\sim 370^\circ\text{F}$) and a slightly superheated steam at the exchanger outlet. Raising the mass flow to 139.4 lb/sec results in a T_p of 381°F and saturated steam at the outlet. Equations (19) and (20) were used to calculate the values of T_p and $\dot{m}$ for the different feed temperatures (T_{in}) considered. These results are presented in Table 11.

Table 11. Regenerative Feed Water Heating Results

Case	No. of Heaters	Inlet Steam Temp(s) to Heaters (°F)	Feed Water Outlet Temp. (°F)	Turbine Inlet		Steam Flow Rate (lb/sec)	Eff. (%)	Turbine Power Output (Mw)	Approximate Pump and Fan Work (Mw)	Approximate Net Work (Mw)
				T (°F)	P (psia)					
1	1	162.2	157.2	387	214	134.7	23.31	35.7	1.41	34.3
2	1	193.2	188.2	381	198	138.8	23.40	35.8	1.46	34.3
3	1	228.0	223.0	374	182	143.7	23.21	35.5	1.51	34.0
3A	2	228.0/162.2	223.0	374	182	143.7	23.47	35.9	1.51	34.4
4	1	250.3	245.3	369	171	147.1	22.88	35.0	1.54	33.5
4A	2	250.3/193.2	245.3	369	171	147.0	23.31	35.7	1.54	34.2
5	1	281.0	276.0	338	114	152.7	20.29	31.0	1.60	29.4
5A	2	281.0/193.2	276.0	338	114	152.8	20.97	32.1	1.60	30.5

The criteria used for feed heater design (see Fig. 6) are:

countercurrent heat exchanger

$$\Delta T_{\text{hot end}} \geq 5^{\circ}\text{F}$$

$$\Delta T_{\text{cold end}} \geq 10^{\circ}\text{F}$$

Given the inlet temperature and pressure to the turbine and the pressures at which steam is to be extracted from the turbine, the program calculates the required mass flow rate of extracted steam and reports cycle power output in terms of gross turbine efficiency after the electrical generator. (Since all of the net power values quoted in this report represent new power into the generator, we must multiply the computed values of efficiency by the total heat input, divide by generator efficiency, and subtract pump and condenser fan work.) A summary of computed results can be found in Table 11.

7.4 T-Q Sample Calculations

This section demonstrates how the use of T-Q diagrams, for a given fluid and cycle, ensures that a temperature pinch is avoided in the heat exchanger of the cycle, and how operating temperatures and flow rates of the working fluid may be determined.

7.4.1 Single Cycle T-Q Sample Calculation

For the purpose of this calculation, it is assumed that a single Freon-11 cycle is to be used to extract the maximum amount of power from a water stream having the following inlet and exit conditions:

	<u>Inlet</u>	<u>Exit</u>
temperature	536°F	131°F
pressure	~2000 psia	~2000 psia
flow rate	683 lb/sec	683 lb/sec
enthalpy	530 Btu/lb-sec	104 Btu/lb-sec

Also, it is assumed that the inlet and exit conditions which are specified are:

	<u>Inlet</u>	<u>Exit</u>
temperature	120°F	480 °F
pressure	assume P = 1000 psia	

The first step in the calculation involves the pressure-enthalpy diagram for Freon-11, included as Fig. 17. The operating pressure of the cycle is specified as 1000 psia, and a constant pressure line is drawn defining the isobaric heating stage of the cycle. The end points are defined by the entropy, the entropy of saturated liquid at 120°F and the entropy of saturated vapor at 120°F.

Having specified the pressure and entropy of the heating stage of the cycle, its temperature-enthalpy profile may easily be constructed by reading the enthalpy as a function of temperature at 1000 psia. It is desirable to normalize the enthalpies so that the exchanger inlet corresponds to zero enthalpy, and the outlet to an enthalpy of unity. The Freon-11 temperature-enthalpy line in Fig. 19 can then be constructed.

Having constructed the temperature profile of the freon stream, the water stream profile may be constructed. It is known that at the inlet to the exchanger, corresponding to the hot temperature end of the freon stream, the temperature of the ground water is 536°F. If we assume that the specific heat of water is essentially constant from 536 to 131°F, then its temperature profile will be linear and it can be plotted as a straight line from 536°F passing within 10°F (specified minimum ΔT in the heat exchanger) of the freon curve. The ordinate intersection corresponds to the water temperature at the cold end of the heat exchanger.

If the two temperature profiles are constructed and the mass flow of the water stream is known, the heat exchanger is completely specified and is known not to be temperature pinched ($\Delta T \geq 10^\circ\text{F}$). In this case the temperatures and enthalpies at the inlet and exit of the heat exchanger are:

	Inlet		Exit	
	T	h	T	h
water	536°F	530 Btu/lb	170°F	143 Btu/lb
Freon-11	120°F	38 Btu/lb	480°F	136 Btu/lb

The heat exchanged by the water stream is:

$$\begin{aligned}
 Q_s &= \dot{m}_s (h_2 - h_1)_s = 683 \text{ lb/sec} (530 - 143 \text{ Btu/lb}) \\
 &= 264,500 \text{ Btu/sec} = 278.9 \text{ Mw}
 \end{aligned}$$

Since all of this heat is transferred to the Freon-11, the required flow rate of freon may be calculated from Eq. (11):

$$\dot{m}_{wf} = \frac{Q_{wf}}{\Delta h_{wf}} = \frac{264,500 \text{ Btu/sec}}{(136 - 38) \text{ Btu/lb}} = 2699 \text{ lb/sec}$$

Thus, the actual heat exchanger design is:

	<u>Water</u>	<u>Freon-11</u>
inlet temperature	536°F	120°F
exit temperature	170°F	480°F
pressure	2000 psia	1000 psia
flow rate	683 lb/sec	2699 lb/sec

7.4.2 T-Q Calculation for the Isobutane Topping/Bottoming Cycle

The concepts in this sample calculation are identical to those presented in Sect. 7.4.1. The only difference is that the heat source for the bottoming cycle is the superheated isobutane topping cycle rather than the ground water stream which results in a second T-Q diagram. The P-H and T-Q diagrams (Figs. 23, 24, and 25) for this cycle have been included in Sect. 4.4.1.

7.5 Sample Calculation of Power Cycle Performance

This appendix contains a calculation of power output using the Freon-11 single cycle as an example. With specifications of the heat exchanger operation (Appendix 7.4.1) it is possible to calculate overall cycle performance. This consists of evaluating power input or output from: (1) the turbine, (2) the pump, and (3) the condenser fan. The single Freon-11 cycle may be seen in Figs. 17, 18, and 19. The thermodynamic conditions of an ideal cycle are:

	<u>Heat Exchanger</u>		<u>Turbine</u>	<u>Pump</u>
	<u>Inlet</u>	<u>Outlet</u>	<u>Outlet</u>	<u>Inlet</u>
pressure (psia)	1000	1000	33	33
temperature (°F)	120	480	120	120
enthalpy (Btu/lb)	38	136	108	33
entropy (Btu/lb-°R)	0.066	1.195	0.066	1.195
sp. vol. (ft ³ /lb)	0.0113	0.0370	0.0113	1.4

The turbine power output may be expressed as

$$W_T = \dot{m}_{wf} \Delta h_{wf} = 2699 \text{ lb/sec}(136 - 108) = 75,570 \text{ Btu/sec (26)}$$

This is the ideal power output; however, it is assumed that the turbine is 85% efficient (no vapor present) so the actual output becomes:

$$\begin{aligned} W_T &= 0.85 \dot{m}_{wf} \Delta H_{wf} = (0.85)(75,570) = 64,240 \text{ Btu/sec} \\ &= 67.73 \text{ Mw} \end{aligned} \quad (27)$$

This inefficiency affects the turbine outlet conditions, assuming an adiabatic turbine.

$$h_{\text{outlet}} = 136 - 0.85(136 - 108) = 112 \text{ Btu/lb}$$

At a pressure of 33 psia this corresponds to a temperature of 160°F.

The pump power input for a reversible isentropic liquid compression is

$$\begin{aligned} W_p &= \int_{P_1}^{P_2} v_{wf} dP_{wf} \approx \bar{v}_{wf} \Delta P_{wf} \quad (28) \\ &\approx 2699 \text{ lb} (0.0113 \text{ ft}^3/\text{lb}) (1000 - 33 \text{ lb/in.}^2) (144 \text{ in.}^2/\text{ft}^2) \\ &\quad \times (1 \text{ Btu}/778 \text{ ft-lb}) \\ &\approx 5459 \text{ Btu/sec} \end{aligned}$$

The pump is assumed to be 75% efficient, so the actual pump input power is

$$W_p = \frac{1}{0.75} \bar{v}_{wf} \Delta P_{wf} = \frac{5459}{0.75} = 7278 \text{ Btu/sec} = 7.67 \text{ Mw} \quad (29)$$

Alternatively the pump work could be calculated as the enthalpy change for an isentropic rise in pressure. By applying the 75% pump efficiency the proper enthalpy for the heat exchanger inlet could be calculated.

The calculation for the condenser fan input power involves a linear scaleup based on the heat transferred in the condenser. In the original LASL proposal it was estimated that the fan power to transfer 108,000 Btu/sec of latent heat was 1.2 Mw. On this basis the power requirement for the Freon-11 cycle would be:

$$\text{heat transferred} = 2699 \text{ lb/sec} (112 - 33 \text{ Btu/lb}) = 213,800 \text{ Btu/sec}$$

$$\text{fan power} = \left(\frac{1.2 \text{ Mw}}{108,000 \text{ Btu/sec}} \right) (213,800 \text{ Btu/sec}) = 2.38 \text{ Mw}$$

To summarize these power calculations:

$$\text{turbine output} = 67.73 \text{ Mw}$$

$$\text{pump input} = 7.67$$

$$\text{fan input} = 2.38$$

$$\text{Thus, net power output} = 67.73 - 7.67 - 2.38 = 57.68 \text{ Mw}$$

7.6 Cycle Efficiency Sample Calculation

The results of Appendix 7.5 indicate that the single Freon-11 cycle has a net power output of 57.7 Mw. This output is based on a ground water flow rate of 683 lb/sec and a temperature drop from 536 to 170°F. The heat transferred from the ground water stream as calculated from Eq. (11) is 278.9 Mw. Then the thermal efficiency of the cycle will be:

$$\eta = \frac{\text{output power}}{\text{input heat}} = \frac{57.7}{278.9} = 20.7\% \quad (30)$$

From Fig. 10 the Carnot efficiency of the ground water source from 536 to 170°F is 28.8%, and hence the relative efficiency of the Freon-11 single cycle is:

$$\eta_r = \frac{\eta}{\eta_c} = \frac{20.7}{28.8} = 71.9\% \quad (31)$$

7.7 Sources of Thermodynamic Data

Table 12. Sources of Thermodynamic Data

<u>Fluid</u>	<u>References</u>
All freons	(6) & (7)
Water	(9)
Ammonia	(10)
SO ₂	(5)
CO ₂ - (T-S Diagram)	(8)
- P-H Diagram Other Data	(6)
Ethylene	(6)
Propane	(6)
Isobutane	(6)

7.8 Nomenclature

C_p	heat capacity at constant pressure, Btu/lb-°F
E	total energy, Btu/sec
h	specific enthalpy, Btu/lb
H	total enthalpy, Btu/sec
$\dot{m}$	mass flow rate, lb/sec
P	pressure, psia
Q	total heat, Btu/sec
s	specific entropy, Btu/lb-°R
S	total entropy, Btu/sec-°R
T	temperature, °F, °R
v	specific volume, ft ³ /lb
W	total work, Mw, Btu/sec
η	thermal efficiency, %
η_c	Carnot efficiency, %
$\overline{\eta_c}$	integral Carnot efficiency for a variable temperature heat source, %

Subscripts

c	at critical point, condenser
C	cold, heat sink
f	final
F	feed water
gen	generated by system
H	hot, heat source
HH	hot condition of a variable heat source
HC	cold condition of a variable heat source
i	initial

in	inlet to system
net	net output
o	outlet heat exchanger condition of turbine bleed stream
out	outlet, from system
P	pinch point, pump
r	relative
R	condition of turbine bleed stream
s	geothermal stream, shaft
T	turbine
w	water
wf	working fluid

Superscript

— average

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