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STUDY OF PRACTICAL CYCLES FOR GEOTHERMAL POWER PLANTS

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June 15, 1975 to March 15, 1977**

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FOREWORD

This final report documents the work performed under ERDA Contract E(11-1)-2619, "Study of Practical Cycles for Geothermal Power Plants," initiated June 15, 1975 for the Geothermal Energy Division. The work was conducted at the General Electric Research and Development Center, Schenectady, New York. The ERDA program manager was C.B. McFarland.

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SUMMARY

A comparison is made in this study of the performance and cost of geothermal power cycles designed specifically, utilizing existing technology, to exploit the high temperature, high salinity resource at Niland and the moderate temperature, moderately saline resource at East Mesa in California's Imperial Valley. Only two kinds of cycles are considered in the analysis. Both employ a dual flash arrangement and the liberated steam is either utilized directly in a condensing steam turbine or used to heat a secondary working fluid in a closed Rankine (binary) cycle.

The performance of several organic fluids was investigated for the closed cycle and the most promising were selected for detailed analysis with the given resource conditions. Results show for the temperature range investigated that if the noncondensable gas content in the brine is low, a dual flash condensing steam turbine cycle is potentially better in terms of resource utilization than a dual flash binary cycle. (The reverse is shown to be true when the brine is utilized directly for heat exchange.) It is also shown that despite the higher resource temperature, the performance of the dual flash binary cycle at Niland is degraded appreciably by the high salinity and its output per unit of brine flow is almost 20% lower than that of the steam turbine cycle at East Mesa.

Turbine designs were formulated and costs established for power plants having a nominal generating capacity of 50 MW. Three cycles were analyzed in detail. At East Mesa a steam turbine and a binary cycle were compared. At Niland only the binary cycle was analyzed since the high CO₂ content in the brine precludes the use of a steam turbine there. In each case only the power island equipment was considered and well costs and the cost of flash separators, steam scrubbers and piping to the power plant boundary were excluded from the estimate. Steam and organic turbine generator costs were estimated to be approximately \$200 and \$100 per kilowatt respectively although savings in the binary cycle turbine costs are shown in this study to be more than offset by the heat exchanger costs and the high cost of organic condensers. Total installed cost for the steam turbine cycle at East Mesa was estimated to be almost \$600/kW compared to \$660/kW and \$700/kW for the dual flash binary cycles at Niland and East Mesa, respectively.

NOMENCLATURE

A	Area (ft ²)
c _{is}	Isentropic velocity (ft/s)
C _p	Specific heat (Btu/lb/F)
D	Diameter
f	Friction coefficient
f _{foul}	Fouling factor (dimensionless)
G	Mass velocity (lb/hr/ft ²)
h	Blade height (ft) and heat transfer coefficient (Btu/hr/ft ² /F)
H	Specific enthalpy (Btu/lb)
H _{fg}	Heat of Vaporization (Btu/lb)
K	Blade taper factor (dimensionless)
L	Length (ft)
LMTD	Logmean temperature difference (F)
m	Mass flow rate (lb/hr)
M	Mass flow rate of organic fluid (lb/lb brine)
M _{s1}	Mass flow rate of primary steam (lb/lb brine)
M _{s2}	Mass flow rate of secondary steam (lb/lb brine)
n	Number of turbine stages
n _e	Number of turbine exhaust ends
N	Speed (rpm)
N _b	Number of heat exchanger baffles
N _s	Specific speed (dimensionless)
N _t	Number of heat exchanger tubes
P	Pressure (lb/ft ²)
r	Stage pressure ratio (dimensionless)
R	Overall pressure ratio (dimensionless)
Re	Reynolds number (dimensionless)
s	Specific entropy (Btu/lb/F)
T	F or R
T _p	Pinch temperature difference (F)
T _{pc}	Condenser pinch temperature difference (F)
T ₀	Condenser temperature
T _{B1}	Resource temperature (F)
ΔT _{sup}	Exhaust superheat (F)
T _{w1} , T _{w2}	Condenser cooling water inlet and exit temperatures (F)
U	Blade speed (ft/s)
U _c , U _d	Clean and dirty overall heat transfer coefficients (Btu/hr/ft ² /F)
v	Specific volume (ft ³ /lb)
W	Specific output (Btu/lb brine)
X _c	Weight concentration of NaCl in brine (ppm)

X_{CO2} Weight concentration of CO₂ in brine (lb/lb brine)

Greek Symbols

α	Coefficient in Starling equation of state and blade height to pitch diameter ratio, h/D _p (dimensionless)
η	Efficiency (dimensionless)
ξ	Turbine leaving loss coefficient (dimensionless)
ρ	Density (lb/ft ³)
σ	Stress (lb/in ²)
φ	Conversion factor in Starling equation of state and turbine flow coefficient (dimensionless)
ψ	Turbine blade loading coefficient (dimensionless)

Subscripts

c	Critical value
e	Property defined at turbine exhaust
g	Gas
is	Isentropic process
l	Liquid phase
r	Reduced property, i.e., absolute value divided by critical value
s	Saturation value, steam property or heat exchanger shell side property
s1	Value of primary flash in dual flash cycle
s2	Value at secondary flash in dual flash cycle
sup	Superheat
t	Heat exchanger tube-side property
v	Vapor

Superscript

o	Ideal gas property
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Section 1

INTRODUCTION

The heat stored in the earth's interior is without question one of the largest energy reserves known to man. However, much of this heat is inaccessible since it is stored either at too great a depth or at too low a temperature to be exploited economically. Of the energy that can be exploited much of that too is at a comparatively low temperature and can be considered useful only as a source of space or process heat. The remaining energy which is of sufficiently high grade to exploit for power generation comprises only a small fraction of the total, but nevertheless, if utilized could supply a significant portion of the Nation's future electric energy needs.

Up to now, geothermal exploitation has been confined to "hydrothermal" systems, in which heat is transferred from hot rock to reservoirs of steam or water circulating by natural convection to relatively shallow depths. The so-called steam-dominated systems, such as the Geysers in the United States and the Lardarello field in Italy, are commercially very attractive since the steam can be utilized directly with proven steam turbine technology, but they are comparatively rare, accounting for perhaps no more than 5 percent of the accessible resource worldwide. Hot water systems are more common. They vary widely in temperature with low to moderate temperature fields ranging from about 100 F to 400 F and high temperature fields reaching as high as 700 F.

Of the liquid-dominated fields known to exist, only the higher temperature systems, such as those at Cerro Prieto in Mexico and Wairakei in New Zealand, are currently being exploited on a large scale for power generation. Because of the high temperatures, the wells of such systems are self-flowing. As a result of the decrease in hydrostatic pressure, the liquid flash boils as it flows towards the surface. The buoyancy of the expanding steam forces the remaining water upward, increasing its velocity and ejecting residual water from the well. In all the fields currently in production, the steam is separated from the mixture at the wellhead and utilized in steam turbines. The water is either rejected at wellhead pressure, as at Cerro Prieto, or flashed at a lower pressure if dual pressure level turbines are employed.

The compatibility of steam turbines with these high temperature systems has been aided thus far by the fact that the resources are relatively clean, i.e., they contain moderate to low concentrations of dissolved salts and, what is perhaps more important if condensing steam turbines are to be employed, little or no noncondensable gas. Future development of geothermal energy is, however, dependent upon the ability to utilize resources which are in a less favorable form where steam turbines may not be able to operate effectively.

In this study, no attempt has been made to compare the relative merits of all the energy conversion cycles which are possible. Only the steam turbine cycle and a flash steam binary cycle in which the resource energy is transferred via steam to a secondary working fluid in a closed Rankine cycle were selected for analysis. Although the binary cycle analyzed may not exploit the resource as effectively as some of the cycles proposed, it can unlike these other cycles be implemented using proven technology and would therefore appear to have the best chance of immediate success. The study is restricted to the power island only. The objectives are to determine the performance and cost of the above cycles for the given resource conditions and to define any research needs that may be necessary for their implementation. The program was divided into the following four primary tasks:

- Task 1 -- Geothermal resource evaluation and recommendations of geothermal fluid conditions for use in analysis
- Task 2 -- Preliminary analysis of the energy conversion cycle and selection of the most promising cycles for the resource conditions selected in Task 1
- Task 3 -- Detailed analysis of the cycles selected in Task 2, including a technical-economic evaluation of the power island equipment
- Task 4 -- Recommendations for future development work

Section 2

GEOTHERMAL RESOURCE EVALUATION

In the initial phase of this study, a number of potential geothermal resources were reviewed and three specific brines were selected* for the preliminary cycle analysis. The brines originated in the Salton Sea region of California's Imperial Valley and their temperatures and compositions span a range that is of practical interest for power generation. One brine is from well 6-1 at East Mesa where the bottom hole temperature of 400 F is generally considered to be close to the lower practical limit for economic recovery. The other two brines are from Niland, some 30 miles northwest of East Mesa. They are from wells Sinclair No. 4 and Elnore No. 1 and have bottom hole temperatures of 500 F and 700 F, respectively. An analysis of each brine and typical well depths is given in Table 1.

Table 1
BRINE ANALYSES

	NILAND		EAST MESA
	2500' (500 F)	5000' (700 F)	Well 6-1 8000' (400 F)
Cl	137000	250000	13100
Na	56000	79000	8600
K	10000	33000	1445
Li	1600	600	147
Ca	22000	36000	1034
Ba	640	-	-
Sr	20	-	193
Mg	120	-	11
Mu	740	-	-
B	220	514	18
Fe	200	3073	20
Cu	2	-	-
Zn	300	-	-
Pb	45	-	-
F	14	-	-
SO ₄ ⁼	20	-	65
S	2	-	-
SiO ₂	500	800	340
HCO ₃ ⁻	-	-	231
Si(OH) ₄	-	-	-
TDS	220000	-	24800
pH	-	5.2	-
SG	-	1.241	-
Noncondensable gas (% of Brine Weight)	5%	-	-

*Selection of brines for continuing study was made by the ERDA Program Manager, Division of Geothermal Energy

Unfortunately, there is at present relatively little field data on the operating characteristics of the wells at Niland. None have yet flowed for extended periods and thus, very little information exists on how wellhead conditions vary with productivity. Preliminary tests indicate however that for the 500 F and 700 F resources, wellhead steam conditions are approximately 370 F at 150 psia and 415 F at 225 psia when flow rates are the order of 400,000 and 600,000 lb/hr, respectively. The superheat at the wellhead is simply due to high salinity. Thermal characteristics of the Niland brine have not yet been fully investigated either, and at the time of performing the preliminary analysis of this report, temperature/enthalpy relationships had not been accurately determined. At East Mesa the flow rate/wellhead condition relationship is better understood. Tests on well 6-1 show wellhead temperatures and pressures to be on the order of 300 F and 80 psia when the flow rate is 45,000 lb/hour. Reducing the pressure to 25 psia decreased the temperature to 250 F and increased the flow rate to 100,000 lb/hour. Because the salinity is moderately low, the specific heat of the East Mesa brine is very close to 1.0.

Despite the high temperatures, the Niland resource is perhaps one of the most difficult to recover. Total dissolved solids, which consist mostly of sodium chloride, exceed by a factor of ten those in the moderately saline brine at East Mesa and in the high temperature brine at Cerro Prieto, 50 miles due south. The gas in the Niland brine is mostly CO₂ and can amount to between 3 and 5 percent of the brine weight. This translates to between 12 and 20 percent of the total steam flow in a dual flash system, which precludes the use of steam turbines with the resource, since the CO₂ extraction pump work required to maintain a reasonable condenser vacuum would be of the same order or even exceed the extra power derived from the turbine's subatmospheric exhaust. Furthermore the deleterious effect of the diffusion of CO₂ on condensing heat transfer coefficients would only serve to increase an already large assembly in the condenser.

The major obstacle to the exploitation of the Niland resource is the formation of scale due primarily to dissolved silica. Its solubility is highly sensitive to temperature and although it is never present in large concentrations (about 500 ppm), large quantities accumulate because of the large mass of brine that must be handled in the low efficiency cycles. If steam is used instead of brine for the primary heat exchange process, the problem of silica is made more manageable by confining the scale formation to the flash tank.

This scaling problem is in fact so severe at Niland that in tests performed by the San Diego Gas and Electric Company in 1973 on a heat exchanger of conventional tube/shell design employing steam flashed from Niland brine, there was so much silica carried over with the steam that complete fouling occurred in only 100 hours. An analysis of the scale taken from this test is shown in Table 2. Subsequent development work by this Company on improved separator designs and the introduction of a scrubbing device into the steam

line reduced the carry-over of solids to just 10 to 20 ppm and satisfactory operation of the heat exchanger was extended to 3200 hours before fouling began to limit performance.

Table 2
ANALYSIS OF SCALE FROM
HEAT EXCHANGER TEST AT NILAND

	Percent
SiO ₂	72.96
Fe ₂ O ₃	3.86
MnO ₂	2.37
CuO	2.25
ZnO	7.59
Mg	1.20
Ca	5.30
Ag	0.12
Cr	0.80
Ni	0.40
Pb	2.20

Section 3

PRELIMINARY ANALYSIS

A. INTRODUCTION

Because its operation is relatively insensitive to noncondensable gas, the binary cycle in which the resource energy is transferred to a secondary working fluid in a closed Rankine cycle has received considerable attention as a means of exploiting the Niland resource. This binary system has several advantages. First, the power cycle can be implemented using existing technology. Second, the CO₂ can easily be vented under pressure from the heat exchanger without appreciable loss of vapor and without the need for extraction pump work. Third, it is possible with judicious selection of the secondary working fluid to make the turbine/condenser assembly and other closed cycle components fairly compact. Fourth, again with proper selection of the working fluid, the turbine efficiency can be improved and possible erosion damage avoided by eliminating condensation from the expansion process.

The major disadvantage with this system is, of course, that the overall complexity of the cycle is increased by the need for a heat exchanger and circulating pump. The cycle is in reality further handicapped by the fact that potential scaling problems at present preclude the use of brine in direct contact with a metallic heat exchange surface. The brine must first be flashed to steam and steam utilized to transfer energy to the secondary working fluid and a heat exchanger is required for each stage of flash.

In the following analysis, the binary cycle has been analyzed in some detail. The performance of several different working fluids are compared for the various resource conditions at Niland and East Mesa. Since the brine at East Mesa contains very little noncondensable gas, the use of steam turbines is feasible, and in this case a meaningful comparison is made between the performance of the steam turbine cycle and the binary cycle.

B. SELECTION OF BINARY CYCLE WORKING FLUIDS

A total of ten compounds were selected as possible working fluids for the geothermal binary cycle. With the exception of ammonia, all are organic compounds—six hydrocarbons and three fluorocarbons—and are listed in Table 3. The fluorocarbons and ammonia were identified as possible candidates in Reference 1. However since supercritical cycles, because of their superior performance characteristics, were selected where possible for analysis, all fluids were selected primarily on a basis of critical temperature. This was required to be sufficiently below the available resource temperature to permit operation at supercritical pressure.

Table 3
CANDIDATE WORKING FLUIDS FOR THE BINARY CYCLE

1	Propane (C ₃ H ₈)	(R-290)
2	Propylene (C ₃ H ₆)	
3	Isobutane (C ₄ H ₁₀)	(R-600a)
4	N-butane (C ₄ H ₁₀)	(R-600)
5	Isopentane (C ₅ H ₁₂)	
6	N-pentane (C ₅ H ₁₂)	
7	Chlorodifluoromethane (CHClF ₂)	(R-22)
8	Difluoromethane ((CH ₂ F) ₂)	(R-32)
9	Chloropentafluoroethane (C ₂ ClF ₅)	(R-115)
10	Ammonia (NH ₃)	(R-717)

C. DERIVATION OF THERMODYNAMIC PROPERTIES

All thermodynamic properties of the various working fluids analyzed in this study were generated from an equation of state and the following fundamental thermodynamic identities which relate thermodynamic properties to P-v-T data:

$$dh = \left\{ d(Pv) + T \left(\frac{\partial P}{\partial T} \right)_v - P \right\}_T dv + \left\{ C_p dT \right\}_P \quad (1)$$

$$ds = \left\{ \left(\frac{\partial P}{\partial T} \right)_v dv \right\}_T + \left\{ C_p \frac{dT}{T} \right\}_P \quad (2)$$

For an ideal or infinitely dilute gas, the specific heat, C_p , is a function only of temperature since molecular interactions have negligible effect on the energy of the gas. This important fact greatly assists the estimating of changes in thermodynamic properties between equilibrium states. For example, if Equation 1 is integrated along an isotherm from state P, T to the corresponding ideal gas state $P=0$ ($v=\infty$), $T=T$, then,

$$H-H^0 = H(P, T) - H(0, T) = Pv - RT + \int_{\infty}^v \left\{ T \left(\frac{\partial P}{\partial T} \right)_v - P \right\} dv \quad (3)$$

This quantity, often referred to as the enthalpy departure, is the enthalpy difference between the real gas and the ideal gas at the same temperature. If the integration is repeated from states P_2, v_2 , and P_1, v_1 , to their respective corresponding ideal gas states, then the enthalpy difference between states 1 and 2 becomes

$$H(P_2, T_2) - H(P_1, T_1) = H(0, T_2) - H(0, T_1) + (P_2 v_2 - P_1 v_1) - R(T_2 - T_1) + \int_{\infty}^{v_2} \left\{ T \left(\frac{\partial P}{\partial T} \right)_v - P \right\}_{T_2} dv - \int_{\infty}^{v_1} \left\{ T \left(\frac{\partial P}{\partial T} \right)_v - P \right\} dv \quad (4)$$

where the ideal gas enthalpy difference is

$$H(0, T_2) - H(0, T_1) = \int_{T_1}^{T_2} C_p dT \quad (5)$$

For most fluids, the ideal gas specific heat is known over a wide temperature range and can usually be expressed conveniently in polynomial form.

The corresponding expression for the specific entropy change is derived in essentially the same manner as the above. In this case, we define the entropy departure as the difference in the entropy of the compound at the temperature-pressure condition of interest and the entropy of the compound in the ideal gas state at the same temperature and unit pressure.* The entropy departure is related to the equation of state by the following equation:

$$s - s^0 = s(P, T) - s(1, T) = -R \log_e \frac{RT}{v} + \int_{\infty}^v \left\{ \left(\frac{\partial P}{\partial T} \right)_v - \frac{R}{v} \right\} dv \quad (6)$$

The resulting expression for the change in entropy between real gas states P_1, v_1 , and P_2, v_2 therefore becomes

$$s(P_2, T_2) - s(P_1, T_1) = \int_{T_1}^{T_2} C_p \frac{dT}{T} - R \log_e \left\{ \frac{T}{v} \right\}_{T_1, v_1}^{T_2, v_2} + \int_{v_1, T_1}^{v_2, T_2} \left\{ \left(\frac{\partial P}{\partial T} \right)_v - \frac{R}{v} \right\} dv \quad (7)$$

a. EQUATIONS OF STATE

It was found necessary in this analysis to employ two different equations of state since there is insufficient empirical data available for any one equation to be able to predict the behavior of all organic compounds of interest here. The first equation of state, the Starling equation (Ref. 2), was used exclusively for the light hydrocarbon compounds for which it was developed for the petrochemical industry. The second equation, the Martin-Hou equation (Ref. 3), was used for the fluorocarbon

*Because the logarithm of pressure appears in the expression for the change in entropy, zero pressure is avoided. However the error incurred by substituting unit pressure is negligible.

compounds and ammonia. It was also used for the isolated case of one hydrocarbon (isobutane) to compare results from the two equations. It will be shown later that the Martin-Hou equation suffers serious limitations in its ability to predict density roots in the compressed liquid region and as a result gives less accurate performance predictions than the Starling equation.

1. The Starling Equation of State

Like all equations of state, the Starling equation expresses the pressure as a function of temperature and density, i.e.,

$$P = \rho RT + \left(B_0 RT - A_0 - \frac{C_0}{T^2} + \frac{D_0}{T^3} - \frac{E_0}{T^4} \right) + \left(b RT - a - \frac{d}{T} \right) \rho^3 \quad (8)$$

$$+ \alpha \left(a + \frac{d}{T} \right) \rho^6 + c \frac{\rho^3}{T^2} (1 + \gamma \rho^2) \exp(-\gamma \rho^2)$$

where the units of pressure, temperature, and density are psia, °R, and lb-moles/ft³, respectively. R is the gas constant and has the value 10.7335 psia ft³/lb-mole/°R.

For a given temperature-pressure condition, the density must be calculated numerically by an iterative trial and error procedure. The Newton iteration scheme was found to be satisfactory although judicious judgment was required with the initial estimate since the equation has several density roots. Only the smallest and largest roots have physical significance and correspond to the vapor and liquid densities, respectively.

The equation of state expression for the enthalpy departure is obtained by substituting Equation 8 into Equation 3, i.e.,

$$\phi(H-H^0) = \left(B_0 RT - 2A_0 - \frac{4C_0}{T^2} + \frac{5D_0}{T^3} - \frac{6E_0}{T^4} \right) \rho + \frac{1}{2} \left(2b RT - 3a - \frac{4d}{T} \right) \rho^2 \quad (9)$$

$$+ \frac{1}{5} \alpha \left(6a + \frac{7d}{T} \right) \rho^5 + \frac{c}{\gamma T^2} \left[3 - \left(3 + \frac{1}{2} \gamma \rho^2 - \gamma^2 \rho^4 \right) \exp(-\gamma \rho^2) \right]$$

where ϕ is the conversion factor from Btu/lb to psia ft³/lb-mole. Similarly the equation of state expression for the entropy departure becomes

$$\phi(s-s^0) = -R \log_e (\rho RT) - \left(B_0 R + \frac{2C_0}{T^3} - \frac{3D_0}{T^4} + \frac{4E_0}{T^5} \right) \rho - \frac{1}{2} \left(b R + \frac{d}{T^2} \right) \rho^2$$

$$+ \frac{\alpha \rho^5}{5T^2} + \frac{2c}{\gamma T^3} \left[1 - \left(1 + \frac{1}{2} \gamma \rho^2 \right) \exp(-\gamma \rho^2) \right] \quad (10)$$

The *fugacity* of the compound may be expressed in terms of the enthalpy and entropy departures by the thermodynamic relation:

$$RT \log_e f = [(H-H^0) - T(s-s^0)] \phi \quad (11)$$

On substituting the equations for enthalpy and entropy departures, this expression becomes

$$RT \log_e f = RT \log_e (\rho RT) + 2 \left(B_0 RT - A_0 - \frac{C_0}{T^2} + \frac{D_0}{T^3} - \frac{E_0}{T^4} \right) \rho \\ + \frac{3}{2} \left(bRT - a - \frac{d}{T} \right) \rho^2 + \frac{6\alpha}{5} \left(a + \frac{d}{T} \right) \rho^5 + \frac{c}{\gamma T^2} \left[1 - \left(1 - \frac{1}{2} \gamma \rho^2 - \gamma^2 \rho^4 \right) \exp(-\gamma \rho^2) \right] \quad (12)$$

Again, the density value used in this expression is computed from Equation 8.

The vapor pressure P_s at a given temperature is obtained from the simultaneous solution of the following equations, which describe liquid-vapor equilibrium, i.e.,

$$P_l = P_v = P_s \quad (13)$$

$$f_l = f_v \quad (14)$$

where the subscripts l and v refer to liquid and vapor, respectively. Again a trial and error iterative procedure is required to solve this equation set. First an estimate is made of the vapor pressure and is set equal to the liquid and vapor pressures, respectively. The liquid and vapor density roots are then obtained from the equation of state and substituted into Equation 12 to calculate the liquid and vapor fugacities. If Equation 14 is not satisfied, the initial estimate of vapor pressure is adjusted and the procedure repeated until agreement between fugacities is sufficiently close. The saturated liquid and vapor enthalpies and entropies are calculated simply by using in Equations 9 and 10, the saturated vapor and liquid densities determined in calculating the vapor pressure. The *enthalpies and entropies of vaporization* are then calculated as the difference between the saturated vapor and liquid property values. The ideal gas enthalpy difference and the ideal gas entropy were both obtained from Reference 4. The eleven parameters appearing in the equation of state are listed along with critical values of pressure, temperature, and specific volume for the six light hydrocarbon compounds listed in Table 4.

2. The Martin-Hou Equation of State

This equation, written in terms of reduced properties, has the following form

$$P_r = \frac{1}{Z_c} \left\{ \frac{T_r}{v_r - b} + \frac{f_1(T_r)}{(v_r - b)^2} + \frac{f_2 T_r}{(v_r - b)^3} + \frac{f_3(T_r)}{(v_r - b)^4} + \frac{f_4(T_r)}{(v_r - b)^5} \right. \\ \left. + f_5(T_r) e^{-av_r} + f_6(T_r) e^{-2av_r} \right\} \quad (15)$$

Table 4

**CRITICAL VALUES AND COEFFICIENTS FOR STARLING EQUATION
FOR THE HYDROCARBONS**

	COMPOUND					
	Propylene C ₃ H ₈	Propane C ₃ H ₈	N-Butane C ₄ H ₁₀	Isobutane C ₄ H ₁₀	N-Pentane C ₅ H ₁₂	Isopentane C ₅ H ₁₂
MW	42.046	44.062	58.121	58.121	72.147	72.147
P _c (psia)	670.27	617.4	550.7	529.06	489.5	494.7
T _c (°F)	197.2	206.13	305.62	275.0	385.92	370.1
V _c (ft/lb)	0.06890	0.07278	0.07028	0.07244	0.06906	0.06838
A _o	6051.36	18634.7	32544.7	37264.0	51108.2	35742.0
B _o	0.114457	0.964762	1.56588	1.87890	2.44417	1.27752
C _o	974762E4	796178E4	137436E5	101413E5	223931E5	228430E5
D _o	705921E6	453708E6	333159E6	853176E6	101769E7	142115E7
E _o	341250E8	256053E8	230902E7	840860E8	390860E8	241326E8
a	81880.4	40066.4	71181.8	47990.7	162185	204344
b	7.64114	5.46248	9.14066	8.58663	16.6070	19.8384
c	294141E5	274461E5	700044E5	406763E5	135286E6	129083E6
d	54193.5E2	150520E2	364238E2	216863E2	388521E2	349220E2
α	1.36532	2.01402	4.00985	4.23987	7.06702	6.16154
γ	4.07919	4.56182	7.54122	7.11486	11.8593	11.7384

Z_c is the critical compressibility factor, and the temperature functions $f_i(T_r)$ are expressed

$$f_i(T_r) = f_i(1) + B_i(T_r - 1) + C_i(e^{-KT_r} - e^{-K}) \quad (16)$$

The expression for the enthalpy departure is obtained by substituting Equation 15 into Equation 3, or into Equation 4, to obtain the enthalpy difference between real gas states P_2, v_2 and P_1, v_1 , i.e.,

$$\begin{aligned}
\frac{H(P_2, T_2) - H(P_1, T_1)}{RT_c} = & \int_{T_{r1}}^{T_{r2}} \frac{C_p^*(0, T_r)}{R} dT_r \\
& + \left\{ Z_c P_r v_r - T_r + \sum_{i=1}^4 \frac{f_i(1) - B_i + C_i \left[(1+KT_r) e^{-KT_r} - e^{-K} \right]}{i (v_r - b)^i} \right. \\
& + \frac{f_5(1) - B_5 + C_5 \left[(1+KT_r) e^{-KT_r} - e^{-K} \right]}{ae^{av_r}} \\
& \left. + \frac{f_6(1) - B_6 + C_6 \left[(1+KT_r) e^{-KT_r} - e^{-K} \right]}{2ae^{2av_r}} \right\}_{T_{r1}, v_{r1}}^{T_{r2}, v_{r2}}
\end{aligned} \quad (17)$$

Similarly the entropy change between real gas stages P_1, v_1 , and P_2, v_2 is obtained from Equation 7, i.e.,

$$\begin{aligned}
\frac{s(P_2, T_2) - s(P_1, T_1)}{R} = & \int_{T_{r1}}^{T_{r2}} \frac{C_p^* dT}{R T} - \log_e \left[\frac{P_r}{P_r} \right]_{P_{r1}}^{P_{r2}} \\
& + \left\{ \log_e \frac{T_r}{(v_r - b)^{P_r Z_c}} + \sum_{i=1}^4 \frac{B_i - C_i K e^{-KT_r}}{i (v_r - b)^i} \right. \\
& \left. + \frac{B_5 - C_5 K e^{-KT_r}}{ae^{av_r}} + \frac{B_6 - C_6 K e^{-KT_r}}{2ae^{2av_r}} \right\}_{T_{r1}, v_{r1}}^{T_{r2}, v_{r2}}
\end{aligned} \quad (18)$$

C_p^*/R is the reduced ideal gas heat capacity and is expressed in a power series expansion of reduced temperature as

$$C_p^*/R = C_0^* + C_1^* T_r + C_2^* T_r^2 + C_3^* T_r^3$$

Other complementary equations include the expression for vapor pressure p_{sat} ,

$$\log_e P_r^{sat} = A_r - B_r/T_r - C_r \log_e T_r - D_r T_r + E_r T_r^2,$$

and the saturated liquid density $\rho_{l,r}^{sat}$,

$$\rho_{l,r}^{\text{sat}} = \frac{\rho_l^{\text{sat}}}{\rho_c} = 1 + A(1-T_r)^{1/3} + B(1-T_r)^{2/3} + C(1-T_r) + D(1-T_r)^{4/3}$$

The latent heat of vaporization ΔH_{vap} is estimated using the Clausius-Clapeyron equation in reduced form, i.e.,

$$\frac{\Delta H_{\text{vap}}}{RT_c} = \frac{\Delta H_{\text{vap}}}{RT_c} = \frac{h_{fg}}{RT_c} = Z_c T_r (v_g - v_l)^{\text{sat}} \frac{dP_r^{\text{sat}}}{dT_r}$$

Values of the various coefficients and critical properties are listed for the relevant compounds studied in Table 5.

D. PRELIMINARY CYCLE ANALYSIS

a. BRINE/BINARY CYCLE

For the purpose of comparing the relative merits of the different working fluids, it was felt that the relatively simple closed cycle shown in Figure 1, in which brine flows directly through the primary heat exchanger, would be adequate as a starting point for the preliminary analysis. Although the cycle may be somewhat unrealistic in view of the potential scaling problem from silica, it does serve to indicate the maximum work that can be obtained from a closed cycle with a brine delivered in liquid form at a given temperature at the wellhead.

In the following section supercritical cycles, because of their superior performance characteristics, were selected where possible for analysis. A prerequisite for supercritical operation is, of course, that the brine inlet temperature minus some reasonable temperature difference for heat exchange be greater than the critical temperature of the working fluid. A typical temperature/enthalpy diagram for the primary heat exchanger of a supercritical cycle is shown in Figure 2. Throughout the analysis, the cycle performance was calculated with the specific output, in terms of brine flow, optimized since this parameter has the greatest significance in terms of utilizing the geothermal resource. Such an optimization requires the heat exchanger pinchpoint temperature difference to be free to "float" since only the brine inlet temperature and organic fluid inlet temperature have known values. For maximum cycle output, the pinchpoint will occur at some position intermediate between the entrance and exit as shown by the heating and cooling paths of the organic fluid and brine depicted by the solid lines in Figure 2. Should the pinchpoint occur at the brine inlet, the high turbine inlet temperature would tend to maximize the thermal efficiency but it would also underutilize the geothermal resource since the brine is discharged at the relatively high temperature T_{B2}'' . The net result is a less than optimum cycle output. Conversely a pinchpoint located at the brine outlet would produce the heating curve H_2T_2' and although it would maximize the heat recovered from the brine since the brine discharge tempera-

Table 5

**CRITICAL VALUES AND COEFFICIENTS FOR THE MARTIN EQUATION
FOR THE FLUOROCARBONS AND AMMONIA**

	COMPOUND			
	R-22	R-32	R-115	Ammonia
MW	86.48	52.03	154.5	17.03
P _c (psia)	721.9	845.6	458.0	1635.7
T _c (°F)	204.8	173.1	175.9	270.1
V _c (ft ³ /lb)	0.03053	0.03729	0.02613	0.06803
z _c	0.267	0.242	0.271	0.242
C ₀ *	2.6073	4.2778	2.4760	4.3032
C ₁ *	5.6582	3.2980	14.0108	-1.8411
C ₂ *	-0.5479	0	-3.6790	3.1995
C ₃ *	-0.1943	0	0	-0.9830
A	1.6677	1.4083	2.4504	1.5489
B	1.1218	3.3657	-3.3572	1.9711
C	-0.6805	-5.0893	6.5815	-1.5957
D	0.6249	3.6366	-3.297	1.0322
Ar	20.5140	15.3618	4.4096	7.3672
Br	7.2003	9.1357	12.5996	10.8839
Cr	-6.8419	0	13.9492	8.5674
Dr	19.4130	10.4699	-8.1830	-2.4647
Er	6.0993	4.2410	0	1.0520
a	16.73381	14.33333	16.733805	16.73381
b	0.065520	0.018333	0.233815	0.065520
K	4.2	6.42	5.48	4.2
f ₁ (1)	-1.356568	-1.545953	-1.476552	-1.356568
f ₂ (1)	0.721958	1.022201	1.092186	0.721958
f ₃ (1)	-0.0366074	-0.1891483	-0.405404	-0.0366074
f ₄ (1)	-0.0617381	-0.0458976	0.0653873	-0.0617381
f ₅ (1)	9.325812 x 10 ³	1.741376 x 10 ³	0	9.325812 x 10 ³
f ₆ (1)	0	1.803382 x 10 ⁵	0	0
B ₁	0.635479	1.658608	0.831589	0.635479
B ₂	0.659672	-3.768077	-0.362615	0.659672
B ₃	-1.021552	4.924337	0	-1.021552
B ₄	0.497061	-1.994371	0.0556471	0.497061
B ₅	-4.11421 x 10 ⁴	2.56467 x 10 ⁴	0	-4.11421 x 10 ⁴
B ₆	0	-7.22990 x 10 ⁶	0	0
C ₁	17.506396	-81.254057	-26.570529	17.506396
C ₂	19.310496	-245.44040	23.639653	19.310496
C ₃	0	647.520530	0	0
C ₄	-2.577068	-323.02139	-1.313044	-2.577068
C ₅	0	4.294690 x 10 ⁶	0	0
C ₆	0	-9.571300 x 10 ⁸	0	0

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ture has now been reduced to T_{B2}' , it would also produce a low turbine inlet temperature and hence a low turbine efficiency. Again a less than optimum cycle output would result.

Clearly this technique of optimizing output requires a detailed knowledge of the temperature/specific enthalpy relationship at all points on the heating curve from the feed pump exit to the turbine inlet for each organic compound studied. With the Starling equation of state this could be accomplished readily. The Martin-Hou equation, however, was found to be less reliable. At supercritical pressures, for example, it will frequently predict densities of compressed liquid that are significantly less than the saturation values and, in fact, in some instances it could not predict density roots at all. For the compounds studied, difficulties in obtaining reliable density roots always occurred in the range extending from the condenser temperature to near the critical temperature. The problem was finally resolved by generating approximate curves, using the saturation line as a datum, which extended from the compressed liquid region to points in the gas state where the equation of state was known to be valid.

The various assumptions made in calculating the performance of each cycle are listed in Table 6. These values were chosen arbitrarily but were considered adequate for the purpose of comparing the relative merits of the working fluids.

Table 6

LIST OF ASSUMPTIONS FOR BRINE BINARY CYCLE

Condenser Temperature	100 F
Pinchpoint Temperature	25 F
Primary Heat Exchanger $\Delta p/p$	0.10
Condenser $\Delta p/p$	0.05
Turbine Efficiency	0.85
Feed Pump Efficiency	0.85
Brine Specific Heat	1.0

Results of the cycle calculations for each of the organic compounds studied are summarized in the performance plots in Figures 3 through 13, in which net specific output is plotted against efficiency for various cycle pressure ratios and brine inlet conditions. A summary of these results is given in Figure 14, which shows the maximum specific output for a given resource temperature plotted against resource temperature. The use of fluorocarbons as working fluids at the high turbine inlet temperatures achieved when brine is utilized directly is somewhat unrealistic. These fluids may become unstable at temperatures much above 250 F especially in the presence of oil, steel, or copper, which act as catalysts for their chemical decomposition. They were included here merely for comparison with their performance in the lower temperature flashed steam binary cycle analyzed in the next section.

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Figure 3. Performance Characteristics of R-22 Brine Cycle

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Figure 4. Performance Characteristics of R-32 Brine Cycle

Figure 5. Performance Characteristics of R-115 (Chloropentafluoroethane)/Brine Cycle

Figure 6. Performance Characteristics of Ammonia/Brine Cycle

Figure 7. Performance Characteristics of Propylene/Brine Cycle

Figure 8. Performance Characteristics of Propane/Brine Cycle

Figure 9. Performance Characteristics of Isobutane/Brine Cycle (Martin-Hou Equation of State)

Figure 10. Performance Characteristics of Isobutane/Brine Cycle

Figure 11. Performance Characteristics of N-Butane/Brine Cycle

Figure 12. Performance Characteristics of Iso-Pentane/Brine Cycle

Figure 13. Performance Characteristics of N-Pentane/Brine Cycle

Figure 14. Maximum Specific Output vs Brine Inlet Temperature for Candidate Working Fluids in Brine/Binary Cycle

At the higher temperatures, N-pentane and isopentane utilize the resource most effectively. At lower temperatures they do not perform so well since their high critical pressures prohibit the use of supercritical cycles and output must be compromised in order to prevent condensation from occurring during the expansion process. The effect on output of such a compromise is shown for isopentane by the dashed line in Figure 14. With most cycles, in fact, it is the norm rather than the exception that some output must be sacrificed either to prevent condensation or, if the expansion is already dry, to minimize the degree of superheat in the turbine exhaust. Fortunately the performance characteristics of most of the compounds studied, particularly the hydrocarbons, permit a fairly wide variation in operating pressure without affecting output appreciably.

At the lower resource temperatures, the spread between the performance curves is considerably reduced. Here isobutane, propane, and propylene appear to be potentially the best working fluids. The two curves given for isobutane are the respective predictions of the two equations of state. At high brine temperatures, there is excellent agreement although there is some discrepancy at lower temperatures due perhaps to errors in the approximate curves fitted to complement the Martin-Hou equation of state. Ammonia appears to be attractive only at turbine inlet temperatures which are unlikely to be achieved, particularly if steam is used instead of brine for the primary heat exchange process. Ammonia does, however, have relatively good heat transfer characteristics, and it will produce the most compact turbine for a given power rating of all the compounds studied as shown in Table 7.

The relative exhaust area, A_{er} , of the last stage is calculated from the continuity equation, i.e.,

$$A_e = \frac{\dot{m}}{V_e} v_e$$

If the exhaust velocity, V_e , is assumed to be some fixed fraction, c , of the exhaust sound velocity, and if the mass flow rate is replaced by the power rating divided by the net work per pound of working fluid, the exhaust area can be written

$$A_e = \frac{P v_e}{c a \Delta H_{NET}}$$

For a given power rating then, the *relative* exhaust area can be expressed

$$A_{er} = \frac{K v_e}{a \Delta H_{NET}}$$

In the comparison of exhaust sizes in Table 7, the value of K was chosen so as to make A_{er} for ammonia equal to 1.0.

Table 7

RELATIVE TURBINE EXHAUST SIZE (A_{er})

Compound	A_{er}
Ammonia	1.0
R-32	1.5
Propylene	2.0
Propane	2.5
R-22	3.0
R-115	5.0
Isobutane	6.0
N-butane	8.0
Isopentane	19.0
N-pentane	24.0

b. THE DUAL FLASH/BINARY CYCLE

In view of the potential scaling problem from silica, it is unlikely that heat exchangers of conventional tube/shell design will ever function successfully in direct communication with the brine in many areas in the Imperial Valley unless the brine chemistry is artificially modified. Moreover, current research on fluidized bed and direct contact heat exchangers which utilize brine directly is still in the preliminary stage and has not yet addressed the problem of dealing with brines of the type encountered in the Imperial Valley. An assessment of the potential of these two devices is beyond the scope of this study; the analysis in this section has, therefore, been based on the assumption that heat exchangers can function successfully only when steam is used for the primary heat exchange process. It has also been assumed, somewhat arbitrarily at this point, that from an economic viewpoint the optimum number of flash stages is two.

A schematic of an organic cycle in which steam is flashed in two stages is shown in Figure 15. Conceptually, subcooled condensate leaving the higher temperature heat exchanger is injected into the second heat exchanger at a point where it can mix with saturated liquid condensate at the same temperature. A temperature/enthalpy diagram of this process for a supercritical organic cycle is shown in Figure 16.

Thermodynamically this dual flash system is fairly complex. It has in fact 4 degrees of freedom—2 more than in the case where the brine is utilized directly. Because of this added complexity, the output was not truly optimized on a basis of specific output

as in the brine/binary cycle. Instead, the problem was made more tractable by fixing the conditions at entrance and exit of the overall heat exchange process. A primary flash temperature was assumed, and the temperature differences between fluids at inlet and exit were both maintained at the minimum pinchpoint value. In a dual flash system, as in any multiple flash arrangement, the second flash temperature is not independent of the first. The energy balance in the primary exchanger

$$\dot{m}_{s1} [H_{fg1} + C_p (T_{s1} - T_{s2})] = \dot{m} (H_B - H_C) \quad (19)$$

must be compatible with that in the second exchanger, i.e.,

$$\dot{m}_{s2} H_{fg2} + (\dot{m}_{s1} + \dot{m}_{s2}) C_p (T_{s2} - T_{s3}) = \dot{m} (H_C - H_A) \quad (20)$$

If the temperature difference between the two fluids at the junction between the two heat exchangers has some specified minimum value, then the only unknown is the secondary flash temperature T_{s2} . The solution is obtained by an iterative procedure.

In the cycle analysis, the temperature difference T_{pC} between the two fluids at this intermediate point and the temperature differences T_{pA} and T_{pB} at the entrance and exit of the overall heat transfer process were all assumed to be 25 F. The primary flash temperature T_{s1} was assumed initially to be the same as that required to yield maximum output in a steam turbine cycle. Near optimum output is obtained from the latter when the temperature difference between source and sink is divided by the number of stages plus one and the intermediate temperatures are chosen for each stage of flash. Thus for a dual flash system:

$$T_{s1} = T_0 + 2/3 (T_{B1} - T_0) \quad (21)$$

and

$$T_{s2} = T_0 + 1/3 (T_{B1} - T_0) \quad (22)$$

The other assumptions used in calculating cycle performance were the same as in the direct brine cycle listed in Table 6.

As mentioned previously in the discussion on resource evaluation, preliminary tests at Niland indicate that wellhead conditions for the 500 F and 700 F resources will be 370 F at 150 psia and 415 F at 225 psia when the flow rates are respectively 400,000 lb/hr and 620,000 lb/hr. The superheat at the wellhead is simply a consequence of the brine salinity. Whether or not the above are optimum conditions for well productivity, no one yet knows. However, the wellhead temperature for the 500 F brine is very close to the recommended value for the primary flash although the corresponding

temperature for the 700 F brine is some 85 F less than the recommended value of 500 F. In this case, the recommended temperature was ignored and the primary flash was assumed to occur at 415 F. Control of the flow from the 400 F resource was assumed to be sufficiently flexible that the recommended primary flash temperature of 300 F could readily be achieved.

Results of the cycle calculations for the organic compounds studied in this section are summarized in the performance plots of Figures 17 through 22. Net specific output is again plotted against efficiency for a number of cycle pressure ratios and different resource temperatures. R-22 and ammonia were not included in this analysis because of their relatively poor showing in the direct brine cycle. A summary of the results is given in Figure 23, in which the maximum specific output for a given resource temperature is plotted against the resource temperature in the range 400 F to 500 F.

For the 500 F resource, the relatively high critical temperatures of isopentane and normal pentane again prohibit their use at supercritical pressures. Consequently output must again be sacrificed in order to produce a workable cycle. The net result for isopentane is that its performance is reduced to a point intermediate between that of isobutane and propane. R-32 which had marginally superior performance compared with isobutane in the direct brine cycle appears to have lost its advantage in the dual flash cycle. The reason for this reversal is not clear although it could be due to approximations made to generate the curves in the direct brine cycle which describe properties in the compressed liquid region at temperatures below the critical point. Interestingly enough, the state point of the freon at the second flash was at a sufficiently high temperature in the dual flash cycle that the Martin-Hou equation was able to describe fluid properties adequately and no curve fitting was necessary. The same was true for refrigerant R-115.

Because of their potentially poor stability, the freons are limited to use with the 400 F resource but even at this temperature neither one appears to offer any advantage over the hydrocarbons except perhaps for compactness of the turbomachinery and other closed cycle components.

The most versatile fluid in the 400 F to 500 F temperature range would seem to be isobutane. At the lower reservoir temperature it must operate at subcritical pressure; nevertheless it outperforms the two freons and propylene and propane, all of which function at supercritical pressure. Because maximum operating temperatures in the dual flash system are considerably lower than in the direct brine cycle, operating pressures are also lower. Again it is possible to have a wide variation in turbine inlet pressure without appreciable effect on the output. With isobutane, for example, operating on the 500 F resource, maximum output is obtained when the inlet pressure is 800 psia, but a reduction to

Figure 17. Performance Characteristics of R-32/2-Stage Flash Cycle

Figure 18. Performance Characteristics of R-115/2-Stage Flash Cycle

Figure 19. Performance Characteristics of Propylene/2-Stage Flash Cycle

Figure 20. Performance Characteristics of Propane/2-Stage Flash Cycle

Figure 21. Performance Characteristics of Isobutane/2-Stage Flash Cycle

Figure 22. Performance Characteristics of N-Butane/2-Stage Flash Cycle

Figure 23. Maximum Specific Output vs Bottom Hole Brine Temperature for Candidate Working Fluids in Dual Flash/Binary Cycle

650 psia or an increase to 1250 psia will reduce the output by less than 3%. However, at the condition of maximum output there is considerable superheat in the turbine exhaust as can be seen from the expansion lines in the pressure-enthalpy diagram of Figure 24. In this case, by increasing the pressure ratio, a relatively large reduction in superheat can be achieved with only a small reduction in output. If the expansion process is to remain dry, the maximum allowable turbine inlet pressure is about 1000 psia. By operating at this pressure the superheat will be reduced from 71 F to 38 F and the loss in output will be less than 1%. When isobutane is coupled with the 400 F resource, the operating pressure must be reduced to about 500 psia. Clearly if condensation in the expansion process is to be avoided, some exhaust superheat is inevitable. However, minimizing the degree of superheat will have an appreciable effect on condenser size.

The operating pressures and temperatures quoted thus far apply only to cases where the primary flash temperature is described by Equation 21. An example is given in Figure 25, again for the isobutane cycle, of how changes in this temperature affect the output. The resource temperature is 500 F and the numbers on each curve refer to the turbine inlet pressure in psia. In this case, the output approaches a maximum when the primary flash temperature is approximately midway between the resource temperature and the sink. Because the flash temperature—hence the turbine inlet temperature—is reduced, so too is the operating pressure.

Figure 24. Pressure/Enthalpy Diagram for Isobutane/2-Stage Flash Cycle

Figure 25. Effect of Primary Flash Temperature on Performance of Isobutane/2-Stage Flash Cycle

For maximum output this is about 510 psia. Fortuitously, it is also the pressure at which the expansion remains dry and there is a minimum of superheat in the turbine exhaust. Because operating pressures are reduced and the turbine efficiency is assumed to remain constant, the exhaust superheat is also reduced, as shown in Figure 26. The reduction in efficiency is simply a consequence of the lower turbine inlet temperature. This reduction in efficiency is, however, more than compensated for by the increased flow of organic fluid which results from the increased yield of steam—hence the improved specific output. The effect of variations in the primary flash temperature on the organic flow rate, the primary and secondary steam flow rates and the secondary flash temperature is shown in Figure 27.

The performance of several hydrocarbon compounds utilizing the 700 F resource is shown in Figure 28. The primary flash temperature in this case is 415 F. Both N-pentane and isopentane are ideally suited to this resource. Not only do they yield high specific outputs, but their operating conditions at maximum power also produce dry expansions with a minimum of superheat in the turbine exhaust. To achieve this same condition, turbine inlet pressures for isobutane and N-butane would have to be about 3000 psia and 1500 psi, respectively, which in both cases would give considerably less than optimum output. Although the N-pentane and isopentane turbines will be fairly large as a result of their relatively low exhaust pressures and high specific exhaust volumes, 50 MW of generating capacity can be obtained from a single unit, as demonstrated in the following section.

c. TURBINE CHARACTERISTICS AND DESIGN REQUIREMENTS

An attempt is made in this section to identify in a preliminary fashion the turbine characteristics for the various organic compounds studied in the previous section. Fifty MW has been selected as unit plant capacity. The flow rates involved in this size range are beyond the capacity of high efficiency radial flow machines, and only axial flow designs are considered.

Unfortunately there are no precise rules for the design of an axial flow turbine, for it is essentially a trial and error process. However some of the more important design requirements are discussed in the following paragraphs. For more detailed information the reader is referred to Reference 5.

1. Number of Stages

If the stage pressure ratios are assumed equal, the number of turbine stages can be expressed simply as

$$r = R^{1/n} \quad (23)$$

where r and R refer to the stage and overall pressure ratios, respectively. The stage pressure ratio will of course depend on a number of different design considerations or compromises, but

Figure 26. Effect of Primary Flash Temperature on Specific Output, Efficiency Pressure Ratio, and Exhaust Superheat of Isobutane 2-Stage Flash Cycle

Figure 27. Effect of Primary Flash Temperature on Secondary Flash Temperature and Steam Yield for Isobutane 2-Stage Flash Cycle

Figure 28. Performance Characteristics of Hydrocarbon Compounds in 2-Stage Flash Cycles for Brine at 700 F

one can assume for a reasonably loaded stage that it will be close to or above the critical value r_c specifies the critical velocity in each stage.

2. Exhaust Annulus Area

The exhaust annulus area is given by the continuity equation, i.e.,

$$A_e = \dot{m} v_e / V_e \quad (24)$$

where $\dot{m}$ is the mass flow rate, v_e the specific volume of the exhaust flow and V_e the axial component of the exhaust velocity vector. If the exhaust flow is assumed axial (zero swirl), then the exhaust velocity can be expressed in terms of a leaving loss coefficient, i.e.,

$$\epsilon_{LL} = \frac{V_e^2}{2\Delta H_{is}} = \frac{V_e^2 \eta_{TS}}{2\Delta H} \quad (25)$$

where ΔH_{is} is the total to static stage isentropic enthalpy drop, η_{TS} the total to static stage efficiency and ΔH the actual enthalpy drop in the stage. If the stage pressure ratios are now allowed to vary sufficiently to make the stage outputs approximately equal, this loss coefficient becomes

$$\xi_{LL} = \frac{V_e^2 \eta_{TS}}{2 \Delta H_T} \quad (26)$$

where ΔH_T is the total enthalpy drop in the turbine. Substituting this expression into Equation 24 and replacing the mass flow rate by the power rating divided by the net specific work, the exhaust area becomes

$$A_e = \frac{P}{\Delta H_{NET}} V_e \sqrt{\frac{n \eta_{TS}}{2 \xi_{LL} \Delta H_T}} \quad (27)$$

3. Blade Speed

If we assume a value for the velocity ratio σ_{is} , i.e., the ratio of the blade speed to the velocity that would be achieved in an isentropic expansion from the initial stagnation temperature through the total pressure ratio across the stage, then the blade speed can be determined from the relation

$$U = \sigma_{is} c_{is} \quad (28)$$

or

$$U = \sigma_{is} \sqrt{\frac{2 \Delta H_T}{n \eta_{TT}}} \quad (29)$$

where η_{TT} , the total to total efficiency, is approximately related to the total to static efficiency by the expression

$$\eta_{TT} = \eta_{TS} / (1 - \xi_{LL}) \quad (30)$$

4. Blade Root-Stress

In this simplified analysis, bending stress due to aerodynamic loading is ignored and only centrifugal stress is considered. For a tapered blade, the root-centrifugal stress can be written

$$\sigma_r = K 2 \pi \frac{\rho}{g} \left(\frac{N}{60} \right)^2 \frac{A_e}{n_e} \quad (31)$$

where K is a taper factor (<1) and n_e is the number of exhaust ends.

Generally, aerodynamic considerations dictate that the ratio of the blade height to pitch diameter be maintained within some allowable limit α so that if the exhaust area is expressed in terms of the pitch diameter and blade height, i.e.,

$$A_e = \pi D_p h n_e = \pi \alpha D_p^2 n_e \quad (32)$$

the root stress becomes

$$\sigma_r = 2\pi^2 K \frac{\rho}{g} \left(\frac{N}{60}\right)^2 D_p^2 \alpha = 2K\alpha \frac{\rho}{g} \sigma_{is}^2 c_{is}^2 \quad (33)$$

Assuming a taper factor of 0.8 (in a linearly tapered blade this corresponds to a tip area/root area ratio of about 0.6), a density of 0.283 lb/in.³, a velocity ratio of 0.5, and a blade height to pitch diameter ratio of 0.25, the centrifugal root stress becomes

$$\sigma_r = 0.0105 c_{is}^2 \quad (34)$$

For the low temperatures at which the organic compounds are utilized, the value of c_{is} which corresponds to the sound speed at critical total pressure ratios, will be relatively low. Consequently the root stress will also be low and therefore should impose no limitations on the design.*

5. Shape Parameter (Specific Speed)

The shape parameter N_s is defined as

$$N_s = \phi^{1/2} / \psi^{3/4} \quad (35)$$

where ϕ and ψ are the flow coefficient and blade loading coefficient, respectively. They are defined:

$$\phi = \frac{c_x}{U} \text{ and } \psi = \frac{\Delta H}{U^2} \quad (36a,b)$$

where c_x is the entry axial velocity to the blade row. If the stage is of free vortex design, the axial velocity remains unchanged through the stage and thus the flow coefficient can be written

$$\phi = \frac{\dot{m} v_e}{A_e U} \quad (37)$$

The shape parameter in turn becomes

$$N_s = \left(\frac{\dot{m} v_e}{A_e U} \right)^{1/2} / \frac{\Delta H}{U^2}^{3/4} \quad (38)$$

or if A_e is replaced by $\alpha \pi D_p^2$ and U by $\pi D_p N/60$,

$$N_s = \left(\frac{\pi}{\alpha} \right)^{1/2} \frac{N}{60} \left(\dot{m}' v_e \right)^{1/2} / \Delta H^{3/4} \quad (39)$$

where the turbine flow rate $\dot{m}'$ is the total flow rate divided by the number of exhaust ends. Note the absence of turbine dimensions in the expression for N_s . For high efficiency machines, the value

*For 12Cr steel, the maximum allowable root stress will be the order of 25,000 psia.

of N_s should lie between 0.25 and 1.0 and for highest efficiencies should be close to 0.5. The blade loading coefficient should exceed 1.0 and, if the exhaust is to have zero swirl, be less than 2.0.

The number of units is kept to a minimum by increasing the diameter to accommodate the required exhaust flow while decreasing the rotational frequency to maintain the desired blade speed or velocity ratio. If the number of turbine stages has already been established, both the isentropic velocity and the total exhaust flow area have known values. The number of turbine stages selected should provide reasonable values for the design parameters N_s , ϕ , and ψ . For 60 cycle a-c power generation, the rotational frequency must of course be a rational fraction of 3600 rpm.

Turbine characteristics for the various candidate working fluids operating in a dual flash cycle are given for the three resource temperatures in Tables 8, 9, and 10. Except for the isolated case of isopentane operating on the 500 F resource, no attempt was made to compromise output in favor of either eliminating condensation in the turbine or reducing superheat in the turbine exhaust; all specific outputs quoted for the other cycles are maximum values. For the organic turbines, the leaving loss coefficient was assumed to be 0.1. For the steam turbines, the exhaust flow per unit area was kept to less than 30,000 lb/hr/ft².

For the 400 F resource, there is relatively little to choose between the organic compounds. All can generate 50MW of electrical power with a single turbine. Isobutane utilizes the resource a little more effectively than the others and does so at more or less ideal operating conditions, i.e., with a minimum of superheat in the turbine exhaust. However, by far the greatest output is obtained if steam is used directly in a steam turbine. Since the lower temperature resource contains very little noncondensable gas, this steam turbine cycle is feasible for this resource. For the same condenser temperature of 100 F steam will develop about 37% more power than isobutane. In calculating the steam turbine output, the high and low pressure stage efficiencies were assumed to be 0.73 and 0.70, respectively. In order to accommodate the large volume flow rate in the last stage of the single unit shown, it was necessary to increase the blade height to pitch diameter ratio of 0.34, the current practical limit, and to use the largest permissible pitch diameter of 6 ft. The result is an exhaust annulus size that is essentially three times that of an isobutane turbine. The isobutane turbine also has two fewer stages. However, despite its physical dimensions, the steam turbine eliminates the need for a circulating pump and heat exchangers and as shown in Section 7, is a less expensive cycle.

The steam turbine cycle is not handicapped by circulating pump power requirements and losses which occur in the primary heat exchange process of the binary cycle. So despite its lower turbine efficiency it should always yield the greatest output for a given resource temperature provided steam is used for the primary heat

Table 8

TURBINE CHARACTERISTICS FOR DUAL FLASH/400 F BRINE CYCLE

Working Fluid	Specific Output Btu/lb Brine	Efficiency	Total Flow For 50 MW (Lb/s)	No. Exhaust Ends	No. Stages	Overall Pressure Ratio	Speed (rpm)	LAST STAGE					
								Pitch Dia (Ft)	Exhaust Annulus Area (Ft ²)	Flow Coefficient	Blade Loading Coefficient	Reaction	Specific Speed N _s
Propene	20.65	0.1111	2709	1	2	4.12	3600	2.53	5.03	0.5343	1.3163	0.3419	0.5948
Propane	21.08	0.1133	2511	1	2	4.54	3600	2.67	5.61	0.5146	1.2210	0.3895	0.6175
Isobutane	23.35	0.1290	2186	1	3	6.79	1800	4.02	12.70	0.5609	1.4504	0.2748	0.5666*
R-32	20.60	0.1086	3408	1	2	3.91	3600	2.179	3.729	0.5463	1.3768	0.3116	0.5815
R-115	20.28	0.0979	10101	1	2	5.21	1200	3.938	12.179	0.5699	1.4980	0.2510	0.5570
Steam	32.00	0.1370	296	1	5	70	3600	6.000	38.45	--	--	--	-- **

Table 9

TURBINE CHARACTERISTICS FOR DUAL FLASH/500 F BRINE CYCLE

Working Fluid	Specific Output Btu/lb Brine	Efficiency	Total Flow For 50 MW (Lb/s)	No. Exhaust Ends	No. Stages	Overall Pressure Ratio	Speed (rpm)	LAST STAGE					
								Pitch Dia (Ft)	Exhaust Annulus Area (Ft ²)	Flow Coefficient	Blade Loading Coefficient	Reaction	Specific Speed N _s
Propene	35.21	0.1329	1978	1	3	6.18	3600	2.31	4.20	0.5723	1.5105	0.2448	0.5552
Propane	35.48	0.1334	1770	1	3	6.30	3600	2.44	4.69	0.5498	1.3941	0.3030	0.5779
Isobutane	38.11	0.1449	1654	1	4	10.35	1800	3.68	10.62	0.6186	1.7647	0.1176	0.5137
N-Butane	40.33	0.1557	1517	1	4	13.15	1800	4.09	13.15	0.5710	1.5003	0.2484	0.5565
Isopentane	36.51	0.1718	1289	1	5	23.15	1200	5.60	24.65	0.6105	1.7185	0.1407	0.5205†
Steam	53.61	0.1782	240	1	7	173	3600	6.00	38.45	--	--	--	-- **

Table 10

TURBINE CHARACTERISTICS FOR DUAL FLASH/700 F BRINE CYCLE

Working Fluid	Specific Output Btu/lb Brine	Efficiency	Total Flow For 50 MW (Lb/s)	No. Exhaust Ends	No. Stages	Overall Pressure Ratio	Speed (rpm)	LAST STAGE					
								Pitch Dia (Ft)	Exhaust Annulus Area (Ft ²)	Flow Coefficient	Blade Loading Coefficient	Reaction	Specific Speed N _s
Isobutane	77.54	0.1552	1442	1	5	16.17	1800	3.53	9.78	0.6363	1.8671	0.0665	0.4994
N-Butane	82.80	0.1663	1264	1	5	17.61	1800	3.91	11.99	0.5913	1.6125	0.1938	0.5374
Isopentane	86.22	0.1727	1225	1	6	26.65	1200	5.88	27.15	0.5293	1.2916	0.3542	0.6004
N-Pentane	88.49	0.1791	1093	1	6	29.61	1200	5.73	30.90	0.5696	1.4960	0.2520	0.6111††

* Subcritical Pressure Cycles

** Blade Height/Pitch Dia = 0.34

† Primary Flash Temperature T₈₁ = 400 F

†† Blade Height/Pitch Dia = 0.3

exchange process. For as can be seen from the summary, Figure 23, for the brine/organic cycle, the latter is potentially the better cycle in terms of output if the well can be pumped and if pumping can be done efficiently. For the 500 F resource, the output of the dual flash steam turbine cycle is again well over 30% greater than its nearest binary cycle competitor, which in this case is normal butane. Again maximum permissible values were employed for the pitch diameter and the blade height to pitch diameter ratio in order to accommodate the large volume flow in the single exhaust.

Unfortunately, the large concentration of CO_2 in this high temperature brine requires prohibitively large amounts of extraction pump work to remove the noncondensable gas, CO_2 , from the condenser and maintain a subatmospheric exhaust. An example of the amount of pump work required to maintain the turbine exhaust temperature at 100 F is shown in Figure 29. The resource temperature is 500 F and the pump efficiency and the exhaust flow quality were assumed to be 85% and 75%, respectively. If extraction were to occur at the exhaust temperature, it makes no difference how much CO_2 is present in the flow since the entire vapor phase must be extracted with the CO_2 and the pump work required would clearly be prohibitively high. If the steam velocity is low, the total pressure in the condenser will remain essentially constant so that as condensation proceeds, the partial pressure of CO_2 will rise while the vapor pressure will fall, causing the condensate temperature to fall also. If the extraction point is moved away from the turbine exhaust, the reduction in the vapor extracted with the CO_2 causes a marked reduction in the pump work as shown in Figure 29. A similar result would be obtained if the exhaust flow temperature were increased to, say, 110 F and the condensate temperatures decreased in the same increments. For the Niland brine in which the CO_2 concentration can amount to 5% by weight, the pump work will be some 15 Btu/lb brine even when the condensate is cooled some ten degrees below the turbine-exhaust temperature. The output of the steam turbine would thus be reduced approximately to that of normal butane or isobutane cycles.

For the 700 F resource, isopentane and N-pentane utilize the brine most effectively. As pointed out in the preceding section, they produce maximum output at near ideal operating conditions; whereas in the isobutane and N-butane cycles at the condition of maximum power, there is appreciable superheat in the turbine exhausts. However it is possible that by lowering the primary flash temperature, the specific output of N-butane could be increased and the exhaust superheat decreased. Again, 50MW of power can be generated by a single unit by each compound although the N-butane and isobutane turbines are more compact.

d. SUMMARY OF PRELIMINARY RESULTS AND SELECTION OF CYCLES FOR DETAILED ANALYSIS

For the brine/binary cycle, in which the brine is utilized directly to transfer energy to the secondary working fluid, N-pentane and isopentane produce the highest net output per unit of

- brine flow when the brine temperature is 500 F and above. When the brine temperature is 400 F, the highest output is obtained from isobutane. For isobutane and isopentane, conditions which produce maximum output also produce a minimum of superheat in the turbine exhaust.

If the brine is flashed to steam in two stages, and the steam is used to heat the secondary working fluid, then N-pentane, N-butane, and isobutane produce the highest output when the resource or initial brine temperature is 700 F, 500 F, and 400 F, respectively. Again, for each compound, conditions which produce highest output also produce the lowest possible exhaust superheat when a dry expansion is maintained in the turbine.

If the noncondensable gas content in the brine is low, a dual flash steam turbine cycle will produce more power per unit of brine flow than a dual flash/binary cycle but less power than the brine/binary cycle.

For all the compounds studied, including steam, it is possible to generate 50 MW of electrical power with a single turbine without exceeding mechanical design or manufacturing limitations in the size of the last stage.

Since steam turbines have been used exclusively so far to exploit liquid-dominated resources, their cycle performance and the cost of turbomachinery and associated components are fairly well established for a variety of different resource temperatures and operating conditions. Since steam turbines can also be utilized effectively at East Mesa, the steam turbine cycle has been selected for detailed analysis with this resource along with the isobutane cycle, which produced the greatest output of all closed Rankine cycles at this resource condition.

A comparison of steam turbine and binary cycles for the other brines would be academic since condensing steam turbines cannot function effectively on the Niland resource. The best candidate working fluids for the binary cycle in terms of resource utilization are N-butane at 500 F and isopentane or N-pentane at 700 F. N-butane also performs fairly effectively at the higher resource temperature where its net specific output is only some 4 percent less than isopentane. However, in the following detailed analysis, only the N-butane cycle operating on the 500 F resource has been investigated. The higher temperature resource was ignored primarily because published information on the effects of salinity on brine specific enthalpy (at the levels of interest) which are included in the following section, are valid only to 500 F or so. In retrospect, the 700 F brine appears not to represent general reservoir conditions, having been located by only a single well (Elnore No. 1) which was shut-in in 1965.

Figure 29. CO₂ Extraction Pump Work

Section 4

DETAILED CYCLE ANALYSIS

A. THE EFFECTS OF SALINITY AND NON-CONDENSIBLE GAS ON BINARY CYCLE PERFORMANCE

In assessing the relative merits of the various candidate working fluids and selecting the most promising fluids for the closed cycles which are analyzed in detail in this section, no attempt was made in the previous section to account for the effects of brine salinity on cycle performance. The effects of salinity on cycle output can in fact be quite dramatic since dissolved salts reduce the brine specific enthalpy, and therefore, reduce the quantity of steam liberated from a given temperature flash. The overall effect of salinity on the N-butane cycle recommended for use with the Niland resource is shown in Figure 30. The resource temperature and condensing temperature are 500 F and 100 F, respectively. Except for the specific heat, which is now assumed to be a function of temperature, all other assumptions are as listed in Table 6.

Figure 30. Effect of Salinity On The Performance Of The Dual Flash/N-Butane Cycle

Three families of curves are shown for the three different sodium chloride concentrations representative of East Mesa brine, Niland brine and pure water. Each family represents a performance plot for different primary flash temperatures; the numbers on each curve representing the turbine inlet temperature in psia. Again, for reasons explained in the previous section, a decrease in flash temperature results in an increase in output. However, this gain is relatively small compared to the loss in output caused by salinity. Overall a reduction of some 37% in output is experienced when the sodium chloride level increases from zero to 300,000 ppm. The efficiency of the cycle is shown to change little with salinity and the output loss is therefore due principally to the reduced steam flow which in turn supports less organic fluid in the closed cycle. These effects are shown in the plots of organic and steam flow rates vs. primary flash temperature for different salinities in Figure 31. The brine properties on which the above calculations were based were developed by Silvester and Pitzer (B5), and are summarized in APPENDIX B.

Figure 31. Variation of Steam and Organic Fluid Flow Rates With Salinity and Primary Flash Temperature in the Dual Flash/N-Butane Cycle

In addition to reducing the brine specific enthalpy, dissolved salts also reduce the saturation pressure relative to temperature. Therefore, since the two phase brine/steam mixture is in temperature equilibrium, the steam is initially superheated.

Typically, steam flashed at 360 F from Niland brine will be superheated by about 25 F. Compared to the latent heat of condensation, the enthalpy content of the superheat is small and it contributes very little to heating the organic fluid as depicted in the temperature/enthalpy diagram of Figure 32 for the dual flash/organic cycle shown schematically in Figure 33. Thus, the turbine inlet temperature is effectively limited by the saturation temperature and not the superheat temperature of the primary steam*. Some loss in output is experienced as a result of this superheat effect since the primary flash temperature must be increased if operation of the organic cycle is to remain supercritical. The effect of salinity on cycle temperatures and pressure ratios required for maximum output is shown in Figures 34 and 35 for different primary flash temperatures.

The cycle performance is further degraded by the presence of non-condensable gas. At Niland for example, the dissolved gas is principally CO₂ and can amount to 5% of the brine weight. Since nearly all of this gas is released during the primary flash process, this concentration translates to over 40% by weight of the primary steam flow. In such a system, condensation is not of course a constant temperature process. Since the total pressure in the condenser will remain essentially constant, any condensation of steam will cause the vapor pressure to fall and the condensing temperature will fall also. Hence, condensation will tend to follow the curved path depicted in Figure 32. The vapor which remains at the downstream end of the condenser at temperature T_{S_2} is vented with the gas and is therefore lost to the cycle. The fact that some vapor remains at the condenser exit means that the secondary flash temperature is greater than it would be with CO₂ absent. This increased secondary flash temperature produces a reduction in secondary steam flow which combined with the vapor loss in the primary condenser** produces the reduction in output indicated in Figure 36. The two families of curves shown represent CO₂ weight concentrations of zero and 5%, respectively. The resource temperature, organic condensing temperature and sodium chloride concentration are 500 F, 100 F and 300,000 ppm. For the range of primary flash temperatures shown, 5% CO₂ in the brine produces about a 5% reduction in cycle output. The effect of CO₂ concentration on cycle flow rates, secondary flash temperature and fraction of steam lost in the primary condenser is shown as a function of primary flash temperature in Figure 37. As will be shown later, the lost output due to inert gas is further compounded by the gases' deleterious effect on condensing heat transfer coefficients which results in a substantial increase in the size of the condenser.

*In reality, any superheat in the steam will be removed by the steam scrubbers.

**For 5% CO₂ in the brine, the vapor loss is approximately 7.5% of the primary steam flow.

Figure 32. Temperature/Enthalpy Diagram for Primary and Secondary Heat Exchangers in Dual Flash/Organic Rankine Cycle

Figure 33. Schematic of Dual Flash/Organic Rankine Cycle

Figure 34. Effect of Salinity on Cycle Temperatures in Dual Flash/N-Butane Cycle

Figure 35. Effect of Salinity on Pressure Ratio and Specific Output of the Dual Flash/N-Butane Cycle

Figure 36. Effect of CO₂ Concentration on Performance of the Dual Flash/N-Butane Cycle

Figure 37. Effect of CO₂ Concentration On Secondary Flash Temperature, Flow Rates and Output for the Dual Flash/N-Butane Cycle

The lower family of performance plots in Figure 36 is reproduced in Figure 38 to show the degree to which output must be compromised in order to minimize superheat in the turbine exhaust—a condition obtained when the expansion line lies tangent to the saturated vapor line. Clearly the effect on output is negligible when the primary flash temperature is greater than 360 F.

The performance of the dual flash/isobutane cycle operating at East Mesa is shown as a function of primary flash temperature in Figure 39. The resource temperature, isobutane condensing temperature and salinity are 400 F, 100 F and 25,000 ppm, respectively, and the gas content is assumed to be zero. Again, output increases with decreasing primary flash temperature due to the increased steam yield more than compensating for the loss in cycle efficiency at the lower operating temperature. There is, of course, a limit to which the primary flash temperature can be lowered if the organic cycle is to remain supercritical. At the lower primary flash temperature shown, the isobutane is in fact at subcritical pressure and any further reduction in temperature would decrease output. With only 25,000 ppm of sodium chloride present, the effect of superheat on operating characteristics and output is negligibly small since the superheat is, in fact, only a degree or so. Again, the loss of output is seen to be small when operating conditions are adjusted to minimize exhaust superheat. Cycle flow rates, secondary flash temperature and pressure ratio required for maximum output are plotted as a function of primary flash temperature in Figure 40.

Figure 38. Operating Characteristics for the Dual Flash/N-Butane Cycle

Figure 39. Operating Characteristics for the Dual Flash/Isobutane Cycle

It is interesting to note at this point that the performance of the N-butane cycle at Niland has been so degraded by the high salinity that its output is only some 20% greater than that of the isobutane cycle and almost 20% less than that of the steam turbine cycle both operating on the moderate temperature brine at East Mesa. A comparison of the three cycles is shown for different primary flash temperatures in Figure 41. In calculating the steam turbine output, the unit was assumed to be of dual admission type, i.e., the secondary steam is admitted and mixed with primary steam at some intermediate point between throttle (primary inlet) and exhaust. To be consistent with values calculated for the detailed design described later, high pressure and low pressure expansion efficiencies were assumed to be 0.817 and 0.736 for primary and secondary flash temperatures of 300 F and 200 F, respectively. The latter temperatures were selected because as shown in APPENDIX C, they give close to maximum output from the turbine.

B. SELECTION OF CYCLE OPERATING CONDITIONS

Before proceeding further, it is worthwhile at this point to consider how the various parameters such as condenser temperature and turbine efficiency, etc., influence the performance of the organic cycle. An example is given in Figure 42 of their effect on the N-butane cycle. The resource temperature is 500 F and the salt concentration 30%. W represents output and W_0 is the output

Figure 40. Effect of Primary Flash Temperature on Secondary Flash Temperature, Flow Rates, Pressure Ratio and Output for the Dual Flash/Isobutane Cycle

of the basic cycle when influence parameters have the values indicated at the intersection with the ordinate. As expected, performance is very sensitive to condensing temperature. An increase of 20 F reduces output by about 15%. The turbine efficiency also produces significant changes although for the efficiency range likely to be encountered in practice it has less influence than condensing temperature. Since circulating pump power requirements are small compared to turbine power, the effect of the pump efficiency on cycle output is also small. The pinch temperature difference maintained in the primary and secondary heat exchangers exerts moderate influence on output. A decrease from 25 F to 10 F will increase output by about 7% although such a change will, of course, be accompanied by an increase in heat exchanger size. The effect of the condenser pressure drop $(\Delta p/p)_c$ is relatively minor. Doubling its value from 0.05 to 0.10 decreases output by only 2%. The series pressure drop through the primary and secondary heat exchangers only provides extra work for the circulating pump. Consequently, it has an almost insignificant effect on plant performance, and therefore, it was omitted from the graph.

Figure 41. Comparison of the Performance of the Dual Flash Isobutane, N-Butane and Steam Turbine Cycles

In view of the relative insensitivity of cycle performance on machine efficiencies and pressure drops in the ranges of practical interest and of the uncertainty in predicting optimum values prior to equipment design, all influence parameters with the exception of condenser temperature were assumed to remain unchanged from the values used in the basic cycle.

Normally, the condensing system of a conventional power plant is designed according to a yearly average dry bulb temperature, and if the system employs wet cooling towers, a corresponding relative humidity. However, because of their short energy range it was decided in the case of the geothermal power cycles to depart from this convention and base the condensing temperature on the average temperature for the hottest summer month, thereby, guaranteeing on a monthly average basis, the quoted power output all year round. The variation of ambient temperature from January to December for Imperial California as recorded at the El Centro airport is listed in Table 11. The highest average temperature recorded for July is 91.5 F when the average relative humidity is 37%. In the following analysis of plant design, ambient conditions of temperature and humidity were assumed for convenience to be 95 F and 40%, respectively. The corresponding

wet bulb temperature is 75 F. Thus if wet atmospheric cooling is employed and an 'approach' of 15 F is assumed, the condenser inlet cooling water temperature will be 90 F. Under these conditions, a reasonable condensing temperature is 120 F. A summary of the basic assumptions used in this section is given in Table 12. In each cycle, the total mass of steam condensate always slightly exceeds the evaporative water loss in the wet cooling towers and since an external water source is unlikely to be available, condensate has been used for cooling water makeup throughout.

Figure 42. Effect of Condenser and Pinch Temperatures, Turbine and Circulating Pump Efficiencies and Condenser Pressure Drop on the Performance of the Dual Flash/N-Butane Cycle

Performance characteristics of the isobutane and N-butane cycles when the condensing temperature is 120 F are shown in Figure 43(A) and 43(B) for two different primary flash temperatures. In neither case does operating the cycle to minimize exhaust superheat result in a loss in output of more than 1%. In the following analysis of plant design, cycle performance was based on the lower primary flash temperature shown, i.e., 310 F for the isobutane cycle and 360 F for the N-butane cycle. Also, to ensure minimum exhaust superheat, turbine inlet pressures were 530 psia and 540 psia, respectively. The isobutane cycle is therefore almost exactly critical while the N-butane cycle is slightly subcritical. Operating pressures, temperatures and flow rates within the two cycles are listed in Table 13. Also listed are the operating conditions and approximate performance of the steam turbine cycle. Primary and secondary flash temper-

Table 11

AMBIENT TEMPERATURES FOR IMPERIAL CALIFORNIA

Month	T _{max} (°F)	T _{min}	T _{av,dry}	T _{av,wet}	Av. Rel. Humidity (%)
Jan.	68.8	38.3	53.6	45.3	52
Feb.	73.1	42.7	57.9	47.3	44
Mar.	79.6	47.8	63.7	50.8	39
Apr.	86.7	54.7	70.7	55.3	36
May	94.2	61.3	77.8	59.0	31
June	101.7	67.7	84.7	63.0	28
July	107.0	75.9	91.5	71.2	37
Aug.	105.2	76.0	90.6	71.1	38
Sept.	100.9	68.7	84.8	64.3	32
Oct.	90.2	57.5	73.9	57.4	35
Nov.	78.6	45.9	62.3	49.7	39
Dec.	70.1	40.4	<u>55.3</u>	<u>45.5</u>	<u>45</u>
			Av. 72.2	56.6	38

(Source: U.S. Dept. of Commerce National Climatic Center, Ashville N.C.)

Table 12

LIST OF ASSUMPTIONS FOR
DUAL FLASH/ORGANIC CYCLES

Condenser Temp.	120 F
Pinchpoint Temp. (Primary and Secondary Heat Exchangers)	25 F
Series Pressure Drop Through Primary and Secondary Heat Exchangers $(\Delta p/p)_b$	0.10
Condenser Pressure Drop $(\Delta p/p)_c$	0.05
Turbine Efficiency	0.85
Pump Efficiency	0.85
Ambient Dry Bulb Temp.	95 F
Relative Humidity	40%

Figure 43A. Operating Characteristics for the Dual Flash/Isobutane Cycle

Figure 43B. Operating Characteristics For The Dual Flash/N-Butane Cycle

Table 13

OPERATING CONDITIONS OF
DUAL FLASH/ISOBUTANE AND N-BUTANE CYCLES

<u>RESOURCE CONDITIONS</u>			
T_{B1} (°F)	400	400	500
X_C (ppm)	25,000	25,000	300,000
X_{CO_2} (lb/lb brine)	0	0	0.05
Working Fluid	Isobutane	Steam	N-Butane
<u>OPERATING TEMPERATURE & PRESSURE</u>			
P_0 (psia)	98.1	-	69.5
P_1 "	583.0	-	594.0
P_2 "	530.0	-	540.0
P_3 "	103.0	-	73.0
T_0 (°F)	120.0	120	120.0
T_1 "	124.7	-	123.8
T_2 "	284.0	-	310.3
T_3 "	161.7	-	160.9
T_{S1} "	310.0	300	360.0
T'_{S1} "	309.0	299	335.3
T_{S2} "	252.7	200	282.2
T'_{S2} "	252.0	199.4	265.1
T_{S3} "	149.7	-	148.8
<u>FLOW RATES</u>			
M (lb/lb brine)	0.9538	-	0.9096
M_{S1} " " "	0.0983	0.1085	0.1175
M_{S2} " " "	0.0548	0.0898	0.0515
<u>CYCLE OUTPUT AND EFFICIENCY</u>			
W (Btu/lb brine (shaft))	18.77	28.64	23.03
W (kW _e /million lb brine/hr)	5419	8268	6649
η cycle	0.1158	-	0.1347
η utilization	0.3561	0.5433	0.3186

atures were assumed to be 300 F and 200 F, respectively. As shown in APPENDIX C, these are close to the optimum values for producing maximum power. To be consistent with the detailed designs described later, the turbine is assumed to be of dual admission type in which primary and secondary steam flows are mixed at the secondary admission point. Again, to be consistent with the detailed design described later, high and low pressure expansion efficiencies were assumed to be 0.817 and 0.736, respectively, and the turbine produces 24% and 52% (respectively) more power than the dual flash primary cycles at Niland East Mesa. The steam turbine cycle is of course not subject to the limitations of the binary cycle where the secondary flash temperature, since it is coupled to the organic heating curve, is fairly high and the yield of secondary steam therefore fairly low. The utilization efficiencies shown are defined as the actual net work produced by the cycle divided by the theoretical maximum work obtained if the brine is expanded isentropically from reservoir conditions to the condensing temperature.

Section 5

HEAT EXCHANGER DESIGNS

All heat exchangers employed on the closed organic Rankine cycle are assumed to be of conventional floating head shell/tube design. In all, three different functions are performed - condensation of organic fluid, condensation of primary and secondary steam and the subcooling of steam condensate. The various heat transfer mechanisms which control these processes and the empirical relations which apply to shell/tube designs are discussed in APPENDIX A. The organic fluid properties on which designs were based were derived from the empirical relationships listed in APPENDIX B.

A. STEAM CONDENSERS AND SUBCOOLERS

Details of the primary and secondary steam condensers and subcoolers are listed in Tables 14 and 15. In each unit a single pass and double pass is employed on the shell side and tube side, respectively. Because of the relatively large degree of subcooling required for both primary and secondary steam condensate and since subcooling is achieved with relatively poor efficiency by natural convection if performed in the condenser, separate subcoolers are employed in each case. Such an arrangement allows the condensate to be force circulated which improves heat transfer and leads to a more compact design overall. Each unit was designed using a fouling factor of 0.005 which although relatively large for conventional systems is not unreasonable for geothermal applications in view of the potential scaling problem from carryover in the steam. For 50MW of electrical generating capacity it is possible to perform each condensing heat transfer process with a single unit in both the 400 F and 500 F brine cycles without exceeding manufacturing or shipping limitations. The large shell diameters employed result in large numbers of tubes which, despite two passes, impose little restriction to the flow of organic fluid. The shell side pressure drop is, by comparison, fairly large due to the initially large volumetric steam flow and to the fairly long serpentine flow path created by baffling. Although condensing coefficients are theoretically independent of flow modifications induced by baffling, a few baffles are in fact necessary to distribute the steam uniformly over the entire body of the tube bank. In the subcoolers where external heat transfer is dependent on flow velocity, the baffle frequency is increased considerably over that employed in the condensers. In calculating heat exchanger characteristics, the organic fluid properties were assumed constant and were based on the arithmetic mean of the inlet and exit temperatures.

Details of the effect of CO₂ concentration on condenser design are summarized for single and double units in Tables 16 and 17. The actual surface area required to maintain an overall fouling factor of 0.005 is shown as a function of CO₂ concentration in Figure 44.

Table 14
PRIMARY & SECONDARY HEAT EXCHANGERS

Unit	No. Units	Brine Temp. (°F)	Organic Fluid	HOT FLUID (SHELL SIDE)					COLD FLUID (TUBE SIDE)					LMTD (°F)	Duty (10 ⁶ Btu/hr)
				T _{in} (°F)	T _{out} (°F)	P _{in} (psia)	P _{out} (psia)	m (10 ³ lb/hr)	T _{in} (°F)	T _{out} (°F)	P _{in} (psia)	P _{out} (psia)	m (10 ³ lb/hr)		
PC	1	400	Isobutane	309.5	304.5	77.0	71.5	899	235.0	284.5	533.2	530.0	8730	44.9	817
PS	1	400	Isobutane	304.5	251.0	71.5	66.0	899	226.0	235.8	541.4	533.2	8730	44.5	49.7
SC	1	400	Isobutane	251.0	242.0	30.4	25.9	502	150.0	226.0	544.8	541.4	8730	51.7	478
SS	1	400	Isobutane	248.0	149.4	28.8	23.3	1401	124.4	150.0	552.0	544.8	8730	54.4	143
PC	1	500	N-butane	335.3	330.3	110.5	104.5	827	251.0	310.3	542.1	540.0	6795	49.1	735
PS	1	500	N-butane	330.3	263.3	104.5	100.4	827	238.3	251.0	546.7	542.1	6795	49.5	56.8
SC	1	500	N-butane	263.3	257.9	37.4	34.0	375	160.5	238.3	548.6	546.7	6795	54.0	362
SS	1	500	N-butane	261.0	148.6	36.0	31.9	1202	123.6	160.5	553.0	548.6	6795	55.6	141

PC = Primary Condenser

PS = Primary Subcooler

SC = Secondary Condenser

SS = Secondary Subcooler

Note: All Tubes 1.0" OD x 0.834" ID (14 BWG) on 1.25" Triangular Pitch

Configuration: Floating Head 1-2 Tube/Shell Exchangers

Table 15
PRIMARY & SECONDARY HEAT EXCHANGERS

Unit	No. Units	Brine Temp. (°F)	Organic Fluid	L (ft)	D (ft)	N _t	N _b	Duty (10 ⁶ Btu/hr)	G _t lb/hr	G _s ft ²	Re _t	Re _s	f _t x10 ⁴	f _s x10 ³	LMTD (°F)	h _t	h _s (Btu/ft ² hr°F)	U _c	U _d	f _{foul}
PC	1	400	Isobutane	52.0	12.5	9216	4	817	499000	31000	363000	57300	1.08	1.55	44.8	716	1624	374	130	0.0050
PS	1	400	Isobutane	10.0	7.5	3318	6	49.7	1390000	420000	771000	25300	0.89	1.81	44.5	1253	807	388	129	0.0052
SC	1	400	Isobutane	40.0	11.0	7137	3	478	644000	22800	275000	42100	1.17	1.64	53.3	496	1290	280	119	0.0048
SS	1	400	Isobutane	22.5	8.0	3775	8	143	1220000	350000	391000	21100	1.06	1.87	54.3	732	727	295	119	0.0050
PC	1	500	N-butane	52.0	12.0	8493	5	735	421000	36100	216000	66600	1.24	1.50	49.1	483	1634	287	117	0.0050
PS	1	500	N-butane	11.0	7.5	3318	6	57	1080000	355000	429000	21400	1.04	1.87	49.5	744	736	299	122	0.0049
SC	1	500	N-butane	34.0	11.0	7137	3	361	502000	20600	159000	38000	1.34	1.67	54.6	357	1232	220	104	0.0051
SS	1	500	N-butane	22.5	8.0	3775	8	141	949000	299000	222000	18000	1.23	1.93	55.6	528	665	241	114	0.0046

PC = Primary Condenser

PS = Primary Subcooler

SC = Secondary Condenser

SS = Secondary Subcooler

Note: All Tubes 1.0" OD x 0.834" ID (14 BWG) on 1.25" Triangular Pitch

Configuration: Floating Head 1-2 Tube/Shell Exchangers

Table 16

PRIMARY STEAM/CO₂ CONDENSER (500 F BRINE DUAL FLASH N-BUTANE CYCLE)

%CO ₂ **	HOT FLUID (SHELL SIDE)						COLD FLUID (TUBE SIDE)					ΔT _{wtd} (°F)	Duty (10 ⁶ Btu/hr	No. Units
	T _{in} (°F)	T _{out} (°F)	P _{in} (psia)	P _{out} (psia)	$\dot{m}_v$ (10 ³ lb/hr)	$\dot{m}_g$	T _{in} (°F)	T _{out} (°F)	P _{in} (psia)	P _{out} (psia)	$\dot{m}$ (10 ³ lb/hr)			
0	335.3	330.3	110.5	104.5	827	0	251.0	310.3	542.8	540.0	6795	49.1*	735	1
1	335.3	263.3	110.5	103.2	837	71	238.1	310.3	542.8	540.0	6795	41.1	789	1
3	335.3	264.5	110.5	97.8	859	219	239.1	310.3	543.5	540.0	6794	36.2	787	1
5	335.3	265.1	110.5	98.5	878	373	240.2	310.3	544.0	540.0	6794	32.7	786	1
1	335.3	263.3	110.5	106.3	418	35	238.1	310.3	540.8	540.0	3398	41.1	394	2
3	335.3	264.5	110.5	102.9	430	109	239.1	310.3	541.1	540.0	3397	36.2	393	2
5	335.3	265.1	110.5	100.8	439	186	240.2	310.3	541.2	540.0	3397	32.7	393	2

* LMTD Value

** % of Brine Weight

Note: All Tubes 1.0" OD x 0.834" ID (14 BWG) on 1.25" Triangular Pitch

Configuration: Floating Head 1-2 Tube/Shell Exchanger

Table 17

PRIMARY STEAM/CO₂ CONDENSER (500 F BRINE DUAL FLASH N-BUTANE CYCLE)

% CO ₂ *	L (ft)	D (ft)	N _d	N _t	No. Units
1	34.0	12.0	8	8493	2
3	40.25	12.0	8	8493	2
5	44.5	12.0	8	8493	2
1	62.0	12.0	9	8493	1
3	71.5	12.0	9	8493	1
5	79.0	12.0	8	8493	1

* % of Brine Weight

Note: All Tubes 1.0" OD x 0.834" ID (14 BWG) on 1.25" Triangular Pitch

Configuration: Floating Head 1-2 Tube/Shell Exchanger

Figure 44. Variation of (Steam) Heat Exchanger Surface Area With Weight Fraction of CO_2 in Brine

Two families of curves are represented - one for a single heat exchanger unit, the other for a double unit. The two curves marked A and B in each family represent respectively the results of calculations in which the transport properties of N-butane were (a) held constant through the exchanger and evaluated at the arithmetic mean of the inlet and exit temperature, and (b) allowed to vary with temperature and evaluated at the mean coolant temperature in each interval. Since properties of the organic compound are shown in APPENDIX B to be a fairly strong function of density, and since compressed liquid densities decrease appreciably as temperatures approach the critical region, the density mean is not coincident with the temperature mean and some discrepancy between results A and B is to be expected. The discrepancy indicated for the single unit was used to modify the length of previous primary condenser designs, where constant organic fluid properties had been assumed, to the values of 52 ft listed in Table 15. No modification of the secondary condensers or subcoolers was necessary since temperatures are sufficiently removed from the critical region and therefore errors incurred by assuming constant coolant properties are negligible.

The consistency of the above results with those of the primary condenser listed for the 500 F brine cycle in Table 15 can be compared by extending the curves A to intersect the ordinate.

This projects the area required for a condenser and subcooler combined operating without influence from CO₂ with a fouling factor of 0.005. A weighted temperature difference for such an arrangement is approximately 44.5 F compared to the log mean temperature difference of 49.1 F used for the condenser alone. Initially, with constant organic fluid properties assumed, the surface area required for the condenser was estimated to be 129,000 ft² and 155,000 ft² for single and double units, respectively. Thus, if condensation is assumed to remain the controlling mode of heat transfer, the area required by the condenser and subcooler combined is obtained by multiplying the above areas by the ratio of the new duty to old duty and the temperature ratio 49.1/44.5. The result for the single and double units is represented in Figure 44 by points 2 and 1, respectively. In view of the different methods employed in designing the condensers with and without the effects of CO₂, it is perhaps somewhat fortuitous that agreement between the two results is so good.

B. ORGANIC CONDENSERS

Despite the fact that the organic condenser must function initially as a desuperheater, designs were based on the assumption that condensation controlled the rate of external heat transfer throughout the unit. This simplification is not unreasonable since the degree of superheat is only some 40 F in both organic cycles, and therefore the tube wall temperature everywhere should be below the dewpoint.

In order to minimize the pressure at the turbine exhaust, the condensers were constructed without baffles. Since baffling serves no useful purpose in condensers other than to distribute the flow uniformly over the tube bank, it was felt that flow uniformity could be achieved as effectively but with less pressure drop by periodically injecting the turbine exhaust in cross flow across the tube bank from a manifold extending the length of the condensers.

To assist the relatively poor condensing coefficients of the organic fluids, condenser tubes were finned. Tubes were nominally 3/4 inch outside diameter, 16 BWG arranged on a 15/16 inch triangular pitch. Fin height was 0.0625 inches which, for 19 fins/inch, produced an external surface area of 0.496 ft²/linear foot - a two and a half fold increase over that of a plain tube. The condensers were designed with a single cross-flow pass on the shell side and a single pass on the tube side. Since contamination is not expected to be a problem within the closed cycle, a relatively small fouling factor of 0.001 was assumed for the condensers.

Typical temperature distributions for the shell and tube side fluids are shown in the temperature/enthalpy diagram in Figure 45. If wet atmospheric cooling is employed, the approach

temperature will be about 15 F, which for a 75 F wet-bulb, corresponds to a condenser cooling water inlet temperature of 90 F. The effect of cooling water temperature on condenser length, circulating pump power requirements and tube side pressure drop is shown for different condenser pinch temperatures in Figures 46 and 47. Two parallel condensing units, each 13 ft diameter are required to accommodate the turbine exhaust flow. The lengths indicated in Figure 45 are for each unit while the power requirements are for the total water flow circulated between the two units. As is to be expected, the condenser size is influenced more strongly by the pinch temperature than by cooling water temperature. However, the reverse is true for the power requirements of the circulating pump since the flow rate changes rapidly with T_{W1} . If the flow rate becomes excessive so too does the tube side pressure drop which is reflected in the power demands of the pump. The pump and drive efficiencies were assumed to be 0.85 and 0.92, respectively, and a head loss of 55 ft was assumed in addition to the tube side pressure drop.

Figure 45. Temperature/Enthalpy Diagram for the Organic Fluid Condenser

A similar set of curves to those presented for the condenser length is shown for the cooling tower in Figure 48. A tower unit is analogous to the surface area in conventional heat exchangers and is numerically equal to a rating factor multiplied by the water flow rate in gallons per minute. The rating factor (obtained from reference 6) is simply a measure of the difficulty the tower has in performing the cooling process. For example, a rating factor of 1.1 signifies a cooling process which is 10% more difficult to perform than one with a rating factor of unity.

Figure 46. Variation of Organic Condenser Length (Surface Area) With Pinch Temperature and Cooling Water Inlet Temperature

The number of tower units is primarily a function of and directly proportional to the approach temperature, i.e., the difference between the final cooling water temperature T_{w1} and the sink temperature which is represented here by the wet bulb. Obviously, as the cooling water temperature approaches the sink temperature, the number of tower units required becomes infinite. As T_{w1} increases, the number of tower units decreases. The decrease is rapid initially but tends to level off when the approach temperature exceeds 20 F or so. Three different curves are presented for different condenser pinch temperatures. For a given approach, an increase in the pinch temperature produces an increase in the mass of water circulated which in turn increases the number of tower units required as shown.

Ideally, the design of the condensing system as a whole should be evaluated using optimum values for the cooling water and condenser pinch temperatures T_{w1} and T_{pc} . However, since types of materials and relative costs are not yet established it is convenient to base designs on some operating point reasonably close to the optimum and extrapolate the calculated

data to conditions of interest later. For this reason, optimization was not attempted at this point and designs were based initially on values of T_{w1} and T_{pc} of 90 F and 10 F, respectively.

Figure 47. Effect of Cooling Water Inlet Temperature and Condenser Pinch Temperature on Cooling Water Circulating Pump Power Requirements and Condenser Tube Side Pressure Drop

To minimize length and facilitate admission of the organic fluid, the condenser design calculations were based on a shell diameter of 13 ft - the maximum value allowed because of manufacturing and shipping limitations. If a single condenser were to accommodate the exhaust flow, its length would be some 60 ft based on an averaged temperature difference ΔT_{ay} for the desuperheating and condensing process combined. This is considered excessive and two units are therefore, recommended. The results of the design calculations for each unit are summarized in Table 18.

Clearly, because the shell size is so large and the flow is split in two, the tube side pressure drop is relatively small.

Figure 48. Effect of Cooling Water Inlet Temperature and Condenser Pinch Temperature on Cooling Tower Units

Also, the water velocity within the tubes is well below typical limits set by manufacturers because of erosion considerations.* Because the shell side fluid is not required to negotiate baffles, the shell side pressure drop is negligible. However, the pressure drop in the supply manifold from the turbine exhaust will be significant. The absolute value will, of course, depend on the manifold's size and configuration, but to be consistent with the assumptions listed in Table 12, it should not exceed 5% of the exhaust pressure level.

*7 ft/sec for Cu/Ni alloys and 9 ft/sec for stainless steels.

Table 18

ORGANIC FLUID CONDENSERS

No. Units	Brine Temp. (°F)	Organic Fluid	L (ft)	D (ft)	N _t	Duty (10 ⁶ Btu/hr)	G _t (lb/hr)	G _s (ft ²)	Re _t	Re _s	f _t x10 ⁴	f _s x10 ³	ΔT (°F)	h _t (Btu/(hr ft ² °F))	h _s (ft ² °F)	U _c	U _d	f _{foul}
2	400	Isobutane	34.0	13.0	17700	655	999000	33300	26600	99900	2.14	1.39	20.6	1188	140	120.4	106.6	0.0011
2	500	N-butane	29.0	13.0	17700	552	853000	30400	22700	93700	2.22	1.41	20.5	1054	150	119.0	105.6	0.0011

No. Units	Brine Temp. (°F)	Organic Fluid	HOT FLUID (SHELL SIDE)					COLD FLUID (TUBE SIDE)					ΔT (°F)	Duty (10 ⁶ Btu/hr)
			T _{in} (°F)	T _{out}	P _{in} (psia)	P _{out}	$\dot{m}$ (10 ⁶ lb/hr)	T _{in} (°F)	T _{out}	ΔP (psia)	$\dot{m}$ (10 ⁶ lb/hr)			
2	400	Isobutane	161.7	117.0	103.2	103.0	4.36	90.0	113.0	3.64	28.5	20.6	655	
2	500	N-butane	160.9	117.0	73.2	73.0	3.40	90.0	112.7	2.38	24.3	20.5	552	

Note: All Tubes Wolverine 'Trufin' 3/4" OD x 16 BWG, 15/16 Triangular Pitch
19 Fins/Inch. Outside Area = 0.496 ft²/lin foot. Fin Height = 0.0625

Configuration: Shell Side Cross Flow, Tube Bundle Single Pass

Section 6

TURBINE DESIGNS

All turbines are of axial flow design with nominal net outputs of 50 MW_e. The designs and cost estimates were furnished by GE's Medium Steam Turbine Department.

A. THE ORGANIC TURBINES

Because the exhaust flow is relatively dense, both organic turbines are fairly compact and can be constructed as single flow units without exceeding manufacturing or shipping limitations in the size of the last stage. However, since two condensers are recommended in each cycle, it is debatable at this point as to whether a single flow unit is the best arrangement, since a double flow machine would lend itself more readily to dividing the exhaust flow in two. To resolve this uncertainty, a comparison of the costs of single and double flow units has been made in the case of the isobutane cycle.

Outline drawings of the single and double flow isobutane units and the single flow N-butane unit are presented as Figures 49, 50 and 51. The single inlet lines employed in each case and the single and dual exhausts were sized to limit the inlet and exhaust velocities to 75 ft/sec and 150 ft/sec, respectively. The absence of foreign matter in the closed cycle eliminates the need for a strainer ahead of the stop valve and modulating spool valve. In estimating turbine performance, the inlet pressure loss (between the stop valve and first-stage nozzle diaphragm) and exhaust loss were assumed to be 4 and 0.6 velocity heads, respectively. Because of the very short energy range of these turbines, emphasis is placed on reducing the exhaust loss to a minimum. The inlet loss is less important since increased pressure upstream of the turbine is accommodated by relatively little work in the compression of liquid by the feed pump.

Turbines were designed with approximately equal energy per stage. Individual stages were designed with subsonic nozzle spouting velocities and low reaction at the blade root. Also, root bending stress due to aerodynamic loading was limited to 5000 psi in each case and velocity ratios were optimized to give a slightly forward running velocity to the gas leaving the stage. Blade profiles, bucket bending stress limitations and materials were assumed to be the same as currently used in conventional steam turbine practice*. A summary of operating conditions, stage dimensions and estimated price of each organic turbine is given in Table 19. The net output of each cycle is

*Materials: Nozzles and rings - 405 stainless steel
 buckets - 403 stainless steel
 rotors - Chrome/Molybdenum/Vanadium alloy
 or nickel/chrome/moly./vanadium alloy

Figure 49. Outline Drawing - Single Flow Isobutane Turbine

Figure 50. Outline Drawing - Double Flow Isobutane Turbine

Figure 51. Outline Drawing - Single Flow N-butane Turbine

Table 19

SUMMARY OF TURBINE DETAILS

WORKING FLUID	ISOBUTANE			N-BUTANE			STEAM				
Inlet Conditions* P(psia)/T(°F)/M(10 ⁶ lb/hr)	530/285/8.812			540/310/6.900			67/300/0.6588 ^{††} 11.5/200/0.5479				
Exhaust Pressure (psia)	103.0			73.0			1.693				
Speed (RPM)	1800			1800			3600				
Output (kW)	58927			57328			50000				
Turbine Efficiency [†]	0.845			0.845			0.817, 0.736 ^{††}				
Configuration	<u>Single & Double Flow</u>			<u>Single Flow</u>			<u>Four Flow (Two Double Flow Units)</u>				
Dimensions ^x (INS)											
Stage	<u>1</u>	<u>2</u>	<u>3</u>	<u>1</u>	<u>2</u>	<u>3</u>	<u>1</u>	<u>2</u>	<u>3</u>	<u>4</u>	<u>5</u>
IRD	40(42.6)	40(42.6)	40(42.6)	44.4	44.4	44.4	46.8	46.8	46.8	46.8	46.8
NMH	5.10(2.52)	8.80(4.50)	13.70(7.40)	3.96	7.20	11.80	1.75	2.81	4.40	11.35	16.48
H _B	5.20(2.58)	8.92(4.60)	13.90(7.50)	4.09	7.32	12.00	1.84	2.90	4.53	11.70	16.80
W _B	3.0(1.45)	3.8(1.90)	4.5(2.34)	2.40	3.10	3.80	0.93	0.93	0.93	3.00	4.25
Cost**	\$ 4,950,000 (5,525,000)			\$ 5,250,000			\$ 10,475,000				

* Measured at Inlet to Strainer and/or Valve System

** Including Generator and Controls

† Includes Throttling Loss Through Valve System (and Strainer for Steam)

†† Values Given for High and Low Pressure Steam Flows Respectively

x Numbers in Parentheses Refer to Double Flow Unit

IRD = Inner Ring Diameter

NMH = Nozzle Mouth Height

H_B = Blade HeightW_B = Blade Width (at Root)

Generator Efficiency = 0.98

Organic Circulating Pump Motor Efficiency = 0.98

Organic Circulating Pump Efficiency = 0.85

50 MW. The outputs quoted include the circulating pump power requirements for which the motor and pump efficiencies were assumed to be 0.98 and 0.85, respectively. The generator efficiency was assumed to be 0.98 and the turbine efficiency in each case was estimated to be 0.845.

The turbine/generator prices are estimated prices assuming delivery in 1978 of a "mature product" and do not include initial development costs. The prices include the turbines, generators, and normal accessory systems such as controls, protective valving, lubrication system, exciter and generator hydrogen control system. The price differential of \$575,000 between the single and double flow isobutane units recognizes the doubling of the number of stages, halving of the airfoil dimensions, increasing the turbine bearing span and larger casing to accommodate the dual exhaust.

The N-butane turbine is similar in configuration to the single flow isobutane turbine, but has a somewhat larger exhaust casing. The estimated price differential between the two of \$300,000 reflects the larger casing and greater weight of the N-butane turbine.

B. THE STEAM TURBINE

The steam turbine is a four flow design comprising two double flow units. Each flow path has five stages with an admission for secondary steam between the third and fourth stage. Details of the turbine layout are shown in Figure 52.

All piping is sized to the nearest commercially available diameters which limit steam velocities to 200 ft/sec. The primary steam is supplied through two 24" diameter manifolds and introduced on the center line of each double flow unit through two 18" diameter inlets. The secondary steam is supplied through four 36" diameter lines - one for each admission in the four flow unit. This is the largest diameter for which control valves are currently manufactured. To avoid turbine damage from foreign matter, each of the six supply lines is equipped with a strainer ahead of the valve system comprised of a stop valve and modulating (control) valve, both of which are of butterfly type, independently controlled.

In calculating turbine performance, a pressure drop of 10% was assumed from the supply connection through the strainer and valve system, to the nozzle inlet with rated steam conditions existing at the supply connection. High and low pressure expansion efficiencies were estimated from past operating experience to be 0.817 and 0.736, respectively. These estimates include the effects of moisture, profile, and tip leakage loss and the throttling loss through the strainer and valve system in both the high and low pressure inlets. The low pressure efficiency also includes an exhaust loss which is the sum of the exhaust kinetic energy, the windage loss in the last stage and the

Figure 52A. Outline Drawing - Four Flow Steam Turbine

pressure loss in the hood. The above efficiencies do not include the generator loss which combined with the turbine bearing loss amounts to 963 kW. A relatively minor steam flow of 2800 lbs/hr bled from the high pressure inlet is required for the labyrinth shaft seals. A summary of operating conditions, stage dimensions and estimated cost is given for the steam turbine in Table 19.

Due to the steam turbines' short energy range, equal emphasis is placed on minimizing both the inlet and exhaust pressure loss. This contrasts with the organic Rankine cycle where the exhaust loss has greater significance since increased pressure upstream of the turbine is accommodated by relatively little work in the compression of liquid by the feed pump. Clearly, when sizing inlet piping and associated components, a compromise must be made between pressure loss and equipment cost. The inlet pressure loss of 10% assumed for this case corresponds to a reduction of approximately 2 1/2% in turbine output. The exhaust loss was limited to 7 1/2% of the energy drop in the turbine. This corresponds to an exhaust mass velocity of about 10,000 lb/ft²/hr for the four flow unit described. If a single double flow unit were used instead and the pitch diameter and blade height of the last stage increased to the practical limits of 72" and 24" respectively, the exhaust mass velocity would increase to 16,000 lbs/ft²/hr which is considered too high for this application.

The four flow steam turbine-generator estimated price for 1978 delivery is approximately \$200/kW. The price of this unit is significantly higher than that of the organic turbines due to its much larger physical size, and the additional cost of the six very large inlets with associated valves, piping and steam strainers. The price is approximately 25 percent higher than if it were a single pressure unit. This difference results from longer bearing spans as well as the addition of the low pressure inlets.

Details of a 4-flow dual admission 50 MW geothermal steam turbine are shown in Figure 52B. The unit was designed for slightly different operating conditions to those described here. However, each unit comprises five stages with admission for secondary steam occurring in this case between the second and third stage. The span between bearings in each double flow unit is 155 inches.

Figure 52B. Detail Drawing - Four Flow Steam Turbine

Section 7

PLANT DESIGNS AND COST ESTIMATES

The following plant designs and estimates of cost were formulated by the Projects Engineering Operation of GE's Installation and Service Engineering Division. Only the power island equipment is considered and well costs and the cost of steam generators, scrubbers and piping to the plant boundary are therefore excluded from all estimates.

A. PLANT ARRANGEMENTS

The general arrangement plans for the N-butane, isobutane and steam turbine cycles are shown in Figures 53, 54, and 55. In each case, the turbine generator set and mechanical auxiliary equipment are housed in a steel structure with roof only, and occupy the center of the diagram. The control building and major electrical equipment are located in the upper right corner and the cooling pond and cooling tower are at the far left. Each plot (excluding the cooling pond) occupies an area approximately 500 ft long by 350 ft wide. Note that two primary and secondary steam condensers are employed in each organic cycle when single units were specified earlier. The change was made on the advice of a major supplier to reduce shell costs and to make the system less expensive overall. Note also that a single flow turbine is employed in each organic cycle.

Details of pipe dimensions, pump capacities, fluid conditions and flow rates can be seen in the flow diagram for each cycle in Figures 56, 57 and 58. Four of the five main feed and booster pumps shown in the organic cycles operate continuously. The fifth is included as a spare. Similarly, one of the three circulating water pumps shown in all cycles is also a spare. Holding tanks for liquid nitrogen and carbon dioxide are required for purging the main cycle of hydrocarbon gas and the generator of hydrogen during shutdown. The symbols S and M appearing with valves refer to operation by solenoid and motor, respectively.

Line diagrams of major electrical equipment are given in Figures 59, 60 and 61. The utility transmission line operates at 69 kV and each turbine generator supplies the main step-up transformer at 13.8 kV. During cycle startup, power to the 13.8 kV bus duct is back fed through this transformer. All electrical equipment is supplied via two service transformers, each capable of delivering the total full load current of all electrical auxiliaries combined. During cycle operation, the 2000A circuit breaker located centrally on the 4.16 kV bus remains open, but closes in the event that either service transformer is lost. Two substation transformers connect the 4.16 kV bus duct to the

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LEGEND

1. ISOBUTANE STORAGE TANK
2. ISOBUTANE TRANSFER PUMP
3. WATER STORAGE TANK
4. COOLING WATER RECIRCULATION PUMP
5. COOLING TOWER
6. CONDENSER
7. MAIN FEED BOOSTER PUMP
8. MAIN FEED PUMP
9. SECONDARY COOLER
10. SECONDARY SUB COOLER
11. PRIMARY COOLER
12. PRIMARY SUB COOLER
13. LUBE OIL STORAGE TANK
14. TURBINE GENERATOR UNIT
15. TURBINE EXHAUST CONTROL
16. TURBINE EXHAUST CONTROL
17. CONTROL BUILDING
18. HIGH VOLTAGE BEND AND STRUCTURE
19. MAIN STEP UP TRANSFORMER
20. 13.8 KV SWITCHGEAR (PROTECTED AREA)
21. RESERVE TRANSFORMER
22. STATION SERVICE TRANSFORMER
23. LUBE OIL TRANSFORMER
24. STATION BATTERY
25. MOTOR CONTROL CENTER

Figure 54. Equipment Arrangement Drawing -
400 F Brine/Dual Flash Isobutane Cycle

<p>DESIGNED BY: [Name] CHECKED BY: [Name] DATE: [Date]</p>				<p>EQUIPMENT ARRANGEMENT 400 F BRINE / ISOBUTANE BINARY CYCLE GEOTHERMAL ENERGY STUDY CORPORATE R&D</p>		<p>GENERAL ELECTRIC PROJECTS ENGINEERING OPERATIONS 14-40 X16-76F-000 0</p>	
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Figure 55. Equipment Arrangement Drawing -
400 F Brine/Dual Flash Steam
Turbine Cycle

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Figure 56. Flow Diagram - 500 F Brine/Dual Flash N-butane Cycle

Figure 58. Flow Diagram - 400 F Brine/Dual Flash Steam Turbine Cycle

Figure 59. Electrical Line Diagram - 500 F
Brine/Dual Flash N-butane Cycle

Figure 61. Electrical Line Diagram - 400 F
Brine/Dual Flash Steam Turbine
Cycle

480V switchgear supplying cooling tower fans, feed booster pumps, motor control centers and the station battery. Further details of mechanical and electrical equipment are given in the balance of the plant equipment list in Table 26.

B. COMPONENT CONSTRUCTION DETAILS

Construction details of heat exchanger equipment and a description of the major balance of plant components are given below.

a. PRIMARY HEAT EXCHANGERS

The primary heat exchanger in each organic cycle consists of two units each 54 ft long by 8.5 ft and 9 ft diameter in the N-butane and isobutane cycles, respectively. Units are two pass U tube construction with floating tube sheets and internal heads. The units are all welded with manways for access and maintenance. Organic fluid is on the tube side. Tube material is 316 type stainless steel while all other material is carbon steel.

b. SECONDARY HEAT EXCHANGERS

These designs are similar to the primary heat exchangers. Again, two units are required, each unit being 34 ft long x 8 ft diameter in the N-butane cycle compared to 40 ft x 9 ft in the isobutane cycle.

c. PRIMARY AND SECONDARY SUBCOOLERS

These are single units of straight tube (single pass) design with floating tube sheets and internal heads. The units are all welded construction with manways for access and maintenance. Tubes which contain organic fluid are 1" diameter 316 type stainless steel. All other materials are carbon steel. Each primary subcooler is 7 1/2 ft diameter. Lengths are 10 ft and 11 ft for the isobutane and N-butane cycles, respectively. Each secondary subcooler is 8 ft diameter x 22.5 ft long.

d. ORGANIC CONDENSERS

In each organic cycle, the condenser consists of two units, each 13 ft diameter. The lengths are 34 ft and 29 ft in the isobutane and N-butane cycles, respectively. Cooling water is on the shell side and the hydrocarbon fluid on the tube side. The units are all welded construction with manways for access and maintenance. Units are straight tube design (single pass) with floating tube sheets and internal heads. Tubes are admiralty brass and finned according to the specifications listed in Table 18. All other material is carbon steel.

e. STEAM TURBINE CONDENSER

The steam condenser is a conventional 2 pass unit 26 ft long by 18 ft wide by 22 ft high, providing 8700 ft² of heat transfer

area. The unit will contain type 316 stainless steel 16,210 tubes, 1" diameter and will be designed for 80% cleanliness. Body and tube sheets are also 316 type stainless steel. Vacuum pumps are provided for leakage air and H₂S removal.

f. COOLING TOWERS

All cooling towers are cross flow, mechanical induced, draft design assembled with Douglas fir with PVC fill, draft eliminators and fiberglass manifolds. Erection is provided by the manufacturer. All towers are designed for a water inlet temperature of 113 F and an outlet temperature of 90 F. Six cells each with a 150 hp fan are required for the N-butane cycle; 7 cells are required for the other cycles. The total circulating flow is 103,000 gpm, 115,000 gpm and 125,000 gpm in the N-butane, isobutane and steam turbine cycles, respectively.

g. CIRCULATING WATER PUMP

The circulating water pumps are vertically mounted and installed in a basin having a minimum water level depth of 8 ft. Each unit includes a suction strainer and is designed for a differential pressure of 40 psi. In the N-butane cycle, each unit is driven by a 1000 hp motor and is designed for a flow of 51,500 gpm. In the isobutane and steam turbine cycles, the motor capacities are 1200 hp and flows 57,500 gpm and 62,500 gpm, respectively.

h. MAIN FEED PUMPS

The main organic feed pumps are 2 stage axially split, horizontal, centrifugal designs. The case is steel with cast iron impellers and wearings and SAE 4140 HT shafts. The units are designed to deliver 6100 gpm and 7800 gpm of flow at differential pressures of 456 psi and 428 psi for the N-butane and isobutane cycles, respectively.

i. MAIN FEED BOOSTER PUMPS

Each booster pump is vertically mounted with steel case, cast iron impeller and wearings and a 4160 HT steel shaft. The pumps are designed for the same flows as the main feed pumps with a differential pressure of 24 psi.

j. MAIN FEED PUMP MOTOR

The main feed pump motors are 4160 V, 3600 RPM drip-proof class B insulation units designed for outdoor operation with a 50 F ambient temperature rise. Power ratings are 2000 hp and 2500 hp for the N-butane and isobutane cycles, respectively.

C. CYCLE PERFORMANCE AND PLANT COSTS

The total auxiliary power requirements and the net plant output of each cycle is shown in Table 20. In each case, the gross electrical output from the turbine minus the electrical requirements of the feed pump (which is zero in the case of the steam turbine cycle) is equal to 50 MW_e.

The plant costs were synthesized from the costs of major components, balance of plant materials and balance of plant labor. From the balance of plant equipment list and the list of major items, a detailed breakdown was obtained of plant labor and material costs. These were added to the major equipment costs and indirect field costs. To this sum total was added a project management fee, a contingency to account for items not included in the original appraisal, and a profit. Finally, a factor was added to account for escalation and interest during the 18 months assumed for the construction period of the plant. The various assumptions used in developing the cost estimates are listed in Tables 21 and 22.

a. MAJOR COMPONENT AND BALANCE OF PLANT COSTS

Major component and subsystem costs for each cycle are summarized in Tables 23, 24 and 25. Also shown is the direct site labor costs for installation services. Clearly, the total

Table 20

SYSTEM OUTPUT AND

AUXILIARY LOSS BREAKDOWN (MW_e)

ITEM	ASSUMPTIONS	CYCLE		
		500 F BRINE/ DUAL FLASH N-BUTANE	400 F BRINE/ DUAL FLASH ISOBUTANE	400 F BRINE/ DUAL FLASH STEAM
Main Feed Pump	Pump Effy. 85% Motor Effy. 92%	7.328	8.927	-
Cooling Water Pumps	Proportional to Heat Duty	1.552	1.856	1.688
Cooling Tower I.D. Fans	Proportional to Heat Duty	0.666	0.789	0.789
Transformers	0.5% of Gross Power	0.287	0.295	0.250
"Hotel" Loads	A/E Estimate	0.200	0.200	0.200
Total Aux. Power		10.033	12.067	2.927
	Gross Cycle Output	57.328	58.927	50.000
	Net Power Plant Output (60 Hz, 69 kV)	<u>47.295</u>	<u>46.860</u>	<u>47.073</u>

Table 21

LIST OF ASSUMPTIONS FOR DEVELOPING COST ESTIMATES

- 1 A composite wage of \$13.8/hour was used for mechanical trades (This is a mean of current wages in the Los Angeles/San Diego area. See Table 22)
- 2 A composite wage of \$12.75/hr was used for civil and structural work
- 3 No rock excavation or dewatering is provided in estimate and no provision is made for site drainage
- 4 Bearing capacity of soil is assumed to be greater than 2000 psf and all foundations are of spread footing type without pilings
- 5 Turbine-generator and other mechanical auxiliary equipment is assumed to be shipped preassembled and received within 2 mile radius
- 6 All pipe is carbon steel, std wt on schedule 40 except circulating water pipe which is concrete
- 7 Large size pipe, except circulating water pipe, is above ground and supported on adjustable pipe roll stand - No walkways are provided above pipe
- 8 All insulation is 2" thick calcium silicate block with a 0.016" Al waterproof jacket
- 9 Main power grid at 69 kV
- 10 Project duration is 18 months and estimate (which does not include engineering costs) is in 1976 dollars

Table 22

LABOR WAGES FOR LOS ANGELES/SAN DIEGO AREA

		Hourly Wage	Fraction of Composite Wage	Elements Of Composite Wage
1	Boiler Makers	13.80	0.05	0.69
2	Millwrights	11.97	0.20	2.39
3	Pipe Fitters	15.38	0.26	4.00
4	Iron Workers	14.37	0.20	2.87
5	Electricians	13.41	0.27	3.62
6	Laborers	10.06	0.02	0.20

Composite Wage \$13.80/hr

Table 23

**500 F BRINE/DUAL FLASH N-BUTANE CYCLE:
MAJOR COMPONENT AND SUBSYSTEM CAPITAL COST SUMMARY**

MAJOR COMPONENT OR SUBSYSTEM	NO. UNITS	COST/UNIT (FOB)	COMPONENT OR SUBSYSTEM COST (FOB)	BOP MAT'L.	DIRECT SITE LABOR	SUBCONTRACT	TOTAL
Turbine/Gen.	1	5250	5250	2	126	-	5378
Organic Cond.	2	1000	2000	-	171	-	2171
Prim. St. Cond.	2	1150	2300	-	97	-	2397
Second. St. Cond.	2	890	1780	-	44	-	1824
Prim. Subcooler	1	380	380	-	5	-	385
Second. Subcooler	1	630	630	-	16	-	
Cooling Towers	6			-		824	824
Pumps	-			1652	73	-	1725
Piping, etc.	-			1014	219	92	1325

Table 24

**400 F BRINE/DUAL FLASH ISOBUTANE CYCLE:
MAJOR COMPONENT AND SUBSYSTEM CAPITAL COST SUMMARY**

MAJOR COMPONENT OR SUBSYSTEM	NO. UNITS	COST/UNIT (FOB)	COMPONENT OR SUBSYSTEM COST (FOB)	BOP MAT'L.	DIRECT SITE LABOR	SUBCONTRACT	TOTAL
Turbine/Gen.	1	4950	4950	2	126	-	5078
Organic Cond.	2	1100	2200	-	185	-	2185
Prim. St. Cond.	2	1260	2520	-	97	-	2617
Second. St. Cond.	2	950	1900	-	44	-	1944
Prim. Subcooler	1	380	380	-	5	-	385
Second. Subcooler	1	630	630	-	16	-	646
Cooling Towers	7	-	-	-	-	960	960
Pumps	-	-	-	1767	73	-	1840
Piping, etc.	-	-	-	1552	230	108	1890

Table 25

**400 F BRINE/DUAL FLASH STEAM TURBINE CYCLE:
MAJOR COMPONENT AND SUBSYSTEM CAPITAL COST SUMMARY**

COSTS (thousands of dollars)							
MAJOR COMPONENT OR SUBSYSTEM	NO. UNITS	COST/UNIT (FOB)	COMPONENT OR SUBSYSTEM COST (FOB)	BOP MAT'L.	DIRECT SITE LABOR	SUBCONTRACT	TOTAL
Turbine/Gen.	1	10475	10475	2	126	-	10603
Cond.	1	870	870	-	139	-	1009
Cooling Towers	7	-	-	-	-	960	960
Pumps	-	-	-	1074	41	-	1115
Piping, etc.	-	-	-	1036	225	100	1361

cost of the turbine/generator and its associated heat exchanger equipment is essentially the same for all cycles. In other words, the cost of the heat exchangers in both organic cycles offsets the savings in turbine costs.

Specifications for balance of plant equipment are summarized in Table 26 as prepared by the architect-engineer (PEO). All specifications are based on continuous operation at the valves wide open (VWO) condition for the turbine flow rates.

Tables 27, 28 and 29 represent the architect-engineer's detailed breakdown of the direct manual field labor in thousands of man hours (MH 1000's) and of balance of plant costs in thousands of dollars (\$1000's) for each major category of the balance of plant for each cycle. Average hourly field labor rates of \$13.8 and \$12.75 were used for mechanical trades and for civil and structural work, respectively, to convert man hours to dollars. These average wages are composites made up of annual wages in the Los Angeles/San Diego area as shown in Table 21.

Table 26

BALANCE OF PLANT EQUIPMENT LIST

<u>EQPT NO.</u>	<u>SERVICE</u>	<u>DESCRIPTION</u>
<u>1.0 ELECTRICAL SYSTEMS</u>		
E1	<u>GENERATOR</u>	67,450 kVA, 1800 RPM ⁽¹⁾ / 69,325 kVA, 1800 RPM ⁽²⁾ / 59,000 kVA, 3600 RPM ⁽³⁾ , * 0.85 pf, 13.8 kV, 3 ϕ , 60 Hz, 30 lb hydrogen cooled, class B insulation. Stator and field maximum temperatures 100 C and 125 C resp. Internal coolers have 90-10 Cu-Ni tubes and iron or steel water boxes for 95 F and 125 psig max water temperature and pressure.
E2	<u>GENERATOR STEP-UP TRANSFORMER</u>	56,000 kVA ⁽¹⁾ / 55,000 kVA ⁽²⁾ / 55,000 kVA ⁽³⁾ , forced oil to air, 55c, 69 kV/13.8 kV transformer has standard no load taps 2-2 1/2% above and 2-2 1/2% below rated 13.8 kV, bushing CTS (6), 69 kV surge arresters (60 kV rating) (3), transformer low voltage bus duct flange (13.8 kV wdq), winding temperature equipment, and fault pressure relay.
E3	<u>STATION SERVICE TRANSFORMERS (2)</u>	10,000/13,333 kVA ⁽¹⁾ / 12,000/ 15,960 kVA ⁽²⁾ / 3200/4000 kVA ⁽³⁾ , OA/FA, 55c, 13.8/ 4.16 kV transformer has standard no load taps 2-2 1/2% above and 2-2 1/2% below rated 4.16 kV, bush- ing CTS (6), bus duct flanges for both 13.8 kV and 4.16 kV connections, winding temperature equip- ment, and fault pressure relay.

*Suffixes (1), (2) & (3) refer to N-butane, isobutane and steam turbine cycles respectively.

Table 26 (Cont'd)

<u>EQPT NO.</u>	<u>SERVICE</u>	<u>DESCRIPTION</u>
E4	<u>STATION RESERVE TRANSFORMER</u>	10,000/13,333 kVA ⁽¹⁾ /12,000/ 15,960 kVA ⁽²⁾ /3200/4000 kVA ⁽³⁾ , OA/FA, 55c, 34.5/ 4.16 kV transformer has standard no load taps 2-2 1/2% above and 2-2 1/2% below rated 4.16 kV, bush- ing CTS (6), bus duct flange for 4.16 kV con- nection, winding temper- ature equipment, fault pressure relay, and equipped with 34.5 kV surge arrestors (rated 30 kV).
E5	<u>OIL CIRCUIT BREAKER, MAIN 69 kV</u>	Outdoor power circuit breaker type FK; 3-tanks, oil filled bushing, oil blast interrupters. Trip mechanism is mechanically and electrically trip free, 5 cycle interrupt- ing time, rated 72.5 kV# RMS max 2000 AMP RMS continuous, 27 kA RMS max. S.C. current.
E6	<u>OIL CURRENT BREAKER</u>	Same as above, rated 38 kV RMS max, 2000 AMP continuous, 36 kA RMS max S.C. current.
E7	<u>13.8 kV SWITCHGEAR (OUTDOOR)</u>	One line-up of vertical lift magne-blast metal clad switchgear rated 1000 MVA. Line-up in- cludes one generator breaker, 3000A, one step up transformer secondary breaker, 3000A; 2 station auxiliary transformers, 1200A; 3000 A aluminum bus; PT and metering com- partment. The line-up is constructed as a protected aisle outdoor assembly with circuit breaker transfer truck and test cabinet.

Table 26 (Cont'd)

<u>EQPT NO.</u>	<u>SERVICE</u>	<u>DESCRIPTION</u>
E8	<u>4.16 kV SWITCHGEAR</u> <u>(INDOOR)</u>	One line-up of vertical lift magne-blast metal clad switchgear rated 250 MVA. Line-up includes three transformer secondary breakers (2 aux, 1 reserve) 2000 A; one bus tie breaker, 2000 A; two transformer primary feeders, 1200 A; eight induction motor feeders 1200 A; 2000 A aluminum bus; PT and metering compartment; transfer truck and test cabinet.
E9	<u>REMOTE CONTROL PANEL</u> <u>(INDOOR)</u>	One panel line-up as necessary for remote control of outdoor equipment: 69 kV breaker, 34.5 kV breaker, 13.8 V metal clad switchgear and main step up transformer.
E10	<u>BUS DUCT</u>	<p>Approximately 20 ft of 3000A aluminum 13.8 kV outdoor bus duct for connection between 13.8 kV metalclad switchgear and 55 MVA step-up transformer, with two elbows and termination sections.</p> <p>Approximately 60 ft of 1200A aluminum, 13.8 kV outdoor bus duct for connections (2) between 13.8 kV metalclad switchgear and the two 12 MVA station auxiliary transformers, with four elbows and termination sections.</p> <p>Approximately 60 ft of 1200A aluminum, 4.16 kV outdoor bus duct for connections (2) between</p>

Table 26 (Cont'd)

<u>EQPT NO.</u>	<u>SERVICE</u>	<u>DESCRIPTION</u>
		three 12 MVA station auxiliary and reserve transformers and the indoor 4.16 kV metalclad switchgear, with 6 elbows and termination sections.
		Approximately 24 ft of 1200A aluminum 4.16 kV indoor bus duct for connections (2) between the indoor 4.16 kV metalclad switchgear and the 1500 kVA load center transformers.
E11	<u>LOAD CENTER UNIT</u> <u>SUBSTATIONS (2)</u>	<p>1000 kVA dry, load center transformer, 4160/480 V bus duct connection on primary and throat connected on secondary to line-up of 480 V switchgear. Transformer will have winding temperature equipment and low sound level treatment.</p> <p>480 V switchgear will be metal enclosed AKD-6 type of metal enclosed. Draw-out protective breakers. Included are the necessary equipment to control, switch and protect the connected electrical loads.</p> <p>The two substations include two (2) transformer transition compartments, two (2) transformer secondary breakers, two (2) motor control center feeders, seven (7) induction motor (150 hp) feeders (FVNR), one (1) crane feeder, and one battery charger feeder. Undervoltage and overcurrent protection on the</p>

Table 26 (Cont'd)

<u>EQPT NO.</u>	<u>SERVICE</u>	<u>DESCRIPTION</u>
		motor feeders, and load center metering.
E12	<u>MOTOR CONTROL CENTERS, 480 VAC (2)</u>	NEMA Class II type B motor control center in NEMA I enclosure, gasketed. 600 ampere aluminum main bus, (300 amp vertical). Each of the two control centers include 10-size 1, 5-size 2, 3-size 3 FVNR combination transformer and 36-circuit panel-board.
E13	<u>MOTOR CONTROL CENTER, 125 VDC</u>	NEMA Class II type B motor control center in NEMA I enclosure, gasketed, with 600 ampere aluminum bus. This includes 1 main breaker, 5-size 1 and 5-size 2 starters, 6 feeders - 50 A ea., 1-12 circuit panelboard-20A ea.
E14	<u>STATION BATTERY & CHARGER</u>	Switchgear control power source will be from a 125 VDC station battery and charger. Lead calcium battery of 120 cells with discharge rate of 240 amperes for one minute. This includes rack and connections between cells.
E15	<u>INTERCONNECTING CABLES</u>	One lot of cable including 15 kV grounded 3/C shielded with ground conductor in the interstices and requires 600 V cable included.

Table 26 (Cont'd)

<u>EQPT NO.</u>	<u>SERVICE</u>	<u>DESCRIPTION</u>
E16	<u>HIGH VOLTAGE TAKE-OFF STRUCTURE</u>	Two bay structure of not over 33 ft each with three towers and cross member provided with clevis for connection of incoming/outgoing 69 kV and 34.5 kV live insulators. Line tension not to exceed 2000 # each conductor and shield wire.
<u>2.0 MAIN FLUID SYSTEMS</u>		
F1	<u>PIPING:</u>	
	<u>STEAM SUPPLY FROM PRIMARY/SECONDARY FLASH</u>	<div> <div> 48"Ø x 310' / 40"Ø x 442' (1) 60"Ø x 310' / 40"Ø x 442' (2) 72"Ø x 400' / 54"Ø x 400' (3) </div> <div> } * All 0.375" Thick </div> </div>
	<u>ORGANIC CYCLE</u>	250' x 48" Ø (1) (x54" Ø (2)) 112' x 30" Ø (1) (x40" Ø (2)) 230' x 24" Ø (1) (x30" Ø (2)) 520' x 18" Ø (1) (x20" Ø (2)) 85' x 10" Ø (1) & (2) 25' x 6" Ø (1) (1) & (2) 300' x 4" Ø (1) (1) & (2)
	<u>COOLING WATER</u>	64" Ø x 650', 48" Ø x 310' (1) 72" Ø x 650', 54" Ø x 310' (2) 72" Ø x 900', 54" Ø x 150' (3)
	<u>CONDENSATE TO STORAGE POND</u>	240' x 10" Ø (1) (x 12" Ø (2)) 50' x 8" Ø (1) (x 10" Ø (2))
	<u>BYPASS TO CONDENSATE STORAGE POND</u>	260' x 10" Ø (1) (x 12" Ø (2)) 50' x 8" Ø (1) (x 10" Ø (2)) 140' x 6" Ø (1) (x 8" Ø (2))
	<u>LUBE OIL</u>	75' x 6" Ø (1) & (2) 75' x 4" Ø (1) & (2)
	<u>SMALL PIPE BELOW 2 1/2" Ø FOR CO₂ SYSTEM, COM- PRESSED AIR, DRAINS, PROCESS PIPING ETC.</u>	3500' (1) & (2)

Table 26 (Cont'd)

<u>EQPT NO.</u>	<u>SERVICE</u>	<u>DESCRIPTION</u>
F2	<u>PUMPS:</u>	
	<u>CIRCULATING WATER PUMP & MOTOR (3 REQ'D)</u>	51,500 gpm, 1000 hp, 40 psi TDH (1) 57,500 gpm, 1200 hp, 40 psi TDH (2) 62,500 gpm, 1200 hp, 40 psi TDH (3)
	<u>MAIN FEED PUMP & MOTOR (5 REQ'D)</u>	6100 gpm, 2000 hp, 456 psi TDH (1) 7800 gpm, 2500 hp, 428 psi TDH (2)
	<u>MAIN FEED BOOSTER PUMP (5 REQ'D)</u>	6100 gpm, 100 hp, 24 psi TDH (1) 7800 gpm, 150 hp, 22 psi TDH (2)
	<u>ORGANIC MAKEUP PUMP</u>	120 gpm, 10 hp, 30 psi TDH (1) 125 gpm, 10 hp, 30 psi TDH (2)
F3	<u>COOLING TOWERS</u>	6 cells/103,000 gpm (1) 7 cells/115,000 gpm (2) 7 cells/125,000 gpm (3) } 150 hp four per cell.

Table 27

BALANCE OF PLANT ESTIMATE DETAIL

500 F BRINE/DUAL FLASH N-BUTANE CYCLE

	<u>DIRECT MANUAL</u> <u>FIELD LABOR</u> <u>(MH 1000's)</u>	<u>BALANCE OF</u> <u>PLANT MATERIAL</u> <u>\$1000's</u>
1.0 <u>TURBINE/GENERATOR (1)</u>		
Install only (supply by others) includes turbine, generator, exciter; auxiliary equipment, integral steam and auxiliary piping, insulation and miscellaneous laborer operations	9.14	2
2.0 <u>MECHANICAL AUXILIARY EQUIPMENT</u>		
2.1 <u>Condenser (2)</u>	12.4	-
Install only (supply by others)		
2.2 <u>Primary Steam Condenser (2) Primary Subcooler (1); Secondary Condenser (2) Secondary Subcooler (1)</u>	11.8	-
Install only (supply by others)		
2.3 <u>Main Circ. Water Pumps (3)</u>	2.7	1044
Includes pump and motor (3 @ \$348,000 each)		
2.4 <u>Main Feed Pump (5)</u>	2.675	608
Includes pump and motor (5 @ \$103,350 each), feed booster pumps and motors (5 @ \$17,000 ea.) and makeup pump @ \$6100		

Table 27 (Cont'd)

	<u>DIRECT MANUAL FIELD LABOR (MH 1000's)</u>	<u>BALANCE OF PLANT MATERIAL \$1000's</u>
2.5 <u>N-Butane Storage Tank</u> (2 @ \$46,000)	0.5	98
25' $\emptyset$ and liquid drainer (5 @ \$1200)		
2.6 <u>Overhead Crane</u>	5	270.3
50/15 tons includes structural steel		
3.0 <u>PIPING</u>		
3.1 <u>Primary & Secondary Steam Lines</u>	2.15	70.5
Includes 48" $\emptyset$ x 310' and 40" $\emptyset$ x 442'		
3.2 <u>N-Butane Cycle</u>	3.34	83.2
Includes 48" $\emptyset$ x 250', 30" $\emptyset$ x 112', 24" $\emptyset$ x 230', 18" $\emptyset$ x 520', 6" $\emptyset$ x 25' and 4" $\emptyset$ x 300'		
3.3 <u>Condensate to Holding Pond</u>	0.5	10.0
Includes 10" $\emptyset$ x 240'; 8" $\emptyset$ x 50'		
3.4 <u>Inlet Bypass to Holding Pond</u>	0.76	12.0
Includes 10" $\emptyset$ x 260', 8" $\emptyset$ x 50', 6" $\emptyset$ x 140'		
3.5 <u>Lube Oil</u>	0.225	1.9
Includes 6" $\emptyset$ x 75', 4" $\emptyset$ x 75'		
3.6 <u>Valves</u>	-	747.9

Table 27 (Cont'd)

	<u>DIRECT MANUAL FIELD LABOR (MH 1000's)</u>	<u>BALANCE OF PLANT MATERIAL \$1000's</u>
3.7 <u>Small Pipe (Below 2 1/2" Ø),</u>	8.87	88.7
Includes instrumen- tation and miscel- laneous operations		
3.8 <u>Insulation</u>	subcontract	92
4.0 <u>ELECTRICAL</u>		
4.1 <u>Switchgear & Load Centers</u>	1.95	908
Includes 13.8 kV metalclad SWGR, 4.16 kV metalclad SWGR, low voltage SWGR Nos. 1 & 2		
4.2 <u>Oil Circuit Breakers</u>	0.175	60
69 kV and 34.5 kV OCB's		
4.3 <u>Bus Duct</u>	0.75	115
13.8 kV, 3000 amps 20'; 1200A, 60' 4.16 kV, 2000 amps 60'; 1200A, 24'		
4.4 <u>Motor Control Centers</u>	0.275	44
480 V MEC, DC motor control & distribution		
4.5 <u>Local Controls</u>	0.15	30
3 control panels		
4.6 <u>Transformers</u>	1.35	511
Includes step up transformer, station service transformer, station reserve trans- former, unit substation transformer		

Table 27 (Cont'd)

	<u>DIRECT MANUAL FIELD LABOR (MH 1000's)</u>	<u>BALANCE OF PLANT MATERIAL \$1000's</u>
4.7 <u>DC Equipment</u>	0.1	16
Station battery & charger		
4.8 <u>Wire & Cable</u>	13.99	153
Includes conduit cable, cable tray, communication system, outdoor lighting & power distribution		
5.0 <u>CIVIL & STRUCTURAL</u>		
5.1 <u>Earthwork in Exca- vation</u>	subcontract	59.65†
5.2 <u>Concrete</u>	29.5	289.1
5.3 <u>Control Room</u>	subcontract	123.2†
6.0 <u>YARDWORK</u>		
6.1 <u>Site Development</u>	subcontract	4495†
Site development, site furnishing, roads, fencing, etc.		
6.2 <u>Holding Pond</u>	subcontract	71.0†
128' x 500'		
6.3 <u>Cooling Towers</u>	subcontract	824.0†
6 units		
6.4 <u>Circulating Water Pipe</u>	5.14	58.18
64" Ø x 650' } concrete 48" Ø x 310' }		

†Balance of plant material costs plus subcontract labor cost =
total subcontract cost

Table 27 (Cont'd)

	<u>DIRECT MANUAL FIELD LABOR (MH 1000'S)</u>	<u>BALANCE OF PLANT MATERIAL \$1000'S</u>
6.5 <u>Valves</u>	with pipe	132.4
48" Ø check valve (3 req'd @ \$7,500 ea.)		
48" Ø butterfly valve (7 req'd @ \$15,700 ea.)		
54" Ø expansion joints and small valves		

Table 28

BALANCE OF PLANT ESTIMATE DETAIL

400 F BRINE/DUAL FLASH ISOBUTANE CYCLE

	<u>DIRECT MANUAL</u> <u>FIELD LABOR</u> <u>(MH 1000'S)</u>	<u>BALANCE OF</u> <u>PLANT MATERIAL</u> <u>\$1000'S</u>
1.0 <u>TURBINE/GENERATOR</u>	9.14	2
Install only (supply by others)		
includes turbine, generator, exciter, auxiliary equipment, integral steam and auxiliary piping, insulation and miscellaneous labor operations.		
2.0 <u>MECHANICAL AUXILIARY EQUIPMENT</u>		
2.1 <u>Condenser (2)</u>	13.4	-
Install only (supply by others)		
2.2 <u>Primary Steam Condenser (2); Primary Subcooler (1); Secondary Condenser (2); Secondary Subcooler (1)</u>	11.8	-
Install only (supply by others)		
2.3 <u>Main Circ. Water Pumps (3)</u>	2.7	1074
Includes pump and motor (3 @ \$358,000)		
2.4 <u>Main Feed Pump (5)</u>		
Includes pump and motor (5 @ \$117,600); feed booster pumps and motors (5 @ \$19,800) and makeup pump	2.575	693
2.5 <u>Isobutane Storage Tank (2 @ \$46,000)</u>	0.5	98
25' Ø and liquid drainer		

Table 28 (Cont'd)

	<u>DIRECT MANUAL FIELD LABOR (MH 1000'S)</u>	<u>BALANCE OF PLANT MATERIAL \$1000'S</u>
2.6 <u>Overhead Crane</u>		
50/15 tons - includes structural steel	5	270.3
3.0 <u>PIPING</u>		
3.1 <u>Primary & Secondary Steam Lines</u>	2.70	91.8
Includes 60" $\varnothing$ x 310' and 54" $\varnothing$ x 442'		
3.2 <u>Isobutane Cycle</u>		
Includes 54" $\varnothing$ x 250' 40" $\varnothing$ x 112', 30" $\varnothing$ x 230', 20" $\varnothing$ x 520', 6" $\varnothing$ x 25', 4" $\varnothing$ x 300'	3.30	89.55
3.3 <u>Condensate To Holding Pond</u>	0.57	10.1
Includes 12" $\varnothing$ x 240', 10" $\varnothing$ x 50'		
3.4 <u>Inlet Bypass to Holding Pond</u>	0.83	14.2
Includes 12" $\varnothing$ x 260' 10" $\varnothing$ x 50', 8" $\varnothing$ x 140'		
3.5 <u>Lube Oil</u>	0.225	1.9
Includes 6" $\varnothing$ x 75', 4" $\varnothing$ x 75'		
3.6 <u>Valves</u>	-	1245
3.7 <u>Small Pipe (Below 2 1/2" $\varnothing$)</u>	9.07	99.1
Includes instrumentation, and miscellaneous		
3.8 <u>Insulation</u>	subcontract	108

Table 28 (Cont'd)

	<u>DIRECT MANUAL FIELD LABOR (MH 1000'S)</u>	<u>BALANCE OF PLANT MATERIAL \$1000'S</u>
4.0 <u>ELECTRICAL</u>		
4.1 <u>Switchgear & Load Centers</u>	1.95	915
Includes 13.8 kV metal clad SWGR, 4.16 kV metal clad SWGR, low voltage SWGR's No. 1 & 2		
4.2 <u>Oil Circuit Breakers</u>	0.175	60
69 kV and 34.5 kV OCBR's		
4.3 <u>Bus Duct</u>		
13.8 kV, 3000 amp 20'; 1200 amp, 60'; 4.16 kV, 2000 amp 60'; 1200 amp, 24'	0.75	115
4.4 <u>Motor Control Centers</u>	0.275	44
480 V MCC, DC motor control & distrib.		
4.5 <u>Local Controls</u>	0.15	30
3 control panels		
4.6 <u>Transformers</u>	1.35	553
Includes step up transformer; station service transformer; station reserve transformer; unit substation transformer		
4.7 <u>DC Equipment</u>	0.1	16
Station battery and charger		
4.8 <u>Wired Cable</u>	13.99	154
Includes conduit, cable, cable tray communication system, outdoor light, and power distribution		

Table 28 (Cont'd)

	<u>DIRECT MANUAL FIELD LABOR (MH 1000'S)</u>	<u>BALANCE OF PLANT MATERIAL \$1000'S</u>
5.0 <u>CIVIL & STRUCTURAL</u>		
5.1 <u>Earthwork in Excavation</u>	subcontract	59.65†
5.2 <u>Concrete</u>	31.89	317.60
5.3 <u>Control Room</u>	subcontract	123.20†
6.0 <u>YARDWORK</u>		
6.1 <u>Site Development</u>	subcontract	44.95†
Site devel., site finishing, roads, fencing etc.		
6.2 <u>Holding Pond</u>	subcontract	71.0†
128 x 500'		
6.3 <u>Cooling Towers</u>	subcontract	960.54†
7 units		
6.4 <u>Circulating Water Pipe (Concrete)</u>	6.10	80.75
72" Ø x 650' 54" Ø x 310'		
6.5 <u>Valves</u>	(with pipe)	161.4
54" Ø check valve (3 req'd @ \$9,800 ea.) 54" Ø butterfly valve (7 req'd @ \$17,500 ea.) 54" Ø expansion joints, misc. flanges etc.		

†Balance of plant material costs plus subcontract labor cost =
total subcontract cost

Table 29

BALANCE OF PLANT ESTIMATE DETAIL

400 F BRINE/DUAL FLASH STEAM TURBINE CYCLE

	<u>DIRECT MANUAL FIELD LABOR (MH 1000'S)</u>	<u>BALANCE OF PLANT MATERIAL \$1000'S</u>
1.0 <u>TURBINE/GENERATOR</u>	9.14	2
Install only (supply by others) includes 50 MW turbine, gener- ator, exciter, auxiliary equip- ment, integral steam and aux. piping, insulation and miscellaneous labor operations		
2.0 <u>MECHANICAL AUXILIARY EQUIPMENT</u>		
2.1 <u>Condenser (1)</u>	10.05	-
Install only (supply by others)		
2.2 <u>Main Circulating Water Pumps (3)</u>	3.0	1074
Includes pump & motor (3 req'd @ \$358,000)		
2.3 <u>Overhead Crane</u>	5.0	270.3
50/15 tons - includes structural steel		
3.0 <u>PIPING & INSTRUMENTATION</u>		
3.1 <u>Primary & Secondary Steam Lines</u>	2.8	107.8
Includes 72" Ø x 400' and 54" Ø x 400'		
3.2 <u>Steam Cycle</u>	2.87	85.2
36" Ø x 800', 24" Ø x 196', 18" Ø x 150'		

Table 29 (Cont'd)

	<u>DIRECT MANUAL FIELD LABOR (MH 1000'S)</u>	<u>BALANCE OF PLANT MATERIAL \$1000'S</u>
3.3 <u>Inlet Bypass to Cooling Pond</u> 12" Ø x 260', 10" Ø x 50', 8" Ø x 140'	0.83	14.22
3.4 <u>Condensate to Cooling Pond</u> 12" Ø x 275'	0.50	16.7
3.5 <u>Lube Oil</u> 6" Ø x 75', 4" Ø x 75'	0.225	1.90
3.6 <u>Valves</u> 48" Ø ball steam (3 @ \$76,000 ea.) 72" Ø ball steam (3 @ \$150,000 ea.) 12" Ø check steam con- densate (\$1,700) 10" Ø butterfly steam condensate (3 @ \$1,300 ea.)	(with pipe)	683.6
3.7 <u>Hangers & Supports</u>	3.39	78.5
3.8 <u>Insulation</u>	subcontract	100.0†
3.9 <u>Instrumentation</u>	1.56	30.0
3.10 <u>Small Pipe (< 2 1/2" Ø)</u> Including small valves, painting, etc.	4.16	18.1
4.0 <u>ELECTRICAL</u>		
4.1 <u>Switchgear & Load Centers</u> Includes 13.8 kV metal clad SWGR, 4.16 kV metal clad SWGR, low voltage SWGR No. 1 & 2	1.65	730

Table 29. (Cont'd)

	<u>DIRECT MANUAL FIELD LABOR (MH 1000'S)</u>	<u>BALANCE OF PLANT MATERIAL \$1000'S</u>
4.2 <u>Oil Circuit Breakers</u> 69 kV & 34.5 kV OCBR's	0.175	60
4.3 <u>Bus Duct</u> 13.8 kV & 4.16 kV	0.75	102
4.4 <u>Motor Control Centers</u> 480 V MCC DC Motor control & distrib.	0.275	44
4.5 <u>Local Controls</u> 3 control panels	0.15	30
4.6 <u>Transformers</u> Includes step up transformer, station service transformer, station reserve transformer and unit substation trans- former	1.20	322
4.7 <u>DC Equipment</u> Station battery & charger	0.1	16
4.8 <u>Wire & Cable</u> Includes conduit, cable, cable tray, communication system, outdoor lighting, and power distribution	12.22	117.25
5.0 <u>CIVIL & STRUCTURAL</u>		
5.1 <u>Earthwork in Excavation</u>	subcontract	261.3
5.2 <u>Concrete</u>	30.36	261.3
5.3 <u>Control Building</u>	subcontract	123.20

Table 29 (Cont'd)

		<u>DIRECT MANUAL FIELD LABOR (MH 1000'S)</u>	<u>BALANCE OF PLANT MATERIAL \$1000'S</u>
6.0	<u>YARDWORK</u>		
6.1	<u>Site Development</u>	subcontract	41.65
	Site development, site finishing, roads, fencing etc.		
6.2	<u>Cooling Pond</u>	subcontract	71.0
	128 x 500'		
6.3	<u>Cooling Towers</u>	subcontract	960.54
	7 units		
6.4	<u>Circulating Water Pipe</u>	8.5	96.45
	(Concrete)		
	72" $\varnothing$ x 900', 54" $\varnothing$ x 150'		
6.5	<u>Valves</u>	with pipe	161.9
	54" $\varnothing$ check valve (3 @ \$9,800 ea.) 54" $\varnothing$ butter- fly valve (7 @ \$17,500 ea.) 54" $\varnothing$ expansion joints, misc. flanges, etc.		

b. PLANT COST ESTIMATES

The major component and balance of plant costs are combined in Tables 30, 31 and 32 and summarized in Table 33 and give the total plant costs for each cycle. Total direct costs include the total material cost, direct site labor and subcontract costs. Indirect field costs were applied to each of these categories to obtain the total field cost. A contingency of 10% of the direct costs was assumed along with a profit of 15% of the total field costs. The addition of these and a project arrangement cost to the total field costs gives the total capital cost. Finally, a factor of 0.125* was added to account for interest and escalation during the 18 month assumed construction period to obtain the total plant costs shown.

Table 30

PLANT CAPITAL COST BREAKDOWN

500 F BRINE/DUAL FLASH N-BUTANE CYCLE

CATEGORIES	MATERIALS	COST (thousands of dollars)		TOTAL
		DIRECT LABOR	SUBCONTRACT	
1 Turbine/Gen.	5252	126	-	5378
2 Mech. Aux. Equip.	9110	483	-	9593
3 Piping	1014	219	92	1325
4 Electrical	1837	259	-	2096
5 Civil and Structural	289	376	183	848
6 Yardwork	191	71	940	1202
Total Direct Costs	17693	1534	1215	20442
7 Field Indirect Costs	254	1090	54	1398
Total Field Cost	17947	2624	1269	21840
8 Project Management, Purchasing, Exped., Est. & Tech. Direc.			474	474
9 Contingency @ 10% of Direct Costs	1769	153	121	2043
10 Profit @ 15% of Total Field Costs	2692	393	190	3275
Total Capital Costs	22408	3170	2054	27632
11 Interest & Escalation During Construction				3454
TOTAL				31086
Plant Output (MWe)				47.296
TOTAL \$/kW				657.3

*Annual rates of 6.5% escalation and 10% interest were assumed.

Table 31

PLANT CAPITAL COST BREAKDOWN

400 F BRINE/DUAL FLASH ISOBUTANE CYCLE

CATEGORIES	COSTS (thousands of dollars)			
	MATERIALS	DIRECT LABOR	SUBCONTRACT	TOTAL
1 Turbine/Gen.	4952	126	-	5078
2 Mech. Aux. Equip.	9765	497	-	10262
3 Piping	1552	230	108	1890
4 Electrical	1887	259	-	2146
5 Civil and Structural	318	407	183	908
6 Yardwork	242	84	1076	1402
Total Direct Costs	18716	1603	1367	21686
7 Field Indirect Costs	254	1121	54	1429
Total Field Costs	18970	2724	1421	23115
8 Project Management, Purchasing, Exped., Est. & Tech. Direc.			513	513
9 Contingency @ 10% of Direct Costs	1872	160	137	2169
10 Profit @ 15% of Total Field Costs	2845	408	213	3466
Total Capital Costs	23687	3292	2284	29263
11 Interest & Escalation During Construction				3658
Total				32921
Plant Output (MWe)				46.86
Total \$/kW				702.5

The steam turbine cycle is clearly the least expensive of the three systems. It is over \$100/kW lower in cost than the isobutane cycle operating on the same resource and \$60/kW lower than the N-butane cycle at Niland. It must be borne in mind of course that if duplicate heat-exchangers are required to permit scale removal during plant operation, these cost differentials would each rise by about \$100/kW.

The costs of electricity shown were based on a fixed charge rate* of 18% of the total plant cost and a capacity factor of

*The fixed charge rate included cost of capital, depreciation, income tax and administration, insurance and ad valorem taxes.

0.65. Included in each cost is 1.8 mils estimated for operating labor and maintenance. The electricity costs do not include the fuel costs. Fuel costs will depend on several factors including well depths and the number of wells required to deliver the brine flows necessary to generate 50 MW_e. Clearly, from the specific outputs listed in Table 13, the steam turbine cycle requires the smallest brine flow, and if other factors are equal, will also have the lowest fuel cost.

Table 32

PLANT CAPITAL COST BREAKDOWN

400 F BRINE/DUAL FLASH STEAM TURBINE CYCLE

CATEGORIES	COST (thousands of dollars)			
	MATERIALS	DIRECT LABOR	SUBCONTRACT	TOTAL
1 Turbine/Gen.	10477	126	-	10603
2 Mech. Aux. Equip.	2214	249	-	2463
3 Piping	1036	225	100	1361
4 Electrical	1421	228	-	1649
5 Civil & Structural	261	387	178	826
6 Yardwork	258	117	1073	1448
Total Direct Costs	15667	1332	1351	18350
7 Field Indirect Costs	255	988	54	1297
Total Field Costs	15922	2320	1405	19647
8 Project Management, Purchasing, Exped., Est. & Tech. Direc.			379	379
9 Contingency @ 10% of Direct Costs	1567	133	135	1835
10 Profit @ 15% of Total Field Costs	2388	348	211	2947
Total Capital Costs	19877	2801	2130	24808
11 Escalation and Interest During Construction				3101
Total				27909
Plant Output (MWe)				47.073
Total \$/kW				592.9

Table 33

TOTAL PLANT COSTS (THOUSANDS 1976 DOLLARS)

	Dual Flash/N-Butane (500 F Brine)	Dual Flash/Isobutane (400 F Brine)	Dual Flash Steam Turbine (400 F Brine)
Turbine-Gen.	5378	5078	10603
Heat Exchangers	7423	7777	1009
Aux. Equip.	2170	2485	1454
Piping	1325	1890	1361
Elec. Equip.	2096	2146	1649
Civil. and Structural	848	908	826
Yardwork	1202	1402	1448
Total Direct Costs	<u>20442</u>	<u>21686</u>	<u>18350</u>
Field Indirect Costs	1398	1429	1297
Total Field Costs	<u>21840</u>	<u>23115</u>	<u>19647</u>
Project Management	474	513	379
Contingency (10% of Direct Costs)	2043	2169	1835
Profit @ 15% of Total Field Costs	3275	3466	2947
Total Cost	<u>27632</u>	<u>29263</u>	<u>24808</u>
Interest & Escalation During Construction	3454	3658	3101
Total	<u>31086</u>	<u>32921</u>	<u>27909</u>
Plant Output (MWe)	47.296	46.86	47.073
Total \$/kW	<u>657.3</u>	<u>702.5</u>	<u>592.9</u>
Cost of Electricity (mills/kWh) (excluding fuel cost)	22.6	24.0	20.5

If the cost of geothermal brine is independent of the brine flow but is instead tied to fossil fuel prices and determined by the equivalent fuel required to generate the same power in a conventional power plant, the 'geothermal' fuel cost will be $3.413 \times 10^{-3} C_o/\eta$ mills/kWh. C_o is the fossil fuel price in \$/million BTU and η is the overall power plant efficiency. If geothermal power were to compete with, for example 'clean' coal derived liquid fuels, estimated to cost between \$3 and \$4 per million BTU, the geothermal fuel cost will be the order of 27-36 mills/kWh, assuming a power plant efficiency of 38%. This would be the initial cost. If account is taken of price escalation and the time value of money, it can be shown that for a 10% interest rate and a 6.5% escalation rate, the average fuel cost over a 20 year period will be 1.65 times the initial value. This would increase to 1.94 over a 30 year period.

Section 8

GENERAL DISCUSSION AND SUGGESTIONS FOR FUTURE DEVELOPMENT WORK

The foregoing analysis and cost estimates apply of course to dual flash systems in which only separated steam is utilized in the energy conversion process. More power per unit of brine flow could be obtained with additional flash stages, but in the case of the steam turbine cycle, the high cost of valving and piping with increased specific volume and the added complexity introduced by additional steam admission points in the turbine would more than offset any gain in steam turbine output. To some extent the same is true in the case of the flashing binary cycle. Additional flash stages require additional heat exchangers which in the dual flash cycle contributed some \$7 million or about 20% of the total plant cost. Although some improvements could possibly be made in this area, it is interesting to note from Table 33 that even if the primary and secondary heat exchanger costs were eliminated completely, the cost of the isobutane power plant would still only be equivalent to that of the steam turbine cycle in this size and temperature range.

By considering only the power island equipment and cycles which operate on relatively clean geothermal steam, the major problems of scaling and corrosion in wells and field equipment are conveniently removed from consideration. However, while no major obstacles are anticipated in implementing the selected cycles, special attention should be given to material selection, (particularly in the heat exchanger equipment) and to the problem of sealing hydrocarbon gas in the organic turbine.

While the steam generated in each flashed system should be relatively free from salt, the presence of hydrogen sulphide and some minor ionic contamination due to inefficiencies in the separation and scrubbing processes are of concern with regard to corrosion and related phenomena such as corrosion, corrosion cracking and corrosion fatigue. Fortunately, the corrosion rates of conventional steam turbine alloys appear to be fairly low in all flashed systems currently producing power and where materials have been tested in fields under development.

The performance of several steam turbine and heat exchanger alloys has been investigated recently by GE's Medium Steam Turbine Department* at Union Oil's New Mexico field which is currently being evaluated as a source of power generation. Tests were run for a maximum exposure of 160 days on 110 psig, 340 F, flashed steam containing approximately 400 ppm of hydrogen sulphide and

*Presented in a Joint General Electric-Union Oil Paper "Metallurgical Evaluation of Materials for Geothermal Power Plant Applications," at the Fall Meeting of the Electrochemical Society, Las Vegas 1976.

an estimated chloride contamination of 10 ppm. Of the materials tested, titanium base alloys and high alloy wrought steels exhibited the most corrosion resistant behavior. Martensitic 12 Cr and Modified 12 Cr alloys behaved similarly, corroding at the relatively low rates of between 0.003 - 0.005 mm/year. Medium alloy turbine steels corroded at 4 times these rates with C-steel sustaining the greatest corrosion rate some 8 times greater than that of the 12 Cr steels. Similar results have been experienced with the same alloys in the Mexican field at Cerro Prieto. When tested as highly stressed U-bends, some of the 12 Cr and Modified 12 Cr Martensitic steels were susceptible to stress corrosion cracking (assumed to be caused by hydrogen sulphide). However, for materials of most interest, no cracking occurred at more meaningful stress levels. Some evidence of very shallow intergranular corrosion was observed in the CrMoV rotor alloy; the nickel containing rotor alloys, NiMoV and NiCrMoV did not appear susceptible to this type of corrosion. Longer exposures at other sites are being run in an attempt to distinguish the resistance of these alloys to intergranular corrosion. All alloys tested exhibited, as expected, significant reductions in the fatigue endurance limit due to corrosion pitting with the most commonly used 12 Cr bucket alloy exhibiting the smallest reduction.

In corrosion rate tests on several heat exchanger alloys, titanium and the higher alloy steels, including Type 316 stainless, exhibited no measurable metal loss. Except for carbon steel, the coupons which had measurable corrosion rates were either copper-base alloys or contained copper as a major alloy addition.

It is important to mention that for the last 10 days of the 160 day test period at the New Mexico site, the test chamber was without steam flow and the coupons were exposed to air. Compared to specimens subjected to much longer exposure at another site, the appearance of all specimens after 160 days' exposure indicated more severe corrosion than would have occurred had the steam flow been maintained up to the time of specimen removal - an observation which stresses the importance of organizing shutdown procedures to minimize corrosion, as in any turbine application, but is particularly, more stringent for geothermal applications.

Because of their relatively good performance in current geothermal turbine cycles and in the tests at the New Mexico site, it is anticipated that steam turbine alloys are available for the steam conditions anticipated in the Imperial Valley. At Niland, the concentration of hydrogen sulphide in the brine can be as high as 30 ppm (Magmamax No. 1). This translates to about 300 ppm in the primary steam flow which is comparable to the concentration in the New Mexico tests. At East Mesa, the concentration of H_2S appears to be considerably less, possibly as low as 1 or 2 ppm and therefore the environment for the steam turbine should be very clean. Since concentrations of hydrogen sulphide are not well specified, no provision was made in the cost analyses for preventing its release into the atmosphere. According to the above figures, the cost of implementing an H_2S removal system (such as the Stretford process) would be considerably higher at Niland than at East Mesa.

In estimating heat exchanger costs in this study, the entire steam turbine condenser and the primary and secondary heat exchanger tubes were assumed to be constructed of austenetic Type 316 stainless steel which appeared relatively inert in the New Mexico tests and which also performed well in the condensing system at the Geysers. The primary and secondary heat exchanger shells and tube sheets were assumed to be constructed of carbon steel since it was argued they could be made sufficiently thick, without significant cost penalty, to contain the relatively low steam pressures and the corrosion rates described. The heat exchangers could of course be constructed with the organic fluid on the shell side. The tubes would still be required to be stainless steel, but the shell would be free from corrosion. However, the tube side condensation coefficients for the steam would be reduced and the shell thickness would be increased to contain the higher organic fluid system pressure.

Based on the test results from New Mexico, there appear to be several relatively inexpensive alloys available which possess the necessary corrosion resistance for flashed steam heat exchanger application. However, the tests at New Mexico were relatively short term and experiments of much longer duration should be performed for both heat exchanger and turbine alloys at the selected sights.

The problem of sealing hydrocarbon gas in the organic turbine requires a decision relative to the type of seal best suited for this particular application. Two types ought to be considered: the face seal and the floating ring seal. In face seals (e.g. Elliot and Kopper) sealing is accomplished through a stationary spring loaded carbon ring in contact with the rotating shaft. The advantages of this design are that relatively little sealing fluid is exposed to the turbine working fluid (isobutane) and no sealing fluid is required when the machine is shut down. The major disadvantage is of course the finite life of the wear surface. Unlike the Koppers seal which has a split construction, the Elliot face seal is further handicapped by a solid carbon ring installation which is difficult to service.

The floating ring seal shown in Figure 64 is the type currently used in GE's hydrogen cooled generators. In this case, sealing is accomplished by a segmented bronze ring held against stationary retaining walls by a garter spring. Clearances (C_A and C_T) for a 14" diameter shaft are typically 0.003" to 0.015" for the gas-side and air-side, respectively. The design's major advantage is its indefinite life and proven trouble free performance. The main disadvantage is that large quantities of sealing fluid are exposed to the turbine working fluid and sealing fluid pressure is required at all times, including shutdown.

Figure 62. Floating Ring Hydrogen-Cooled Generator Seal

For this type of seal to be a viable candidate, dissolved iso-butane must be efficiently separated in relatively large quantity from the sealing fluid. The detrainment scheme proposed by GE is shown in the schematic of the seal fluid system in Figure 65. Three candidate sealing fluids are currently under investigation - a glycol, an aromatic phosphate ester and a standard turbine oil. They were selected primarily because they have moderate to good lubricating qualities and they provide a wide variation in iso-butane solubility which ranges from relatively low in the case of glycol to relatively high in the turbine oil. Because rotating shaft seal designs are often sensitive to the sealing fluid, shaft speeds, pressures, temperatures and the working fluid to be sealed, it is appropriate to make a full scale test of the rotating shaft seals in order to assure the reliability of a demonstration organic turbine.

Figure 63. Schematic Of Fluid System For Floating Ring Seal

REFERENCES

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2. K.E. Starling, Fluid Thermodynamic Properties for Light Petroleum Systems, Gulf Publishing Company, Houston, TX, 1973.
3. J.J. Martin, "Equations of State." Ind. Eng. Chem., Vol. 59, 1967, pp. 34-52.
4. Selected Values of Physical and Thermodynamic Properties of Hydrocarbons and Related Compounds, API Research Project No. 44, Carnegie Press, Carnegie Institute of Technology, Pittsburgh, PA, 1953.
5. J.H. Horlock, Axial Flow Turbines, Butterworths Press, London, 1966.
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APPENDIX A

A. HEAT EXCHANGER DESIGN

1. HEAT TRANSFER CORRELATIONS

a. Condensation Heat Transfer

When a vapor is fed to a condenser and the tube wall temperature is below the dewpoint, a film of condensate forms on the tubes. For a horizontal tube bank, the condensing heat transfer coefficient can be expressed

$$h_c \left(\frac{\mu_f^2}{k_f^3 \rho_f g} \right)^{1/3} = 1.5 \left(\frac{4G''}{\mu_f} \right)^{-1/3} \quad A1$$

The quantity $(4G''/\mu_f)$ is the Reynolds number of the condensate film and the subscript f implies evaluation at the condensate temperature. G'' is the condensate loading per unit of tube length and for a horizontal tube bundle is expressed

$$G'' = \frac{\dot{m}_s}{LN_t} \quad 2/3 \quad \text{lb/ft hr} \quad A2$$

where $\dot{m}_s$ is the mass flow rate of shell side fluid, L is the tube length and N_t is the number of tubes.

b. Forced Convection Heat Transfer

Convection heat transfer is a function of the Reynolds number and is usually expressed in the form

$$j_H = \frac{hd_e}{k} \left(\frac{Cp\mu}{k} \right)^{-1/3} \left(\frac{\mu}{\mu_w} \right)^{-0.14} = f \left(\frac{d_e G}{\mu} \right) = f(Re) \quad A3$$

where

h = convective heat transfer coefficient (Btu/ft²hr °F)

G = mass velocity (lb/ft²hr)

d_e = hydraulic diameter (ft)

(1) Outside Coefficient (h_o)

The outside or shell side coefficient is calculated with shell side values substituted for hydraulic diameter and mass velocity. The mass velocity is defined as

$$G_s = \frac{\dot{m}_s}{a_s} \quad A4$$

where $\dot{m}_s$ is the shell side mass flow rate and a_s is the cross sectional area normal to the shell side flow. For a baffled exchanger, a_s can be written

$$a_s = D_s B (p_t - d_o) / p_t \quad A5$$

where

D_s = shell inside diameter (ft)

B = baffle spacing (ft)

p_t = tube pitch (ft)

d_o = tube outside diameter (ft)

For tubes arranged on a triangular pitch, the hydraulic diameter is

$$d_e = 2.546 (0.433 p_t^2 - 0.393 d_o^2) / d_o \quad A6$$

For the Reynolds number range of interest, the empirical relationship between the shell side heat transfer coefficient and Reynolds number for tube bundles with 25% cut segmental baffles can be expressed reasonably accurately by the expression

$$j_H = 15.6 (Re_s / 1000)^{0.5465} \quad A7$$

(2) Inside Coefficient (h_i)

For flow inside the tubes, the mass velocity is the total tube side flow rate divided by the tube cross sectional area, i.e.,

$$G_t = \frac{\dot{m}_t}{N_t a_t n} \quad A8$$

where N_t is the number of tubes, a_t is area of a single tube and n is the number of tube passes. The hydraulic diameter is, of course, simply the inside tube diameter. For the Reynolds number range of interest, the empirical relationship between the tube side coefficient and Reynolds number can be expressed reasonably accurately by the expression

$$j_H = 6.4 (Re_t / 1000)^{0.805} \quad A9$$

2. PRESSURE DROP CORRELATIONS

The shell side pressure drop is expressed

$$\Delta P_s = \frac{f_s G_s^2 D_s (N + 1)}{2g\rho d_i (\mu/\mu_w) \cdot 0.14} \quad (\text{psf}) \quad A10$$

where $N + 1$ represents the number of times the shell fluid crosses the tube bundle between the inlet and exit. The friction factor f_s is a function of Reynolds number and for the Reynolds number range of interest can be expressed approximately as

$$f_s = 1.152 (Re_s/10)^{-0.1898} \quad A11$$

For condensers, the shell side pressure drop is assumed to be one-half the value computed from inlet conditions.

The tube side pressure drop comprises two components. One accounts for the frictional effects within the tubes and is expressed in essentially the same manner as the above, i.e.,

$$\Delta P_t = \frac{f_t G_t^2 L n}{2g\rho d_i (\mu/\mu_w) 0.14} \quad A12$$

where L is the tube length and n is the number of tube passes. The other component accounts for entrance and exit losses and return losses due to bending the flow 180° , if more than one pass is employed. This combined loss is assumed to equal four velocity heads per pass, i.e.,

$$\Delta P_r = 4n\rho V^2/2g \quad A13$$

3. THE STEAM/CO₂ CONDENSER

When steam condenses from a mixture containing an inert gas, a gradient in partial pressures is developed as depicted in Figure A1, which produces mass transfer of vapor through the gas. In such a system, the heat transfer mechanism depends not only upon Nusselt's condensing mechanism but also upon the laws governing diffusion. When the vapor diffuses through the non-condensable gas film and liquifies at the wall it carries with it its latent heat of condensation. Moreover, since there is a partial pressure gradient in the vapor promoting diffusion, there is also a temperature difference between the gas body T_g and the condensate T_f by which the gas is sensibly cooled. Thus, the total heat flow across the gas film is the sum of the latent heat of condensation and the sensible heat removed from the gas. This heat transfer mechanism may be expressed algebraically as:

$$h_o (T_g - T_f) + K_G M_v H_{fg} (p_v - p_f) = r_o h_i (T_f - T_c)/r_i = U(T_g - T_c) \quad A14$$

where

h_o = shell side dry gas coefficient (Btu/hr ft²°F)

h_i = tube side coolant coefficient (Btu/hr ft²°F)

T_g = gas temperature (°F)

T_f = condensate temperature (°F)

T_c = coolant temperature (°F)

p_v = vapor partial pressure in main gas body (atm)

p_f = vapor partial pressure at condensate film (atm)

m_v = vapor molecular weight (dimensionless)

H_{fg} = latent heat (Btu/lb)

K_g is the mass diffusion coefficient and is expressed as:

$$K_g = \frac{h_o (Cp\mu/k)^{2/3}}{Cp p_{gf} M_m (\mu/\mu_k)^{2/3}}$$

A15

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Figure A1. Condensation Potentials in Non-Condensable Gas/Steam Mixture (Reference A1)

M_m is the mean molecular weight of the gas/vapor mixture, i.e.,

$$M_m = (m_v + m_g) / (m_v/18 + m_g/44) \quad A16$$

and p_{gf} is the log mean of p_g of the main gas body and $p_g' = (p_t - p_c)$ the pressure of the inert gas at the condensate film, i.e.,

$$p_{gf} = (p_g - p_g') / 10 \log_e (p_g / p_g') \quad A17$$

The diffusivity k_d is given by

$$k_d = \frac{0.0166 T^{3/2}}{p_t (V_v^{1/3} + V_g^{1/3})^2} \left\{ \frac{1}{M_v} + \frac{1}{M_g} \right\}^{1/2} \quad A18$$

where

k_d = diffusivity (ft²/hr)

p_t = total pressure (atm)

V_v, V_g = molecular volumes of vapor and gas respectively

T = absolute temperature (°K)

M_v, M_g = molecular weights of vapor and gas respectively

For steam and CO₂, the molecular volumes are 14.8 and 29.6 respectively. Thus for a steam/CO₂ mixture, the diffusivity is

$$k_d = 1.5086 \times 10^{-4} T^{3/2} / p_t \quad A19$$

The above equations are valid for the case of a vapor forming a nonviscous, noncontrolling condensate film. For the condensation of steam from air and CO₂ in shells of horizontal baffled condensers, the external coefficient should be calculated from equations A15 and A7 using average properties of the gas/vapor mixture. Values of h_o estimated this way have been shown to agree well with experimental data.

4. CALCULATION PROCEDURE FOR HEAT EXCHANGER DESIGN

Except for the case where steam contains significant amounts of inert gas, the calculation procedure for all heat exchangers is essentially the same. First, values of tube size and pitch are selected and values are then assumed for the tube length, baffle spacing and inside shell diameter. Shell-side and tube-side heat transfer coefficients are then calculated using Equations A1 or A6 and Equation A9 and used to determine an overall clean coefficient.

$$1/U_c = 1/h_o + r_o \log_e (r_o/r_i)/k_w + r_o/r_i h_i \quad A20$$

where h_o and h_i are the shell-side and tube-side coefficients respectively and k_w is the thermal conductivity of the tube material. An overall dirty coefficient is also determined from the expression

$$U_d = Q/\pi d_o N_t L \Delta T_{\log} \quad A21$$

where $\Delta T_{\log}$ is the log-mean temperature difference in the exchanger. Finally, a fouling factor f_{foul} is calculated using

$$f_{foul} = \frac{\Delta_{scale}}{k_{scale}} = \frac{U_c - U_d}{U_c U_d} \quad A22$$

If the fouling factor is inadequate, the length or diameter of the heat exchanger must be increased and the calculation procedure repeated. As a final check, the shell and tube-side pressure drops must lie within acceptable limits.

5. DESIGN PROCEDURE FOR STEAM/CO₂ CONDENSER

Except for relatively small variations due to friction and dynamical effects, the total pressure in the condenser remains essentially constant. Thus, as condensation proceeds the partial pressure of the CO₂ will rise while that of the vapor will fall causing the condensate temperature to fall also. The overall coefficient of heat transfer varies greatly during the condensation process since the potential for diffusion also varies greatly as condensation leaves a greater percentage of inert gas. To account for the changes in overall coefficient through the condensation process, the condenser is divided into several intervals and the heat transfer equations are solved in a step by step manner. Ideally, the interval sizes are selected so that their heat loads are approximately equal.

Initially, a value is assumed in the first interval for the condensate temperature T_f and Equation (A14) is solved iteratively to provide a value for the quantity $U\Delta T$. The area of each interval is then calculated from the expression

$$a = q/(U\Delta T)_{av} \quad A24$$

where q is the heat transferred in the interval and $(U\Delta T)_{av}$ is the average value of $U\Delta T$ across the interval. The individual areas are then summed to obtain the total area required by the exchanger. A weighted ΔT is obtained from the summation

$$\Delta T_{wtd} = Q/\Sigma(q/\Delta T_{av}) \quad A25$$

where Q is the total heat transferred in the exchanger and ΔT_{av} is the average temperature difference between the main gas body

and the coolant across the interval. Finally, overall clean and dirty coefficients are calculated from the expressions

$$U_c = Q/A_{req} \Delta T_{wtd} \quad A26$$

$$U_d = Q/A_{actual} \Delta T_{wtd} \quad A27$$

and used to determine the fouling factor

$$f_{foul} = (U_c - U_d)/U_c U_d \quad A28$$

If the latter is inadequate, the condenser length and/or diameter must be increased and the design procedure repeated.

An example of the incremental design calculations for the single units cited in Section 5A when the brine contains 5% CO₂ is shown in Table A1. In this case, the condenser is divided into six intervals. Note the temperature difference maintained between the main gas-steam body and the condensate film. Note also the degradation of the overall heat transfer coefficient and the increase in surface area required as the inert gas occupies a larger fraction of the shell space. The variation of surface area and overall coefficient with heat transfer rate is shown for different CO₂ concentrations in Figure A2. Axial temperature distributions in the shell and tube side fluids are also shown for different concentrations of CO₂ in the temperature/enthalpy diagram of Figure A3.

Figure A2. Condensation of Steam-CO₂ Mixture for Different Weight Concentrations of CO₂ in the Niland Brine

Table A1

EXAMPLE OF INCREMENTAL CALCULATION RESULTS FOR STEAM/CO₂ CONDENSER

Station	T _g (°F)	T _f (°F)	T _c (°F)	ΔT (°F)	ΔT _{av} (°F)	UΔT	U	(UΔT) _{av}	(10 ⁶ q) Btu/hr	A _{req} (ft ²)	q/ΔT _{av}
1	335.3	333.3	310.3	25.0		13702	548				
2	334.2	332.2	306.0	28.2	26.6	19522	692	16612	56.42	3396	2,121,052
3	332.0	328.0	304.0	28.0	28.1	23831	851	21676	123.03	5676	4,378,291
4	327.5	322.0	297.0	30.5	29.2	18988	622	21409	160.68	7505	5,502,739
5	318.6	311.6	279.0	38.6	34.5	13960	361	16474	163.27	9911	4,732,463
6	300.8	289.0	255.0	45.8	42.2	11144	243	12552	152.40	12141	3,611,374
7	265.1	256.1	240.1	25.0	35.4	3357	134	7250	123.74	17067	3,495,480
									779.54	55,696	23,841,399

$$\Delta T_{\text{wtd}} = \Sigma q / \Sigma (q / \Delta T_{\text{av}}) = 32.7$$

$$U_c = \Sigma q / (A_{\text{req}} \Delta T_{\text{wtd}}) = 428.0 \text{ Btu}/(\text{ft}^2 \text{ hr } ^\circ \text{F})$$

$$U_d = \Sigma q / (A_{\text{act}} \Delta T_{\text{wtd}}) = 135.4 \text{ Btu}/(\text{ft}^2 \text{ hr } ^\circ \text{F})$$

$$f_{\text{foul}} = (U_c - U_d) / U_c U_d = 0.0050; \Delta p_s = 12.0 \text{ psi}; \Delta p_t = 4.0 \text{ psi}$$

Figure A3 Temperature/Enthalpy Distributions in the Primary Heat Exchanger for Different Weight Concentrations of CO₂ in Niland Brine

REFERENCE

- A1. Kern, D.Q., "Process Heat Transfer," McGraw Hill, 1950.

APPENDIX B

A. FLUID PROPERTIES

1. TRANSPORT PROPERTIES OF HYDROCARBONS

a. Viscosity

For nonpolar compounds such as isobutane and normal butane, the viscosity can be calculated from the following correlation (Reference B1) which covers the reduced density range between 0.1 and 3.0

$$\left((\mu - \mu_0) \xi + 10^{-4} \right)^{0.25} = 0.1023 + 0.023364 \rho_r + 0.058533 \rho_r^2 - 0.040758 \rho_r^3 + 0.0093324 \rho_r^4 \quad \text{B1}$$

where μ = viscosity (centipoises)

μ_0 = low pressure viscosity at the same temperature

$\xi = T_c^{1/6} / (M^{1/2} P_c^{2/3})$

T_c = critical temperature ($^{\circ}\text{K}$)

P_c = critical pressure (atm)

M = molecular weight

ρ_r = reduced density (ρ/ρ_c)

The above equation appears to be reliable for both the liquid and gas phases.

The low pressure viscosity is expressed in B2 as follows

$$\mu_0 \xi = 34.0 \times 10^{-5} T_r^{0.94} \quad T_r \leq 1.5$$

$$\mu_0 \xi = 17.78 \times 10^{-5} (4.58 T_r - 1.67)^{0.625} \quad T_r > 1.5 \quad \text{B2}$$

where T_r = reduced temperature (T/T_c)

b. Thermal Conductivity

The thermal conductivity is obtained from the following approximate analytical expressions in Reference B3

$$X = 14.0 \times 10^{-8} (e^{0.535 \rho_r} - 1) \quad \rho_r < 0.5$$

$$X = 13.1 \times 10^{-8} (e^{0.67 \rho_r} - 1.069) \quad 0.5 < \rho_r < 2.0$$

$$X = 2.976 \times 10^{-8} (e^{1.155 \rho_r} + 2.016) \quad 2.0 < \rho_r < 2.8 \quad \text{B3}$$

where $X = (k - k_0) \delta Z_c^5$

k = thermal conductivity (cals/cm sec °K)

k_0 = low pressure thermal conductivity at the same temp.

$\delta = T_c^{1/6} M^{1/2} / P_c^{2/3}$

Z_c = critical compressibility factor

The low pressure thermal conductivity k_0 can be expressed as (Reference B4)

$$k_0 = 10^{-6} (14.52 T_r - 5.14) 0.6667 C_p / \delta \quad B4$$

where C_p is the specific heat in cals/gram. mol. °K and is obtained with the density as a solution of the Starling Equation of State.

Because thermal conductivity is fairly difficult to measure, experimental data is relatively scarce and Equation B3, although the best generalized correlation at present for both liquid and gas phases, can be in error by as much as 45% at low reduced densities. Fortunately, in the condenser where low reduced densities occur, the heat transfer correlation is a function of the liquid density where errors appear to be somewhat less but still the order of 20% or so.

c. Thermodynamic Properties of Brines

The thermodynamic properties of vapor saturated sodium chloride solutions from 0 - 300 C have been analyzed by Silvester and Pitzer in reference B5. A summary of their results showing the variation of specific enthalpy and saturation pressure with sodium chloride concentration is given in Table B1. For ease of computation, the data was curve fitted by the method of least squares to obtain the following approximate analytical expressions for the specific enthalpy and saturation pressure

$$h_{sen} = A_0 + A_1 T + A_2 T^2 + A_3 T^3 + A_4 T^4 + A_5 T^5 \quad B5$$

$$P_{sat} = B_0 + B_1 T + B_2 T^2 + B_3 T^3 + B_4 T^4 + B_5 T^5 \quad B6$$

where h_{sen} is in joules/gram, P_{sat} is in bars and T is in °C. For the three sodium chloride concentrations of interest, i.e., 0, 25,000 and 300,000 ppm, the coefficients have the values given in Table B1.

Table B1
COEFFICIENTS FOR EQUATIONS B5 AND B6

	SODIUM CHLORIDE CONCENTRATION (ppm)		
	0	25,000	300,000
A ₀	-0.1112×10^3	-0.2284×10^3	-0.6065×10^2
A ₁	0.4498×10^1	0.9724×10^1	0.2217×10^1
A ₂	-0.5548×10^{-2}	-0.9343×10^{-1}	0.1311×10^{-1}
A ₃	0.4361×10^{-4}	0.7212×10^{-3}	-0.9118×10^{-4}
A ₄	-0.1475×10^{-6}	-0.2606×10^{-5}	0.2857×10^{-6}
A ₅	0.2134×10^{-9}	0.3574×10^{-8}	-0.3069×10^{-9}
B ₀	0.6648×10^0	0.4674×10^0	0.9822×10^0
B ₁	-0.3183×10^{-1}	-0.2339×10^{-1}	-0.4537×10^{-1}
B ₂	0.6074×10^{-3}	0.4743×10^{-3}	0.8069×10^{-3}
B ₃	-0.4994×10^{-5}	-0.3967×10^{-5}	-0.6093×10^{-5}
B ₄	0.2606×10^{-7}	0.2212×10^{-7}	0.2561×10^{-7}
B ₅	-0.1503×10^{-10}	-0.9793×10^{-11}	-0.1549×10^{-10}

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APPENDIX C

A. OPTIMUM FLASH TEMPERATURES FOR THE STEAM TURBINE CYCLE

In the multiflash system shown in Figure C1 in which a separate steam turbine is used for each stage of flash, the total output can be expressed as

$$W = \sum_{k=1}^n m_{s,k} H_{is,k} \eta_k \quad C1$$

where $m_{s,k}$ is the mass yield of steam from the k th flash stage, $H_{is,k}$ is the isentropic enthalpy drop between flash conditions and condenser and η_k is the expansion efficiency. If a constant specific heat is assumed for the expansion process, the specific output can be approximated as

$$\eta_k H_{is,k} = C_p (T_k - T_o) \quad C2$$

The quantity of steam liberated from the k th flash stage is obtained from the expression

$$m_{s,k} = (h_{k-1} - h_k) / H_{fg,k} \approx (T_{k-1} - T_k) / H_{fg,k} \quad C3$$

where h and H_{fg} refer to the saturated liquid enthalpy and the heat of vaporization, respectively. Since H_{fg} remains essentially constant over the temperature range of interest, m_s can be assumed to be a function only of h . As a result, the turbine output can be approximated simply as

$$W = C \sum_{k=2}^n (T_{k-1} - T_k) (T_k - T_o) \quad C4$$

where C is a constant.

For maximum output, the partial derivatives of output with respect to each flashing temperature must be set equal to zero. This leads to the following set of equations

Figure C1. Schematic of a Multi-Flash Steam Turbine Cycle

100 F and 25,000 ppm, respectively. The Figure would of course be modified some if plotted in terms of actual output since a reduction in flash temperature would tend to reduce aerodynamic loss and increase quality, both effects of which tend to increase turbine efficiency. However, a reduction in flash temperature would also serve either to increase the inlet velocity thereby increasing the throttling and admission loss or increase the inlet pipe and control valve size resulting in a major increase in equipment cost.

Figure C2. Ideal Output of a Dual Steam Turbine Cycle at East Mesa vs Primary and Secondary Flash Temperatures