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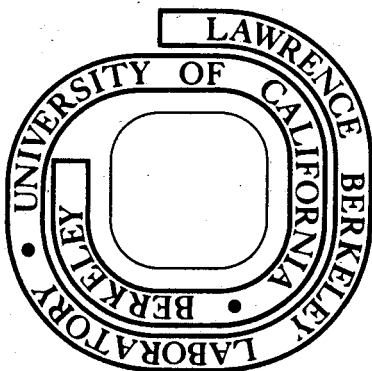
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THE TRANSPOSED CRITICAL TEMPERATURE  
RANKINE THERMODYNAMIC CYCLE

William L. Pope and Padraic A. Doyle

April 1980

Prepared for the U.S. Department of Energy  
under Contract W-7405-ENG-48



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PREFACE

This document contains new thermodynamic criteria for working fluid and turbine state selections for geothermal binary Rankine cycle power plants. This information is interesting and potentially general. A draft version was reviewed by outside experts who only partially share our optimism -- preferring more "proof" before publication.

We plan to continue this work when a more reliable equation of state is available, but feel the results herein warrant broad disclosure at the earliest possible time.

Independent investigations leading to a theoretical explanation of the observed behavior would be a most valuable contribution to power plant design technology.

Constructive criticism and suggestions for useful future work are welcome.

William L. Pope  
Staff Scientist  
Lawrence Berkeley Laboratory

CONTENTS

- I. ABSTRACT
- II. INTRODUCTION
- III. EXECUTIVE SUMMARY
- IV. DISCUSSION
  - (a) LBL Background in Geothermal Power Plant Studies
  - (b) What is the Significance of the Transposed Critical Temperature?
  - (c) Mixture Working Fluids - Previous Work
  - (d) LBL Investigations of Mixture Binary Rankine Cycles
- V. STUDY SCOPE
  - (a) System Schematic Diagram
  - (b) Baseline Input Assumptions
  - (c) Overall Study Scope - The Three Main Groups
  - (d) Features, Uses, and Limitations of Each Group
- VI. RESULTS
  - (a) General Thermodynamic Characteristics of Optimized Organic Mixture Binary Rankine Cycles.
    - 1) General Characteristic A - Supercritical Cycles
    - 2) General Characteristic B - The Optimum Mixture Composition is a Function of several variables.
    - 3) General Characteristic C - Proximity of Optimum Turbine States to the TPCT Line.
    - 4) General Characteristic D - Dry Turbine Expansion with a Minimum of Exhaust Superheat.
  - (b) The Busbar Cost Design Surface
  - (c) Influence of System Design Objective
    - 1) Constrained Yield Optimization - Optimum Composition Determined.
    - 2) Constrained Yield Optimization - Composition Optimizable



CONTENTS (Cont'd)

- (d) Thermodynamic Performance of Cost Optimized Cycles
- (e) Potential Correlation for Optimized Geothermal Binary Rankine Cycles
- (f) A Brief Look at the SDG&E Proposed Demonstration Plant

## VII. POSSIBLE FUTURE INVESTIGATIONS

## VIII. CONCLUSIONS

## IX. ACKNOWLEDGEMENT

## X. REFERENCES

## XI. APPENDIX

- (a) Figures and Graphs
- (b) Tabulated Numerical Results

# THE TRANSPOSED CRITICAL TEMPERATURE RANKINE THERMODYNAMIC CYCLE

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and  
Padraic A. Doyle

## I. ABSTRACT

The transposed critical temperature (TPCT)\* is shown to be an extremely important thermodynamic property in the selection of the working fluid and turbine states for optimized geothermal power plants operating on a closed organic (binary) Rankine cycle.

When the optimum working fluid composition and process states are determined for given source and sink conditions (7 parameter optimization), turbine inlet states are found to be consistently adjacent to the low pressure side of the working fluids' TPCT line on pressure-enthalpy coordinates.

Although the TPCT concepts herein may find numerous future applications in high temperature, advanced cycles for fossil or nuclear fired steam power plants and in supercritical organic Rankine heat recovery bottoming cycles for Diesel engines, this discussion is limited to moderate temperature (150°C to 250°C) closed simple organic Rankine cycle geothermal power plants.

Conceptual design calculations pertinent to the first geothermal binary cycle Demonstration Plant are included.

## II. INTRODUCTION

Commercial electric power production accounts for about 30% of the total U.S. fuel consumption. It has been estimated that by the year 2000, the electric

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\*Sometimes called the pseudo-critical temperature

utility industry may be consuming 50% of all U.S. primary energy resources. Most commercial electric power (about 82%) is currently produced in simple reheat or combined steam Rankine thermodynamic cycles. Only about 40% of the available useful work of the fuel is converted into electricity in existing Rankine cycle generating plants.

Key determinants in Rankine cycle performance are in the selected working fluid and turbine states. Although the steam Rankine cycle has been applied worldwide for over 75 years, little useful information has developed from this technology for general Rankine cycle working fluid selections. Thermodynamic criteria have been lacking for selecting the working fluid and turbine states for even the simple closed organic Rankine cycle for a given set of source and sink conditions.

The objective of this report is to provide new insight for working fluid and turbine state selection for the geothermal organic (binary) Rankine power cycle based on our observations on a single hydrocarbon working fluid mixture system.

### III. EXECUTIVE SUMMARY

- At current and projected brine prices and equipment capital costs, optimum geothermal binary Rankine power cycles for the isobutane/isopentane system between 170°C and 200°C resource temperatures are supercritical cycles when the brine is produced single phase.
- Turbine inlet temperatures for these optimum cycles are consistently within 1% of the Transposed Critical Temperature for the optimum working fluid composition.
- Assuming the Starling specific MBWR equation of state is valid, it is possible to reduce the busbar energy cost of the SDG&E proposed binary Demonstration Plant by about 5% by simply changing the working fluid mixture near mid-life. The annual savings would be about \$2 million.

#### IV. DISCUSSION

##### (a) LBL Background in Geothermal Power Plant Studies

Through funding from the U.S. Department of Energy (DOE), Division of Geothermal Energy (DGE)\*, LBL has been developing a relatively general thermodynamic cycle simulator, GEOTHM (Ref. 1, 2). The GEOTHM Code is coupled with powerful non-linear mathematical optimization routines (Ref. 3) which permit thermodynamic energy conversion systems of specified net output power to be optimized (Ref. 4, 5, 6) for various design objectives (i.e. minimum busbar energy cost, maximum utilization efficiency, etc.). The GEOTHM code is modular in design permitting various types\*\* of thermodynamic cycles and sub-systems to be design optimized (Ref. 7, 8, 9), and contains an extensive separate fluid properties library (Ref. 10).

Because geothermal binary Rankine cycle economics have not been commercially demonstrated, but the binary cycle is likely to be the best conversion system choice for a large fraction of the moderate temperature identified U.S. hydrothermal resource base, the LBL Utilization Technology Group has directed much of its research and development activities toward the understanding and technical demonstration of binary Rankine cycle prototype systems (Ref. 11, 12) and sub-systems (Ref. 13, 14). A fairly up-to-date description of LBL's system analysis capabilities is contained in a forthcoming DOE/DGE Geothermal Sourcebook (Ref. 15).

---

\*Utilization Technology Branch (Mr. Clifton B. McFarland, Chief)

\*\*With modest code changes -- user constraints, penalty functions, etc.

(b) What is the Significance of the Transposed Critical Temperature?

In the course of developing Section 8.2 of Reference 15 for DOE, we noted that the Transposed Critical Temperature\* of a binary Rankine cycles' secondary working fluid might play a key role in the selection of optimum turbine operating states (Ref. 15, Section 8.2.9.5). Specifically it was noted that when fuel costs (of a pure fluid geothermal binary Rankine cycle) clearly dominated (over plant capital costs), the cycles' optimum turbine inlet state approached the Transposed Critical Temperature line of the secondary working fluid.

This behavior was obtained even when the assumed pure fluid was an obviously poor working fluid choice for the assumed resource temperature (i.e.  $T_{cr} \ll T_{res}$ ).

It was further suggested in Ref. 15, Section 8.2.9.6, that if the secondary working fluids' composition (i.e. mole fraction of some assumed hydrocarbon mixture system) was simultaneously optimized (for given source and sink temperatures) with the six independent thermodynamic state parameters of the geothermal binary Rankine cycle, improved thermodynamic and economic performance might be obtained with TPCT cycles when both plant and fuel costs were material, but neither dominated.

(c) Mixture Working Fluids - Previous Work

The potential advantages of hydrocarbon mixtures as secondary working fluids for the geothermal binary Rankine cycle have been emphasized by Starling and others (Ref. 16, 17, 18, 19). A recent document (Ref. 20) contains an informative summary of the potential advantages and disadvantages of mixtures and possible working fluid selection criteria.

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\*The Transposed Critical Temperature line is defined as the locus of points in the working fluids' supercritical vapor region where the fluids' specific heat is a maximum.

In Reference 20, preliminary screening calculations were performed on geothermal binary Rankine cycles assuming various candidate working fluids including a 50/50 mix of isobutane and isopentane ( $iC_4$  and  $iC_5$ ) over a range of potentially suitable resource temperatures. Various working fluid thermodynamic characterization parameters, including Kihara's and Fukunaga's "I-factor" (Ref. 21), were utilized for selecting turbine entrance (and in some cases exit) states for specific ideal (reversible) cycle calculations. Subsequent calculations in Ref. 20 identified  $C_{18}$ -2-Butene as a potentially attractive working fluid candidate for three resource temperatures (300°F, 400°F, and 500°F).

Our primary objective herein is not to compare or advocate various potential working fluids (or mixture systems), but to simply determine whether or not busbar cost optimized Rankine cycles with a given mixture system exhibit certain general thermodynamic characteristics.

We have not investigated the many configuration possibilities of the binary Rankine geothermal cycle (i.e. multiple boiling and condensing stages (Ref. 22)), but have limited our studies to the behavior of the simple (non-regenerative) binary Rankine cycle over sub-critical and supercritical operating regimes assuming the isobutane-isopentane system.

The potential overall economic advantage (relative busbar energy cost) of mixtures over pure fluids has been implied but not clearly defined -- 1) partly because the accuracy of P-V-T correlations for mixtures using conventional BWR type equations of state is questionable (Ref. 10, 19); 2) partly because the thermodynamic behavior of mixtures in some subsystems (i.e. condensers) is complex and has only been approximated (Ref. 20); and 3) partly because previous

studies (without multiparameter optimization capabilities) have not clearly identified the optimum thermodynamic characteristics (mixture composition, turbine state conditions, etc.) of mixture binary Rankine cycles.

(d) LBL Investigations of Mixture Binary Rankine Cycles

Since the Spring of 1979, LBL has been investigating the performance characteristics of busbar cost optimized hydrocarbon mixture binary Rankine geothermal energy conversion cycles assuming the isobutane/isopentane ( $iC_4/iC_5$ ) hydrocarbon system over a range of potentially applicable resource temperatures. This hydrocarbon mixture system has a "retrograde" vapor saturation boundary on temperature-entropy coordinates and an I-factor (Ref. 20, 21) less than 1.0, and therefore may be regarded simply as a sub-set of all potentially attractive candidate working fluids for this resource temperature range.

The Starling specific MBWR (1975) equation of state (Ref. 23) has been used by LBL for this hydrocarbon system because it existed even though its accuracy is questionable (Ref. 10, 19, 24 ). Because of this, the fact that brine price and equipment capital costs are rapidly escalating, current GEOTHM financial routines (Ref. 15, Section 8.2.11.6) are not general\*, and because 7 parameter binary cycle mixture optimizations are considerably more complex and costly than 6 parameter optimizations with a given working fluid, we have conducted our present mixture Rankine cycle studies on a simulated simple binary power plant system with a multitude of very practical simplifying assumptions.

Our goal is to get to the essence of the problem - a clear definition of potentially general thermodynamic features of optimized binary Rankine cycles - with the simplest possible level of economic and thermodynamic system characterization.

---

\* (New generalized economic routines have been developed for GEOTHM by the Mitre Corporation (Ref. 25) under contract to LBL. These routines are currently being incorporated and tested. A new version of GEOTHM will be available in the late Summer of 1980).

## V. STUDY SCOPE

Although our mixture studies to date have been limited to the  $iC_4/iC_5$  system, the scope of this conceptual design study is relatively broad and general. First of all we report only on optimized system designs with as few as possible user input biases (Ref. 15, Section 8.2). The Design Objective for most of the analyses herein is minimum busbar cost. Constrained brine "yield" optimizations are also reported in Section VI (c), but not with arbitrarily selected pinch points. Secondly, we investigate three realistic constraint assumptions for optimized systems, and finally we explore a variety of sub-system cost conditions.

### (a) System Schematic Diagram

Figure 1 is a schematic diagram of the simple (non-regenerative) geothermal binary Rankine cycle. The "brine" (pure  $H_2O$ ) is assumed to be produced at the wellhead as single phase (saturated) liquid using suitable electric motor driven downhole pumps in each production well. Heat extracted from the brine is transferred through an array of conventional shell-and-tube heat exchangers to a secondary working fluid (hydrocarbon mixture) where it is vaporized on the shell-side at either sub-critical or supercritical pressures. Work extracted by the turbine drives a generator which supplies electric power to all the motor driven equipment. Heat is rejected from the cycle via an array of conventional shell-and-tube condensers to a forced draft wet cooling tower. All major parasitic losses (including production and injection pumping) are characterized. The seven independent thermodynamic system parameters (optimizable parameters) are also shown in Figure 1.



(b) Baseline Input Assumptions

Table 1 lists other system input assumptions of this study for the Baseline Case. In order to determine conceptual level busbar costs of such a system, one must have reasonably accurate sub-system costs. Sub-system costs were obtained by normalization to values reported in Reference 17 as described in Ref. 15, Section 8.2.11.6. The thermodynamic and economic performance of binary Rankine geothermal power plants using pure working fluids for the same cycle and input assumptions (as Figure 1 and Table 1) are reported in Ref. 15, Section 8.2.

(c) Overall Study Scope - The Three Main Groups

Because our preliminary results suggested interesting new, and potentially general thermodynamic characteristics for optimized mixture cycles, it became important to study the cycles under a broad variety of input assumptions. Figure 2 lists the overall range of cost, efficiency, and constraint assumptions adopted for this study.

The system cost optimizations herein fall logically into three main groups (see Figure 2):

- |                    |                                                                                                           |
|--------------------|-----------------------------------------------------------------------------------------------------------|
| <u>Group A</u>     | Un-constrained optimizations - Turbine inlet allowed to be anywhere outside the two-phase vapor envelope. |
| <u>Group B, 1)</u> | Constrained optimizations - Turbine inlet entropy $\geq$ maximum saturated vapor entropy.                 |
| <u>Group B, 2)</u> | Constrained optimizations - Brine return temperature $\geq$ 344.26 K (160°F).                             |

Each of the above three groups are used with a variety of cost assumptions to illustrate the following specific features of optimized mixture binary Rankine geothermal power cycles:

- 1) Generally preferred turbine entrance conditions.
- 2) Affect of resource temperature, cost, and constraints on the optimum mixture composition.
- 3) The economic impact of moist turbine expansion.
- 4) Optimum turbine states as affected by inlet state constraints.
- 5) Affect of brine return temperature constraints on optimum turbine states.
- 6) Affect of relative fuel price to primary heat exchanger capital cost on the optimum cycle type--i.e. sub-critical vs supercritical.

(d) Features, Uses, and Limitations of Each Group

GROUP A optimizations were configured to determine the thermodynamic and economic characteristics of binary Rankine cycles under the absolute minimum busbar cost conditions (for given input cost assumptions) with as few as possible input constraints. This group is also used to illustrate the general geothermal binary Rankine cycle busbar cost Design Surface (Section VI, (b)) as a function of turbine inlet conditions, the Global Minimum cost along the TPCT line, and the extreme system economic penalties associated with moist expansion in the turbine. To do this we simulated the turbine for GROUP A studies as a multi-stage expansion engine with a fixed maximum stage pressure ratio. The dry stage efficiency (85%) was penalized by one percent for each percent of stage exhaust moisture.

It should be pointed out that the expander high pressure stage conditions for GROUP A have no built-in margin of safety against moist expansion. If the design optimization puts the exhaust of any stage at (or near) the vapor saturation boundary, expansion in the nozzels of that stage would (could) be in the two-phase vapor region.

GROUP B, 1). The selected mixture system fortunately has a particular vapor isentrope condition that can be useful for design purposes without imposing arbitrarily large expander mid-stage superheat margins -- the isentropes intersect the vapor saturation boundary twice. Because of this, the maximum vapor isentrope can be used to define expander inlet conditions which guarantee dry expansion (if the equation of state is adequate). The GROUP B, 1) constrained cost optimizations herein, then, determine very near absolute minimum busbar cost conditions under practical stage superheat margins on a consistent basis.

The GROUP B, 2) constrained cost optimizations allow a first order assessment of the thermodynamic and economic performance of binary Rankine cycles with the  $iC_4/iC_5$  hydrocarbon system at the Heber geothermal resource. Because of the inverse solubility of silica, brines produced at the Heber (and other) resource could seriously foul the primary exchanger and clog injection wells if cooled to too low a temperature. For the GROUP B, 2) constrained optimizations, we have assumed a minimum primary heat exchanger exit temperature (brine return temperature) of 344.26 K (160 F), consistent with the latest published studies done by Fluor (Ref. 19) for the SDG&E proposed binary Demonstration Plant at Heber. The "fuel cost up by a factor of 5" assumption (Figure 2) simply brings our calculated unit brine costs close to those stated in Ref. 19 on both a mill/kwh basis and a \$/MBtu basis. All the other sub-system baseline unit cost factor multipliers in Figure 2 were arbitrarily selected.

## VI. RESULTS

Because our results depend somewhat upon fuel cost and constraint assumptions (Figure 2), we will present a generally observed trend overview first. This will be followed by a detailed discussion of the observed behavior and the different characteristics of designs in GROUP A, GROUP B, 1), and GROUP B, 2).

(a) General Thermodynamic Characteristics of Optimized  $iC_4/iC_5$  Mixture Binary Ranking Cycles with Single Phase Brine Production  
( $170 \leq T_{RES} \leq 200$  C)

We have found that optimum system designs exhibit the following characteristics:

- A. Working fluid cycles are generally supercritical.
- B. The optimum mixture composition depends upon several variables (and the design approach).
- C. Optimum turbine inlet states lie extremely close to the working fluids' TPCT line with a constant reduced P, T displacement which depends only upon constraint assumptions.
- D. Optimum turbine expansion is dry with a minimum\* of exhaust superheat (consistent with the I-factor of the selected mixture and turbine inlet constraint assumptions).

Because of the relatively fixed thermodynamic characteristics of optimized  $iC_4/iC_5$  mixture binary Rankine cycles, we find that much previous confusion\*\* surrounding the "optimum" binary Rankine cycle as the result of previous studies on pure working fluids potentially goes away. As the result of A, B, C, and D we find that cost optimized  $iC_4/iC_5$  simple binary cycle systems appear to obey similitude rules on reduced thermodynamic coordinates and correlate very well over a range of resource temperatures. The reduced turbine states of optimum cycles exhibit extremely small sensitivities to sub-system cost and efficiency input assumptions for a given set of constraint conditions.

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\*There is an important exception to this noted on page 18.

\*\*Because differences in busbar cost are often imperceptible, it is not uncommon for geothermal system designers to "override" their stated input minimum cost Design Objective with subjective input logic (i.e. minimize superheat and/or avoid "high" working fluid pump parasitic losses in supercritical cycles) and, consequently, adamantly disagree with others on the nature of the "optimum" cycle (i.e. sub-critical or supercritical). The competition can lead to even more complex cycles of questionable economic merit when the real problem appears to be in the selected working fluid and design approach (Objective). See also Section VI, (c).

(1) General Characteristic A - Supercritical Cycles

Figure 3 illustrates the computed optimum turbine inlet temperatures and pressures for isobutane/isopentane mixture binary Rankine geothermal power plants over the arbitrarily selected resource temperature range of  $170^{\circ}\text{C} \leq T_{\text{Res}} \leq 200^{\circ}\text{C}$ . These results are from GROUP A (Figure 2) for baseline cost and efficiency assumptions, Table 1.

First of all it can be noted that all the GROUP A optimum binary cycles for non-dilute fractions of this mixture system (and equation of state) are slightly supercritical (see also Table 2 for non-baseline conditions within GROUP A). During the course of the busbar cost minimization, the working fluid composition (mole fraction of isobutane) is selected (along with the six other independent state parameters) for each resource temperature to maintain the supercritical cycle.

Several changes in fixed input assumptions were made to verify the consistently supercritical optimum cycle. For example, increasing the fuel cost (\$/MBtu) by a factor of 5 (for the  $182^{\circ}\text{C}$  resource case - see Table 2) has a minor effect on the turbine inlet temperature, but the optimum composition goes up (21%) to achieve a reduced critical temperature and increase the energy removed per pound of brine. The primary heater pinch point temperature difference goes down from about  $5.31^{\circ}\text{C}$  to about  $2.79^{\circ}\text{C}$ , and the turbine inlet pressure goes up about  $2 \frac{1}{4}$  bars to maintain the supercritical cycle. This behavior is expected.

Similarly, increasing the primary heater unit capital cost (\$/m<sup>2</sup>) by a factor of 2 (for the  $182^{\circ}\text{C}$  resource case, see Table 2) has a minor effect on both the optimum composition and the turbine inlet state. To accomodate the higher primary exchanger unit capital cost, the pinch point delta T increased from about  $5.31^{\circ}\text{C}$  to  $10.7^{\circ}\text{C}$  to increase the mean delta T, and the cycle

remained supercritical. However, it is clear that if the primary heat exchanger unit capital cost were significantly increased for the low (baseline) brine costs, a sub-critical cycle would be inevitable. This possibility was investigated for GROUP B, 1) constrained cost optimizations (Table 3) for baseline brine costs with various primary exchanger unit capital costs. We find that when the exchanger unit capital cost is a factor of 3 or more above baseline conditions (Ref. 17), a sub-critical cycle is optimum. Because brine prices have increased much more significantly than exchanger unit capital costs since 1976 (see Ref. 17 and Ref. 19), and this trend will probably continue, the likelihood of a sub-critical optimum cycle for these conditions is very remote.

We also tried other intuitively obvious input assumptions to see if we could get the 182°C resource temperature cycle to "go sub-critical" for the minimum busbar cost design objective under GROUP A conditions. For example, increasing the unit cost (\$/kW) of the hydrocarbon circulating pump and its electric motor by a factor of 5 actually increased the optimum turbine inlet pressure very slightly, and all other changes were trivial. The optimum cycle remained supercritical.

Keep in mind, the forgoing "numbers" are likely to be significantly affected by the assumed equation of states' ability to characterize the critical properties of mixtures - the one used here could be appreciably in error (Ref. 10, 24).\*

---

\*The potential inability of BWR type equations (mixing rules?) to predict the critical loci of hydrocarbon mixtures has been pointed out by Silvester (Ref. 10). For example, the Starling specific MBWR predicts that plots of  $P_{critical}$  and  $T_{critical}$  as a function of mole fraction would be concave upward for mixtures of n-butane and n-pentane, whereas carefully measured data plots concave downward. See also Ref. 24, Chapter 3 for other systems.

Finally, in our last attempt to force the optimum 182°C resource system to go sub-critical, we reduced the input adiabatic efficiency of the cycle's hydrocarbon circulating pump from 80% to 60% and selected sub-critical first guess values of turbine inlet pressure and temperature near the vapor saturation boundary. After initially inconclusive runs which required code changes to better differentiate between sub-critical and supercritical cycles very near the critical pressure, and incorporating different unit cost factors (\$/ft<sup>2</sup> of heat transfer surface) for kettle boilers, we obtained the result shown on the bottom line of Table 2.

We feel the foregoing discussion supports General Characteristic A that all practical cycles with this mixture system between 170°C and 200°C resource temperatures will be supercritical cycles if the brine is produced single phase.

(2) General Characteristic B - The Optimum Mixture Composition Is a Function of Several Variables

Figure 4 illustrates the computed optimum mixture composition (mole fraction of isobutane) as a function of resource temperature for un-constrained designs (GROUP A) and for four resource temperatures with a brine return temperature constraint, GROUP B, 2). Also shown on the plot are Holt's conceptual design recommendations (Ref. 17, 1976) for initial Heber resource temperature conditions (Holt suggested a different hydrocarbon system for end-of-life conditions) and Fluor's (1979) recommendations for the Heber Demonstration Plant.

Our results are not very different from Holts' recommendations for initial conditions either for GROUP A or GROUP B, 2 assumptions, but are dramatically different from Fluor.

It should be pointed out that fundamental design "philosophy" differences characterize the current Demo design (Ref. 19) -- Fluor apparently concludes that acceptable economic conditions will be achieved over the entire plant life with no change in the working fluid composition. We will expand upon this later in Section VI, (f).

It is clear from Figure 4 that the optimum composition depends at least upon the resource temperature and brine return temperature constraints. Inspection of Tables 2 and 3 discloses that the optimum composition is also a function of the unit fuel cost, but is little affected by other sub-system unit capital cost or efficiency factor assumptions -- see Table 5.

It is apparent from the foregoing that determining the optimum mixture composition for an assumed resource temperature (even for a given mixture system) can be the difficult part of the problem (multiparameter optimization capabilities are extremely valuable), and the computed optimum composition will depend upon fuel costs, constraint assumptions, and the accuracy of the assumed equation of state.

We will also show later in Section VI, (c), that different "optimum" compositions may be inferred depending upon the method used in thermodynamic (brine) yield optimizations making the analysis approach, or Objective, important.

We feel the foregoing supports General Characteristic B that the optimum mixture composition is a function of several variables.



(3) General Characteristic C - Optimum Turbine Inlet States and Their Proximity to the TPCT Line

By far the most interesting aspect of this study of optimized geothermal binary Rankine cycles with  $iC_4/iC_5$  mixture working fluids is the fact that the turbine inlet states on reduced coordinates lie extremely close to, and at a relatively constant displacement from, the fluids' Transposed Critical Temperature (TPCT) line. This behavior is illustrated for busbar cost optimized cycles in Figure 5 for  $(T_T/T_{cr})_{OPT}$  and Figure 6 for  $(P_T/P_{cr})_{OPT}$ .

In Figure 5 it can be noted that the reduced optimum turbine inlet temperature,  $(T_T/T_{cr})_{OPT}$ , falls consistently within about 0.8% above the reduced transposed critical temperature\* (at the optimum turbine inlet pressure) for all resource temperatures (non-dilute mixture mole fractions), all cost and efficiency assumptions, and all constraint conditions adopted for this study.

Similar behavior for the optimum turbine inlet pressure is illustrated in Figure 6. Again the displacement is relatively constant on reduced coordinates, and large changes in unit cost and efficiency input assumptions have no significant influence for given constraint conditions.

*The obvious importance of the transposed critical temperature on the selection of working fluids and state conditions for Rankine cycle energy conversion systems has never been previously reported to our knowledge.*

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\*The pressure on the TPCT line at the optimum turbine inlet temperature was found by sweeping over various densities on the turbine inlet temperature isotherm (isothermal search) until the maximum specific heat was found (from the equation of state) within one part in  $10^{10}$ . The temperature on the TPCT line at the optimum turbine inlet pressure was found by sweeping over various temperatures (with the foregoing isothermal search as an inner loop) until the pressure at the  $C_p$  maximum agreed with the turbine inlet pressure within one part in  $10^7$ . This TPCT search in no way influenced the selection of either the working fluid or the optimum state conditions of the cycles.

We feel that General Characteristic C is quite adequately supported by the results in Figure 5 and Figure 6.

(4) General Characteristic D - Dry Turbine Expansion  
with a Minimum of Exhaust Superheat

The hydrocarbon expander for GROUP A data points of these studies was simulated as a multistage engine with a specified (0.85) dry stage adiabatic efficiency,  $(\eta_i)_{\text{dry}}$ , and a specified maximum per stage pressure ratio (1.8). The input expander unit cost (\$/kW) was constant, independent of the number of expansion stages required. If the selected local stage exhaust conditions were in the two-phase vapor region, the local dry stage efficiency was arbitrarily reduced 1% for each percent exhaust moisture (Ref. 26). The overall average expander adiabatic efficiency was calculated from:

$$\bar{\eta}_e = \left[ \sum_{i=1}^n (\eta_i \Delta h_i) \right] / \left[ \sum_{i=1}^n (\Delta h_i) \right] \quad (1)$$

where  $n$  is the number of expansion stages required between  $P_{\text{in}}$  and  $P_{\text{out}}$ ,  $\eta_i$  is the actual, moisture penalized adiabatic efficiency of stage,  $i$ , and  $\Delta h_i$  is the actual enthalpy change in stage,  $i$ .

Figure 7 illustrates typical calculated process state points for all the cost optimized GROUP A cycles of this study. Note that the turbine high pressure stage nozzle expansion is in the two phase vapor region whereas the exhaust is virtually on the vapor saturation boundary for the selected optimum working fluid composition. The calculated overall average expander efficiency,  $\bar{\eta}_e$ , was never less than 0.8490 (first stage exit quality always greater than 0.994) for all of the minimum busbar energy cost optimizations reported in this study. The obvious physical implications, of course, are that superheat is undersireable yet expanding into the two-phase vapor dome must be avoided with this retrograde fluid system.

For example, given the GROUP A option of choosing between a totally dry expansion ( $\bar{\eta}_e \equiv 0.85$ ) somewhat to the right of the vapor saturation boundary on P-h coordinates (where isentropes are less steep than at the boundary) and expanding slightly "moist" in a given stage with the previously mentioned assumed stage efficiency degradation, the option consistently chosen in the unconstrained GROUP A cycles, places the higher pressure stage exhausts virtually on the vapor saturation boundary with a minimum of superheat at the desuperheater inlet (consistent with the optimum mixtures' I-factor).

When the turbine inlet conditions were constrained under GROUP B, 1) conditions (see Figure 2), all cycles again optimized with minimum possible superheat at exhaust conditions. The optimum turbine inlet state fell "exactly" at the optimum working fluids' maximum saturated vapor isentrope constraint in close proximity to the TPCT line.

We feel the foregoing discussion supports General Characteristic D that the preferred turbine expansion is dry with a minimum of exhaust superheat (when the optimum fluid composition is selected).

An interesting exception to the previously described minimum exhaust superheat condition occurs when the brine return temperature is constrained - GROUP B, 2). Under these conditions the turbine inlet state (for this mixture system between 170 C and 182 C) moves to higher temperatures along the TPCT line and appreciable superheat at exhaust conditions results for the "optimum" design.

A very simplified sketch of typical optimum turbine expansion paths for the three constraint conditions of this study is shown in Figure 8. At this time we don't understand why the "extra" superheat is acceptable (optimum!) under the brine temperature constraint condition.

(b) The Busbar Cost Design Surface

The busbar energy cost design surface for optimized mixture binary Rankine cycles can be visualized in the 3-D plot, Figure 9, reproduced from Ref. 28. In this figure the computed busbar energy cost has been plotted as a function of the turbine inlet temperature and pressure.\*

The values of  $P_T$  and  $T_T$  along the sharp "trough" at the right of Figure 9 (the Global Minimum busbar cost region) are virtually coincident with the transposed critical temperature line of the optimum chosen mixture fraction. For example, see also the specific heat contour plot, Figure 10, and the busbar cost contour plot with super-imposed TPCT line, Figure 11.

The relatively flat potential operating region in the left side of Figure 9 corresponds to slightly sub-critical and slightly supercritical turbine inlet states below the TPCT to the right of the critical point in the superheated vapor region of a pressure-enthalpy diagram.

The region on the right side of Figure 9, where the busbar energy cost rises abruptly, corresponds to turbine inlet states immediately above and/or to the left of the transposed critical temperature line (on P-h coordinates) which would result in expansion (by one or more exhaust stages) into the two-phase vapor region of the selected optimum mixture.

This plot clearly shows the severe system economic penalties associated with wet expansion in the turbine for geothermal cycles with this mixture fluid. Note that even if the GROUP A expander stage efficiency degradation assumed (1% for each percent of stage exhaust moisture) was reduced signi-

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\*(The 5 other independent system state parameters (optimizable parameters) have been fixed at their computed optimum values for producing this plot. See Ref. 15, Section 8.2.9.3).

ificantly (say a factor of 4), wet expansion would still be undesirable, dry expansion to the right the vapor phase boundary would still be preferred, and no change in the location of the Global Minimum cost would occur.

Figure 9 also shows, however, that there is a relatively broad potential operating region (slightly to the right and/or below the TPCT line on P-h coordinates) where turbine inlet states can be safely chosen to avoid moist turbine expansion with little operating cost penalties --i.e. GROUP B or similar constraint assumptions. The actual margin of safety in the design of the selected cycle depends, of course, upon designer constraint assumptions, but, more important, upon accurate knowledge of the location of the vapor saturation boundary in the near critical region for the chosen working fluid (mixture).

*The transposed critical temperature line, defined by the peaks in the working fluids' anomalous specific heat, and the vapor saturation boundary (of the optimum working fluid), therefore, clearly define the limits of the economically desirable operating region for the expander of the binary Rankine cycle.*

(c) Influence of System Design Objective

In this section we show that if system maximum brine yield, a measure of thermodynamic performance, is the Design Objective, the selected turbine inlet states fall virtually on the TPCT line for the optimum (cost) mixture composition.

1) Constrained Yield Optimization - Optimum Composition Determined

A commonly used design criterion (Design Objective) in geothermal power plant conceptual design studies (Ref. 17) is a user constrained thermodynamic

optimization. In this design mode, the maximum\* specific net energy (of the brine -  $P_{net}/\dot{m}_b$ ) is the Design Objective, but practical (ostensibly economic) values are assumed for the exchanger pinch points and cooling tower approach temperature difference. In this design mode all costs are "immaterial".

We present our results of a 3 parameter specific net energy, or "yield", optimization assuming that the turbine inlet temperature and pressure and the condenser bubble point temperature are the independent thermodynamic state parameters (optimizable parameters - Ref. 15, Section 8.2.9). This thermodynamic optimization was constrained (but not arbitrarily) by using the selected mixture composition and the three pinch points from a previous 7 parameter busbar cost optimization as fixed parameters. We used the previous "baseline" cost optimized 182°C resource temperature case from GROUP A for comparison (see Table 2, TPCT010).

Table 6 lists the input and computed optimum values of the independent thermodynamic variables for the 7 parameter cost and 3 parameter constrained yield optimizations for the 182°C resource temperature case.

It can be noted from Table 6 that when thermodynamic performance is the Design Objective, the optimum turbine inlet states are even closer to the transposed critical temperature line for the optimum working fluid.

*This example reinforces all previous statements about the importance of the transposed critical temperature to Rankine cycle power plants.*

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\*The absolute maximum specific net energy can be obtained (for fixed sub-system efficiencies) by setting the temperature differences of all non-work producing processes in the cycle equal to zero (i.e. the pinch points), but this of course, leads to near infinite plant costs. This condition has only theoretical significance, and is a poor indicator of relative busbar costs in pure working fluid selections for geothermal binary cycles (see Ref. 15, Fig. 16).

Depending upon the Design Objective, significant differences can exist in the computed mass flow rates which can affect sub-system selections. In this section we will show that even when the "correct" working fluid composition and pinch points are assumed in the constrained thermodynamic optimization (the previous example), the overall design is not at the economic optimum system condition.

The constrained thermodynamic optimization places too much importance on the fuel sub-system, and overdesign in the plant sub-system results. Because all costs are immaterial in thermodynamic optimizations, subtle but important expander efficiency considerations can be overlooked.

Table 7 lists a comparison of the computed "brine" exit temperature, sub-system mass flow rates, system efficiencies, and relative costs for the examples in Table 6.

Upon brief inspection of Table 7 it would appear that there is no significant different at the bottom line (busbar cost) between the two design philosophies, however this is an incorrect conclusion. The busbar costs just happen to be close for these two Design Objectives, because we did not assume arbitrary values for the working fluid (mixture composition) and the pinch points (see Table 6).

The fuel cost is, of course, lower by about 3% in the yield optimized design, because this was the design "goal", but the plant sub-system is over-designed by about 5.7%. In addition, the lower fuel cost may not in fact be obtainable, because the low brine exit temperature would obviously promote accelerated scaling in both the primary heater and the injection wells (i.e. at Heber). Arbitrarily increasing the brine flow rate with everything

else the same to prevent potential brine scaling would be counter-productive if economic feasibility of the system was marginal. If a minimum brine exit temperature constraint had also been applied, the relative economics might be worse.

Because the turbine inlet temperature is lower and the turbine inlet pressure is higher (in the Yield case) for the same working fluid mixture composition, the turbine expansion is quite moist in the first two stages, and the expander efficiency is down about 1.8%. This ostensibly acceptable "minor" expander efficiency degradation has contributed to much of the increase in the hydrocarbon and cooling water flow rates and the resulting plant capital cost increase.

The next section illustrates even more significant "optimum" system design differences when maximum thermodynamic performance (rather than minimum busbar cost) is the Design Objective, but the cost optimum fluid composition is not assumed.

## 2) Constrained Yield Optimization - Composition Optimizable

In this section the results of a 4 parameter brine yield optimization will be compared to the previous busbar cost optimized system for the 182°C resource case. This analysis will illustrate an additional potential pitfall of a thermodynamic Design Objective for mixtures--not only may system economic penalties be high, but also the selected mixture composition may be quite different from its optimum value (busbar cost).

Table 8 briefly summarizes the differences between the previous 7 parameter busbar cost optimized design and a 4 parameter brine yield optimized design. The only difference between this constrained yield optimization and the previous 3 parameter example is that the isobutane mole fraction is assumed to be "optimizable" here.



By making the working fluid composition "optimizable" in this 4 parameter constrained yield optimization, the additional degree of freedom has reduced the brine flow rate another 4% (over the 3 parameter yield optimization), and re-configured the turbine states (new composition) for dry expansion, but the busbar cost has gone up another 4.6 percent (Table 8)!

The plant sub-system is now oversized by about eighteen percent, largely because of the vastly oversized heat rejection system, and the wrong working fluid mixture has been selected even though optimum pinch conditions were assumed.

The foregoing two constrained yield optimizations reiterate design technique inconsistencies found in previous pure fluid geothermal power plant design studies (Ref. 15, Section 8.2) which could potentially compromise commercial development of binary Rankine cycle geothermal power plants. As we have found with pure working fluids, busbar cost is the only consistent commercial plant Design Objective for mixture binary Rankine geothermal power plants.

(d) Thermodynamic Performance of  $iC_4/iC_5$  Binary Mixture Cycles

We have previously shown that minimum busbar cost and constrained maximum thermodynamic yield optimized geothermal power plants with the  $iC_4/iC_5$  mixture system are generally supercritical cycles with the turbine inlet state in close proximity to the transposed critical temperature line for all resource temperatures (non-dilute mixture compositions), cost factors, and constraint conditions considered. It is intuitively clear that if these TPCT cycles are economically preferred (over the infinite variety of other possible sub-critical and supercritical operating states), either the annualized fuel cost or the annualized plant capital cost must somehow be consistently low.

If the busbar cost optimized system has a high economically achievable fuel utilization efficiency,  $(\eta_u)_{OPT}$ , the annualized fuel cost will be low. On the other hand, the higher the economically justified net plant efficiency,  $(\eta_c)_{OPT}$ , the lower the annualized plant capital cost will be.

The computed "brine" economic optimum wellhead utilization efficiency,  $(\eta_u)_{WH}$ , and the economic optimum net plant efficiency,  $\eta_c$ , for this study are shown as a function of resource temperature,  $T_{RES}$ , in Figure 12.

From this plot it may be noted that  $(\eta_u)_{OPT}$  is virtually constant at about 0.44 for GROUP A and GROUP B, 2) designs between 170°C (isobutane mole fraction of 0.70) and 200°C (isobutane mole fraction of 0.26) and only changes significantly as the mixture becomes more dilute approaching either of the two pure fluids. The utilization efficiency of GROUP B, 1) optimum designs (with a factor of 5 above baseline fuel cost) are, of course, higher, but not comparable in the same context, because calculated brine return temperatures are well below 150°F. The utilization efficiencies are relatively constant, however, independent of resource temperature.

The constant  $\eta_u$  behavior for these economically optimum mixture cycles is a distinct departure from previously reported results for pure fluids (Refs. 15, 27) and is believed to be the result of; 1) the selection of the optimum fluid composition, and 2) selection of process states which minimize overall system thermodynamic irreversibilities (consistent with economics) for each resource temperature and constraint condition.

The change in  $\eta_u$  (and  $\eta_c$ ) at the two resource temperature extremes (dilute mixtures) is only shown approximately -- we had difficulty with convergence on the optimum designs using the MBWR mixture routines at both near zero and unity isobutane mixture mole fractions.

The foregoing sections have illustrated the now obvious importance of the well known specific heat anomaly (and the vapor saturation boundary) in the selection of working fluids and turbine operating states for Rankine cycle power plants.

The simple closed organic binary Rankine cycle geothermal power plant with the retrograde isobutane/isopentane hydrocarbon mixture system have been used to illustrate these extremely important characteristics. This study was limited to single phase brine production (pumped wells), a single hydrocarbon mixture system, an approximate equation of state, and used a very crude financial model and cost assumptions.

We have, however, investigated these characteristics for optimized designs with a wide variety of input sub-system unit cost and efficiency assumptions, two fundamentally different system Design Objectives (minimum busbar cost and maximum brine yield), and three possible system design constraint conditions:

- 1) Turbine inlet allowed to be anywhere outside the two-phase vapor envelope (moist and dry expansion with fixed expander stage pressure ratio).
- 2) Turbine inlet entropy  $\geq$  minimum saturated vapor entropy (dry expansion only).
- 3) Brine exit temperature  $\geq$  344.26 K (160°F).

Although the turbine states and mixture compositions vary depending upon resource temperature, unit fuel cost, Design Objective, and constraint assumptions, we have demonstrated a consistent behavior for all optimized designs:

- A. The optimum turbine inlet temperature is within 1% of the temperature of the TPCT line (at the optimum turbine inlet pressure), and the optimum turbine inlet pressure is within about 6% of the pressure on the TPCT line (at the optimum turbine inlet temperature).
- B. Moist turbine expansion is extremely detrimental to system economics for binary power plant cycles utilizing these retrograde (on T-S coordinates) hydrocarbon fluids.

Because of the foregoing, we feel the principles discussed may apply to varying degrees to all energy conversion systems with similar fluid systems operating on a Rankine cycle. We have therefore introduced the "Transposed Critical Temperature Rankine Thermodynamic Cycle" concept (Ref. 28).

(e) Potential Correlation for Optimized Geothermal Binary Rankine Cycles

Because of the very specific selected operating conditions ( $T_T$  and  $P_T$  at the TPCT line) and working fluid properties of these non-dilute TPCT cycles on reduced coordinates, a strong possibility existed that economic optimum system thermodynamic state parameters of this study could be "correlated" non-dimensionally with resource temperature. We were interested in determining whether or not a simple function of key optimum reduced cycle states coupled to optimum reduced fluid properties (which were found to be unique for given fuel cost, resource temperature, and constraint conditions) would plot smoothly against a ratio of the optimum turbine temperature,  $T_T$ , and the resource temperature.

After a limited search, we found that  $Z^*(F,C)$ , a "Fluid-Cycle" parameter, given by equation (2):

$$Z^*(F,C) = [(P_T/P_{Cr})(V_T/V_{Cr})(T_T/T_{Cond})]_{OPT} \quad (2)$$

plotted reasonably well against  $T^*(C,R)$ , a "Cycle-Resource" parameter, given by equation (3):

$$T^*(C,R) = (T_T)_{OPT}/T_{RES} \quad (3)$$

Because  $(T_T)_{OPT}$  is consistently in the neighborhood of the TPCT, we find that  $Z^*(F,C)$  given by equation (2) plots equally well against;

$$[(T_{TPCT})_{P_T}]_{T_{OPT}}/T_{RES}$$

The suggested "correlation" for this mixture system is shown in Figure 13. The correlation is exceptionally good for all the cost optimized cycles in GROUP A and GROUP B, 1) considering the fact that our system has seven independent thermodynamic state parameters but only four (the fluids' critical state, the turbine inlet state (2), and the condensing temperature) were needed to achieve the better than six percent  $Z^*$  "fit" (recall that sub-system costs were varied as much as 500%). All three groups exhibit only a 12.4%  $Z^*$  spread.

Another way to illustrate the "consistency" of these cost optimized designs is shown in Figure 14 where the independent parameter chosen is a "Fluid-Resource" parameter given by:  $T'(F,R) = (T_{CR}/T_{RES})_{OPT}$ .  $T'(F,R)$  exhibits only a two percent variation for all the optimum designs of this study.

We have included several tables of some of our calculated results in the Appendix for those who might be interested in investigating other possible "correlations" for these optimized cycles.

It is interesting to note in passing that  $T^*$  and  $T'$  for the SDG&E plant at 182°C (Ref. 19) look clearly inconsistent in both Figure 13 and Figure 14, whereas the 170°C point "groups" quite well. (See also Figure 4).

The foregoing  $Z^*$  correlation was presented because it, at last, suggests a simple functional relationship between process states, working fluid properties, and resource temperatures for optimized binary Rankine cycle geothermal systems. The correlation is strongly dependent upon the assumed equation of state and is not intended as a substitute for conventional design methods.

(f) A Brief Look at the SDG&E Proposed Demonstration Plant

This report would not be complete without a comparison between the currently proposed plant (Ref. 19) and our optimum conceptual designs. We have previously shown (Figure 4) that our optimum mixture compositions are at least in the same

ballpark as the initial condition design recommended by Holt in 1976 (Ref. 17), but markedly different than the "revised" Demo (Ref. 19).

To make an "exact" economic comparison of designs with GEOTHM requires a carefully done major sub-system cost coefficient normalization as described in Section 8.2.11.6 of Ref. 15. However, inadequate detailed sub-system cost information exists in Ref. 19 to do this. In fact, inadequate process state information is contained in Ref. 19 at both the beginning and end-of-life conditions to make even a thermodynamic comparison. Consequently, we had to perform very tedious, iterative heat, mass flow, and sub-system power comparisons to drive out the several un-specified assumptions.

However, when this thermodynamic "normalization" was finally complete, we found that by simply upgrading the fuel cost of Ref. 17 by a factor of 5 and assuming an overall plant capacity factor of 70%, we could adequately "predict" the Ref. 19 plant capital costs, brine costs, O&M costs, and busbar costs, with no significant changes to the direct and indirect cost factors that were used for the example problems in Ref. 15, Section 8.2.

The first comparison between SDG&E's system and our optimum design was done assuming Ref. 19's cycle and P&ID conditions. The second comparison was done assuming the general system assumptions of this report (Figure 1 and Table 1) with modified assumptions listed in Table 9. The relative goodness (or otherwise) of all important categories in each comparison was virtually the same, so we chose to report the second comparison results for brevity and clarity within this report.

Table 10 gives a brief summary of important economic and thermodynamic differences between the SDG&E proposed plant (by our analysis assuming the Starling specific MBWR and SDG&E's proposed process states) and what we feel represents a near optimum plant (see Figure 1) for the same system, equation of state, unit costs, financial factors, and constraint conditions.

First it may be noted that a conservative (see U factor footnote) four to five percent reduction in the busbar cost can be achieved if the mixture composition is changed between the 182°C condition and the assumed end-of-life condition, 170°C. Based on the Ref. 19 busbar cost of 134 mills/kwh, 5% corresponds to an annual savings of about two million dollars.

This savings comes about mainly because a significant (roughly 12%) improvement in utilization efficiency can be obtained with other than a 90% mixture composition and different independent process states (the 6 other optimizable parameters). Note (see Table 9) that our turbine inlet temperature for beginning-of-life conditions is much higher (15 C°) than Fluor's. See also Figures 5, 6, 13, and 14. This high turbine inlet temperature is largely the result of the 160°F minimum brine return temperature constraint.

Another significant difference between "them and us" is apparent in the primary heat exchanger (minimum) area requirements. We compute smaller optimum pinch points for the primary heater, and consequently require more surface area for the same primary heater U-factors. However, a 42% area change must be anticipated for the fixed composition design, whereas with the optimum compositions, the change is only about (we didn't look at intermediate temperature conditions) five percent.

When we look at the effect of the change in mixture (and turbine states and volumetric flow rates) on the turbine characteristics, we see one potential technical weakness in "our design" -- the 4.82% specific size change will produce a larger off-design efficiency degradation. We certainly do not profess expertise in this area, but a horse-back guess would be that the off-design efficiency degradation difference could be held to within roughly 1/2 percent of the SDG&E design with proper attention to specific speed and specific size selection details.

If it would cause no significant delay in getting the First Geothermal Binary Demonstration Plant on stream, the foregoing analysis suggests that a mixture change near mid-life is a good idea. It was apparently a good idea in 1976 (Ref. 17), and new economic trade-offs to justify the current proposed design are not adequately presented (Ref. 19).

## VII. POSSIBLE FUTURE INVESTIGATIONS

The correlations shown in Figures 5, 6, 13, and 14 and the previously shown minor cost and constraint assumption sensitivities, illustrate that the General Characteristics of busbar cost optimized, non-dilute  $iC_4/iC_5$  mixture binary Rankine cycle geothermal power plants can indeed be simply described. It is quite possible that optimized TPCT type simple binary cycle plants with other working fluid mixture systems would be equally well behaved over a limited resource temperature (mole fraction) range.

However, in future preliminary screening of other working fluid candidates, it might be desirable to slightly modify Kihara's and Fukunaga's I-factor (Ref. 21) definition:

$$I = 1 - \frac{ds}{dT} \left( \frac{T_{COND}}{C_p} \right)_{Sat.Vap.} = 1 - T \left( \frac{ds}{dh} \right)_{Sat.Vap.} \quad (4)$$

An  $I = 1$  implies that the candidate fluids' saturated vapor boundary coincides with an isentrope (at the dew point temperature). The above definition is convenient for initial working fluid screening assuming reversible (isentropic) expansion (Ref. 20), but a  $\Phi$ -factor given by:

$$\Phi = \bar{\eta}_e - T \left( \frac{ds}{dh} \right)_{Sat.Vap.} \quad (5)$$

would appear to be a more appropriate screening measure for non-reversible processes in the turbine.



With this latter definition, a thermodynamically desirable working fluid would have a saturated vapor locus at the dew point temperature "coincident" (or parallel if constrained) with the actual turbine "expansion" and a  $\Phi$ -factor of 1. Because of the previously shown undesirability of moist expansion in the turbine with hydrocarbon fluids in geothermal binary cycle applications, a reasonable value for  $\bar{\eta}_e$  in equation (5) above would be:  $\bar{\eta}_e \equiv (\bar{\eta}_e)_{\text{dry}} \approx 0.85$ .

Table 11 illustrates how the I-factor and  $\Phi$ -factor compare for three pure fluids assuming I-factor numerical values from reference 20\* and  $\bar{\eta}_e = 0.85$ .

|                                                    | <u>Table 11</u> |                  |                   |
|----------------------------------------------------|-----------------|------------------|-------------------|
|                                                    | <u>Propane</u>  | <u>Isobutane</u> | <u>Isopentane</u> |
| I-factor                                           | 1.39            | 0.81             | 0.77              |
| $T \left( \frac{ds}{dh} \right)_{\text{Sat.Vap.}}$ | -0.39           | 0.19             | 0.23              |
| $\Phi$ -factor<br>( $\bar{\eta}_e = 0.85$ )        | 1.24            | 0.66             | 0.62              |

A cursory examination of the critical properties and relative slopes of vapor saturation loci and isentropes (P-h plots) for these three fluids suggests that mixtures of propane and isobutane would be desirable for low temperature geothermal binary cycle applications, whereas mixtures of propane and isopentane might exhibit "desirable" thermodynamic characteristics for higher temperature geothermal applications. The first system, of course, has been previously investigated (Ref. 17), but over a limited set of resource temperatures and independent state conditions.

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\*Note that these I-factors (from Ref. 20) differ from those listed in reference 21, especially for propane.

When a new P-V-T correlation (based on Corresponding States principles) is available, we plan to continue these studies to lower and higher resource temperatures ( $150^{\circ}\text{C} \leq T_{\text{Res}} \leq 230^{\circ}\text{C}$ ) and include other saturated light hydrocarbon mixture constituents (propane, etc.).

#### VIII. CONCLUSIONS

- The transposed critical temperature line in the region of the anomalous specific heat has been shown to be an important thermodynamic property which should be given serious consideration in the selection of working fluids and turbine states for geothermal binary Rankine cycles. Accuracy in the location of the vapor saturation boundary to avoid moist turbine expansion is critical to the economics of geothermal binary cycles which will utilize these hydrocarbon fluids.

- Although this study was limited to simple closed binary Rankine cycle geothermal power plants with a single mixture system and a limited resource temperature range, the TPCT results suggest much broader potential applicability which should be investigated in future studies of working fluid selection techniques for Rankine cycles.

- Optimized geothermal mixture binary Rankine cycles appear to offer significant potential utilization efficiency and economic advantages over pure fluid binary Rankine cycles, and could play an important role in electric power development at moderate temperature geothermal resources. However, we have shown that the chosen system Design Objective can have a significant impact on the selected optimum mixture composition. When maximum specific net energy is the Design Objective in a constrained brine yield optimization, significant plant sub-system overdesign may result, the wrong mixture composition may be selected, and the resulting busbar cost may be as much as 6% too high--compromising commercial feasibility.

- A high priority should be placed on the development of a new, more accurate equation-of-state for hydrocarbon mixtures. Current DGE funded PVT property measurements and Corresponding States development work by NBS (Gaithersburg and Boulder) should result in significant improvements over existing BWR-type mixture formulations with modest additional data investments. A better thermodynamic properties foundation in the system simulation will permit a much more accurate picture of the economic advantage of mixtures over pure fluids.

#### Other Potential Applications - Topping and Bottoming Cycles

Because of increasing demand for electric power with continually increasing incentives to conserve fuel, improvements in conventional central steam power station efficiencies become imperative. The only practical way to achieve significant efficiency improvements with steam power plants appears to be with topping processes in view of the technological problems associated with metallurgical limits.

Although gas turbine (Brayton cycle) topping processes have been used in a large number of power stations (Ref. 29), and recent improvements in Brayton cycle efficiency have been achieved for marine applications (Ref. 30), the Rankine cycle is thermodynamically superior to the Brayton cycle (Ref. 29) for topping purposes.

Many of the technological problems associated with the use of alkali-metals in the Rankine topping process have been thoroughly investigated for breeder reactors and space applications (Ref. 31, 32), so renewed attention to this type of advanced steam plant appears warranted (Ref. 33). The TPCT concepts discussed herein for geothermal applications may be useful in the selection of working fluids and process states for other supercritical systems and these advanced steam power plants.

The foregoing TPCT concepts can be employed in Diesel bottoming cycle applications now. Organic Rankine bottoming cycles can increase the output power from Diesel plants by 15% or more and quickly pay for themselves (Ref. 34).

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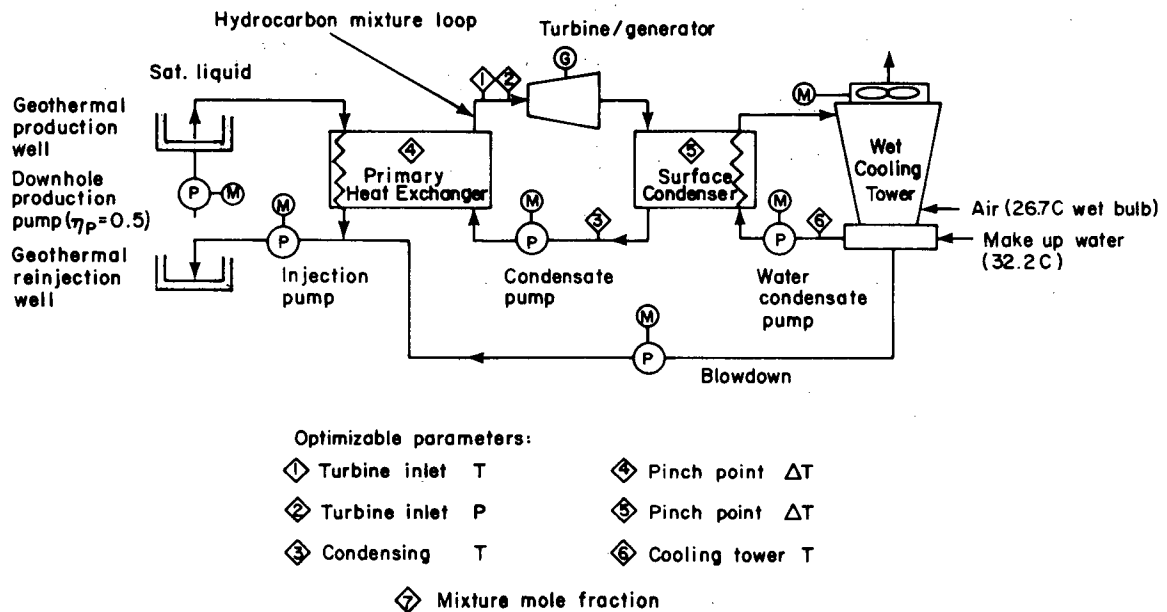
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XBL806-1134

**Fig. 1**

Simplified schematic of a simple organic mixture binary Rankine cycle power plant without regenerative heat exchange. Seven optimizable parameters completely characterize the thermodynamic performance of the plant. Input assumptions for the power plant at baseline conditions are listed in Table 1. For various non-baseline input assumptions, see Figure 2.

## TPCT STUDY PLOT DATA POINT LEGEND

### Group A. Un-Constrained Cost Optimizations

- 1) Various Resource Temperatures ( $170\text{C} \leq T_{\text{res}} \leq 200\text{C}$ )
  - ⊙ Baseline Cost and Efficiency Assumptions‡ (See Table 1)
- 2) Various Cost or Efficiency Assumptions (@ 182C)
  - ▽ Fuel cost up by factor of 5
  - ▣ Primary heat exchanger cost up by factor of 2
  - Hydrocarbon pump and motor cost up by factor of 5
  - ◆ HC pump adiabatic efficiency reduced 0.8 → 0.6

### Group B. Constrained Cost Optimizations

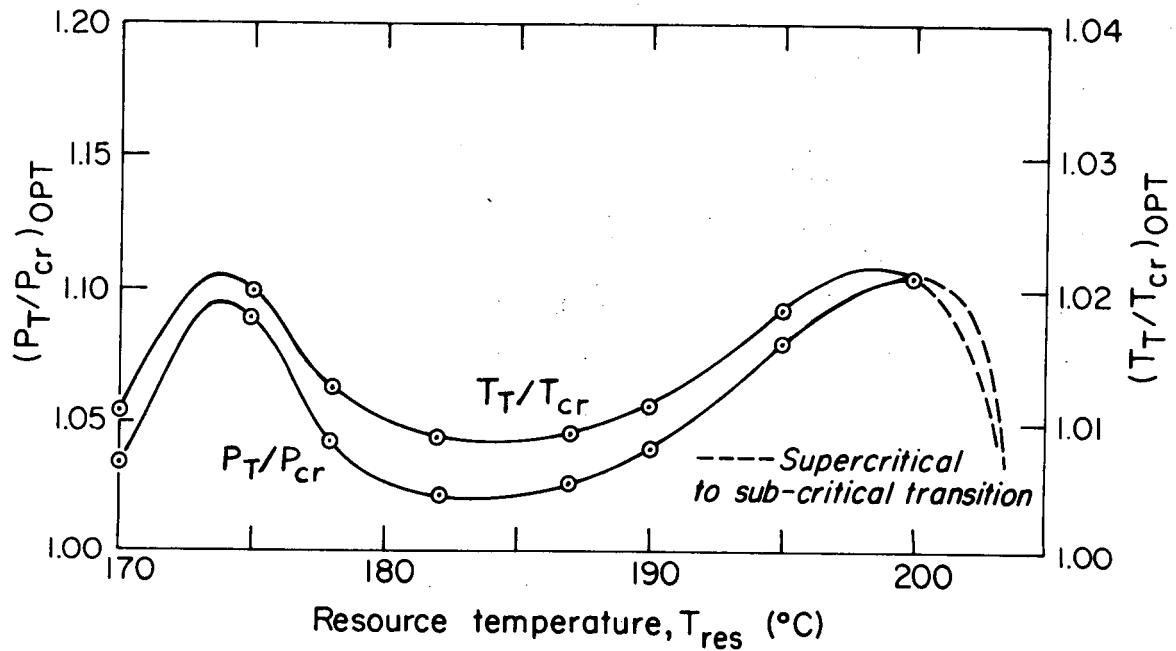
- 1) Turbine Inlet Entropy  $\geq$  Maximum Saturated Vapor Entropy
  - Fuel cost up by factor of 5 (170C)
  - ⊗ Fuel cost up by factor of 5 (175C)
  - ⬠ Fuel cost up by factor of 5 (178C)
  - ⊗ Fuel cost up by factor of 5 (182C)
  - ⊕ Baseline cost and efficiency assumptions (182C)
  - + Primary heat exchanger cost up by factor of 2 (182C)
  - × Primary heat exchanger cost up by factor of 2.5 (182C)
- 2) Brine Return Temperature  $\geq 344.26\text{K}$  (160F)
  - (Plant availability factor = 0.7)
  - ▼ Fuel cost up by factor of 5 (170C)
  - ▲ Fuel cost up by factor of 5 (175C)
  - ▷ Fuel cost up by factor of 5 (178C)
  - ◇ Fuel cost up by factor of 5 (182C)
  - same as B-2) except  $U_c = 283.35 \text{ W/m}^2\text{K}$  (50 Btu/hr ft<sup>2</sup>F)—
  - ⊗  $T_{\text{res}} = 170\text{C}$  [Basis for economic comparison with
  - ⊗  $T_{\text{res}} = 182\text{C}$  [SDG&E proposed demonstration plant]

(Ref. 19)

‡Plant subsystem capital costs normalized to Ref. 17 (Heber Binary) with Brine cost normalized on \$/MBtu basis (Ref. 15, Section 8.2, 11.6)

Fig. 2

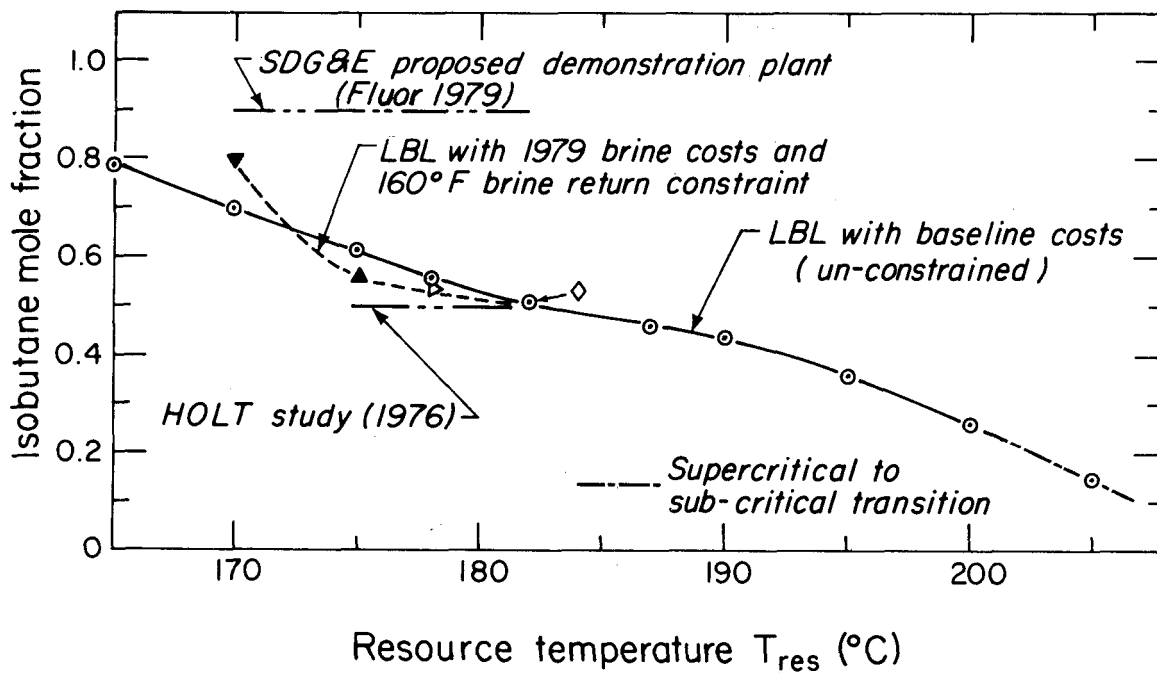
Resource temperature, sub-system unit cost and efficiency, and constraint condition variations investigated in this study. Except as noted here, all input assumptions are as shown in Table 1. Each data point corresponds to a separate, 7 parameter optimized system.



XBL 806-1139

**Fig. 3**

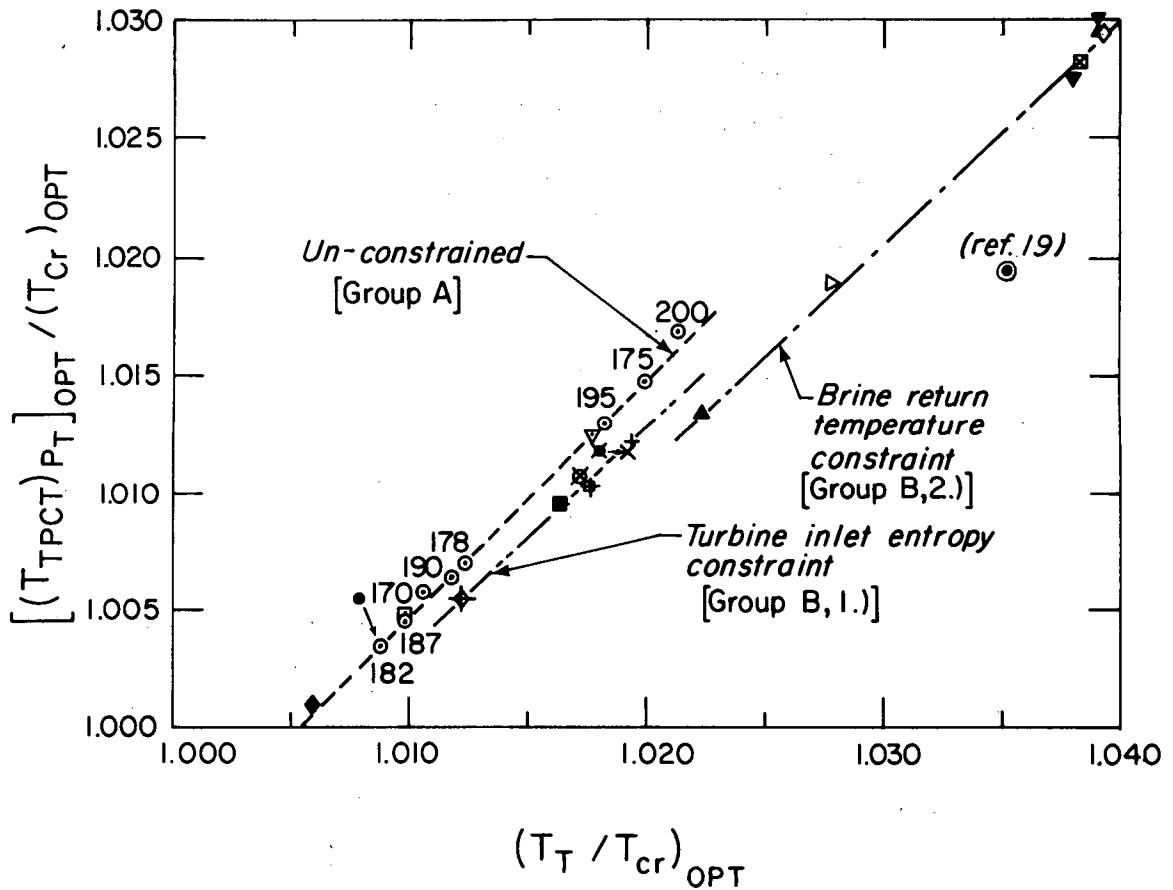
Computed busbar cost optimum turbine inlet states as a function of resource temperature for power plants of Figure 1 with the  $iC_4/iC_5$  mixture working fluid under baseline cost and efficiency input assumptions of Table 1. The plotted points correspond to GROUP A conditions, Figure 2 (7 parameter un-constrained optimizations).



XBL 805-963A

Fig. 4

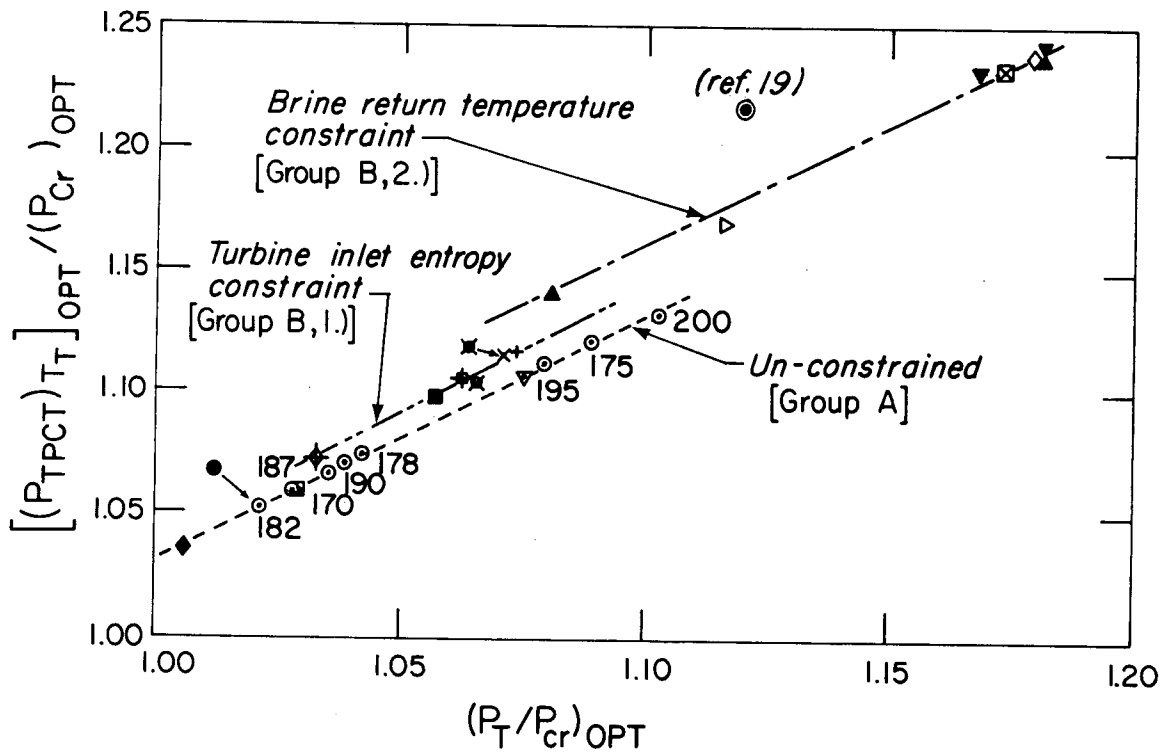
Computed optimum  $iC_4$  mole fraction as a function of resource temperature for power plants of Figure 1 with the  $iC_4/iC_5$  mixture working fluid for GROUP A (baseline costs, un-constrained) and GROUP B, 2.) conditions of Figure 2 (see Table 2 and Table 4 for details). Also shown above are the recommended working fluid mixtures for the binary cycle of Figure 1 at Heber resource conditions from two previous studies.



XBL 805-965A

**Fig. 5**

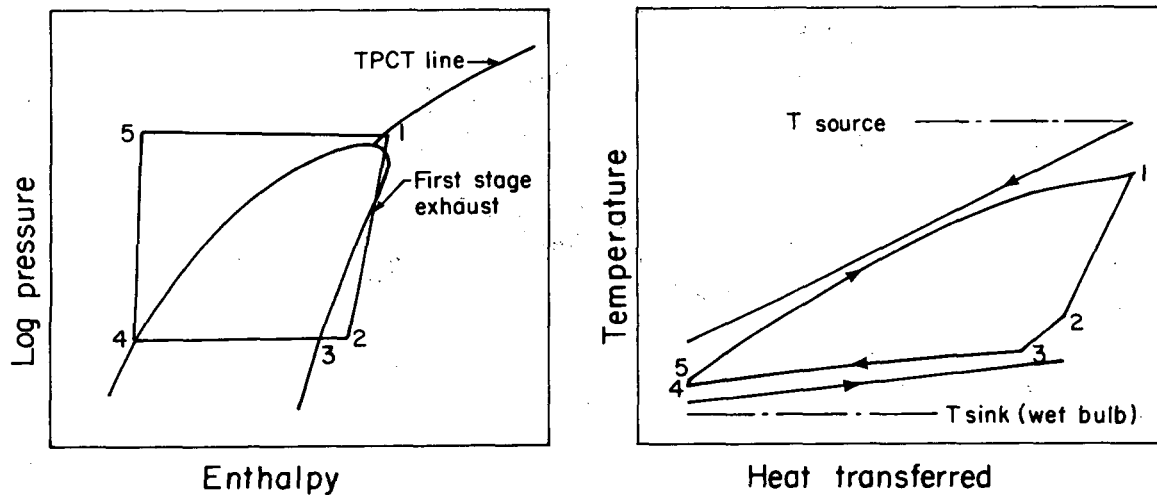
Correlation between the temperature on the transposed critical temperature line (at the optimum turbine inlet pressure) and the optimum turbine inlet temperature for busbar cost optimized  $iC_4/iC_5$  binary Rankine cycle power plants of Figure 1 for all sub-system unit cost and efficiency, constraint conditions, and resource temperatures listed in Figure 2 (7 parameter optimizations).



XBL 805-964 A

Fig. 6

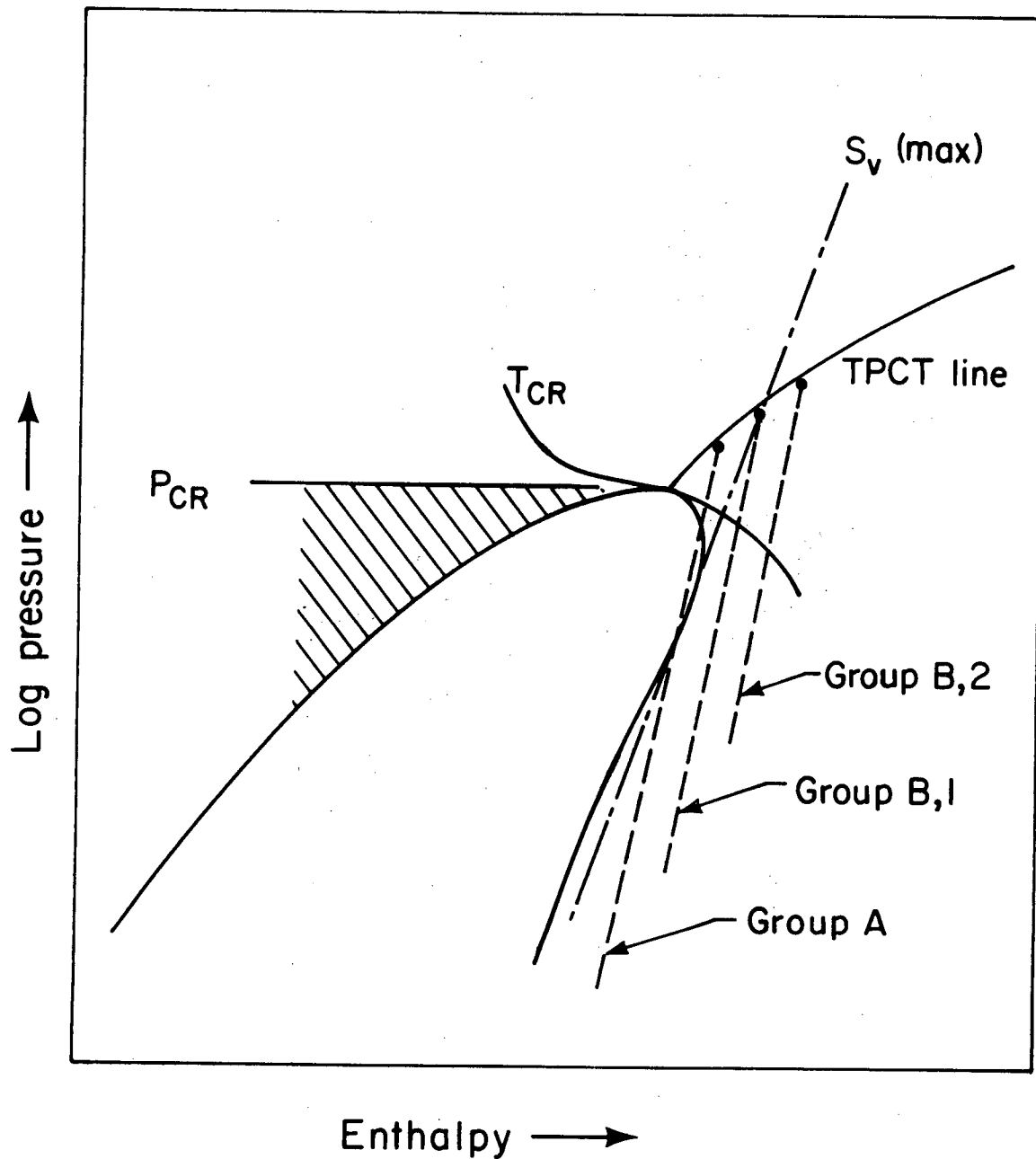
Correlation between the pressure on the transposed critical temperature line (at the optimum turbine inlet temperature) and the optimum turbine inlet pressure for optimized binary Rankine cycle power plants of Figure 1.



XBL 806-1137

Fig. 7

Typical process state diagrams on P-h and T-Q coordinates for all optimized binary Rankine cycle power plants of Figure 1 with the  $iC_4/iC_5$  mixture working fluid for GROUP A (un-constrained) conditions of Figure 2. The curves shown were traced from computer plots for the 195°C resource case at baseline (Table 1) conditions. See Table 2 for details.



XBL806-1138

Fig. 8

Pressure enthalpy sketch illustrating typical turbine expansion paths for each constraint group of this study. Turbine inlet states in the two-phase and shaded region were disallowed in all cases. Differences in optimum composition (vapor dome location and shape) were ignored for illustration purposes.



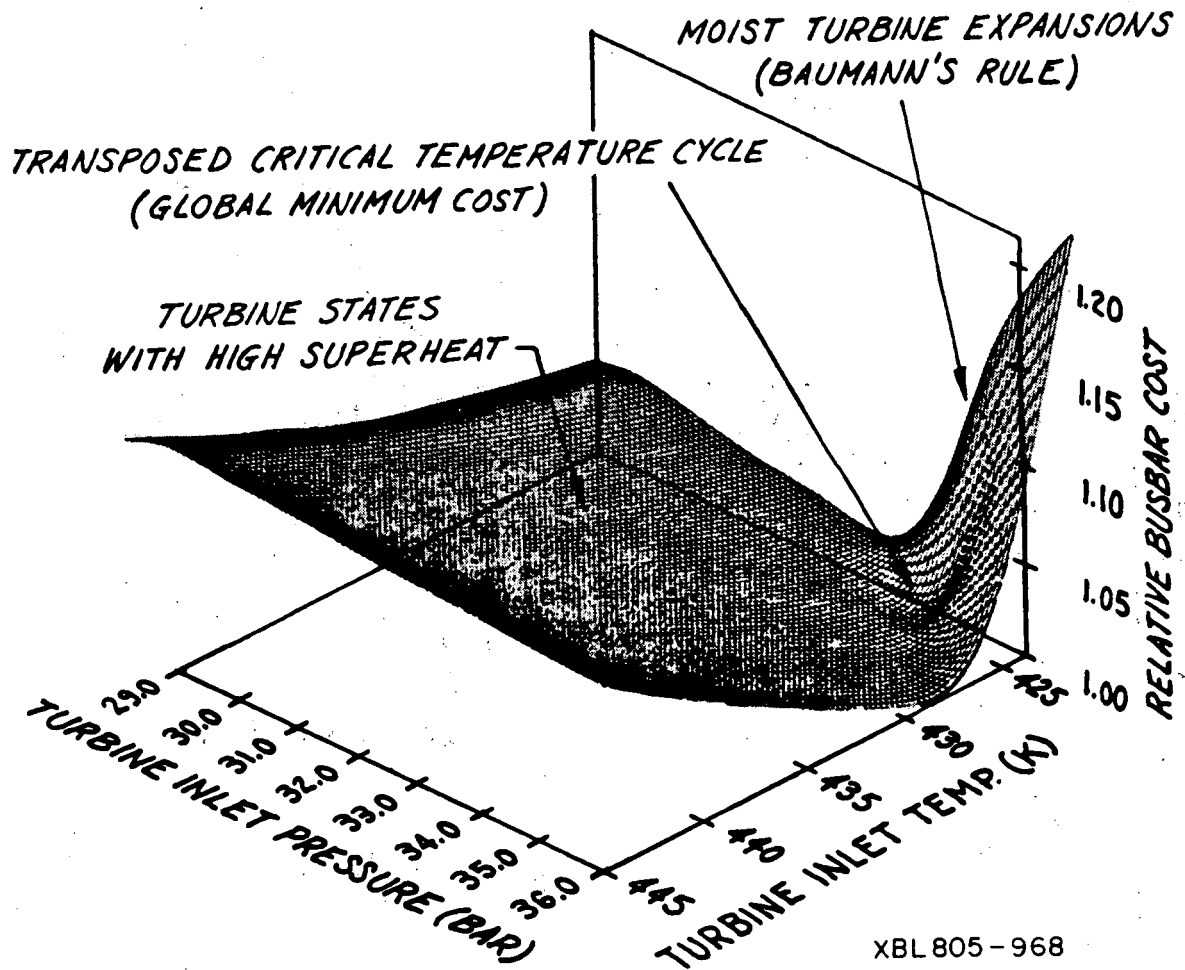
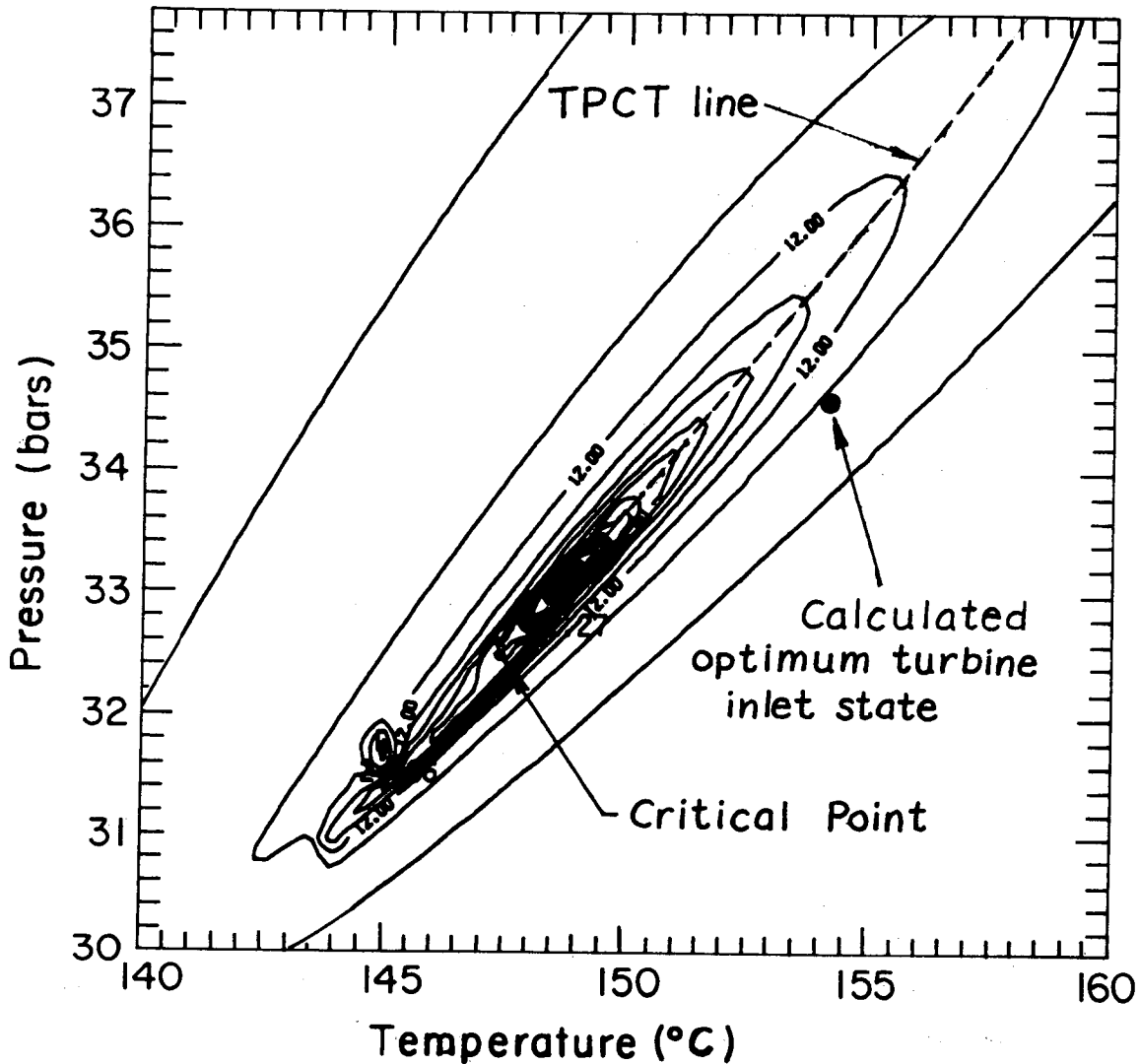


Fig. 9

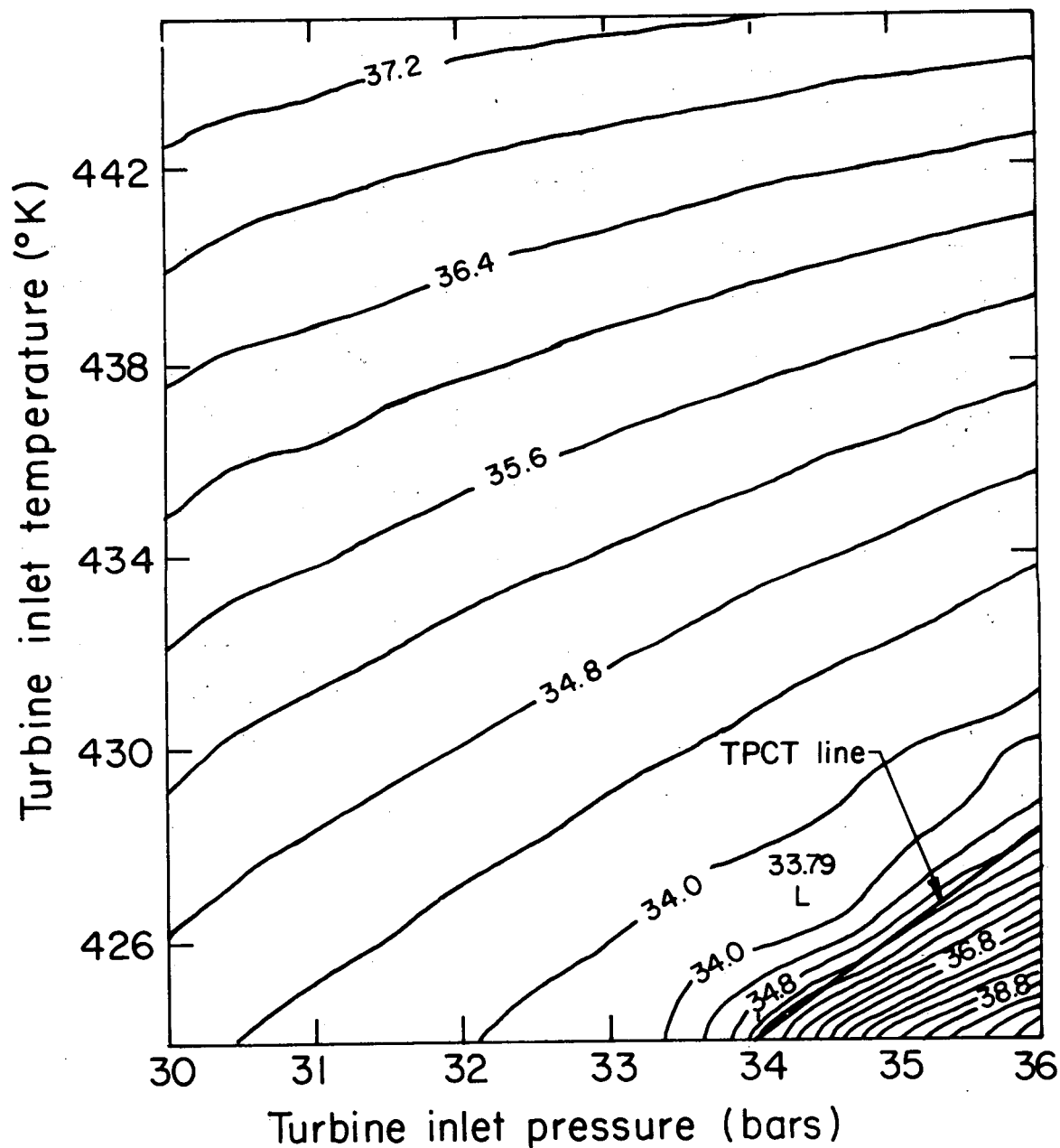
Relative busbar energy cost design surface as a function of turbine inlet pressure,  $P_T$  and temperature,  $T_T$ . Busbar energy cost optimized hydrocarbon mixture ( $iC_4/iC_5$ ) binary Rankine cycle geothermal power plants of 50 MWe (net) capacity. 182°C resource temperature. Figure reproduced from Ref. 28. (See also [Figure 10](#) and [Figure 11](#).)



XBL 806-1135

Fig. 10

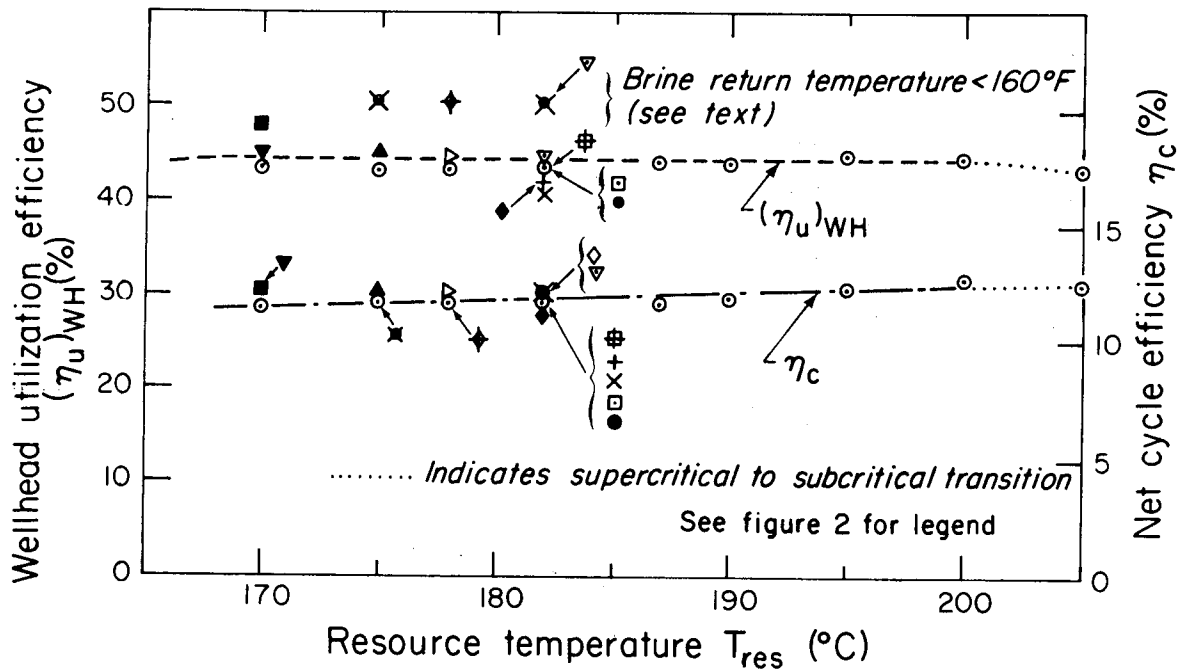
Contour plot of the heat capacity at constant pressure for isobutane/isopentane mixture (0.5639/0.4361) as a function of temperature and pressure illustrating relative location (typical) of optimum turbine inlet state with respect to TPCT line. 50 MWe (net) mixture binary cycle. Figure reproduced from reference 28. (Documentation: L.F.S. Cp computation, 11/30/79).



XBL 806-1136

Fig. 11

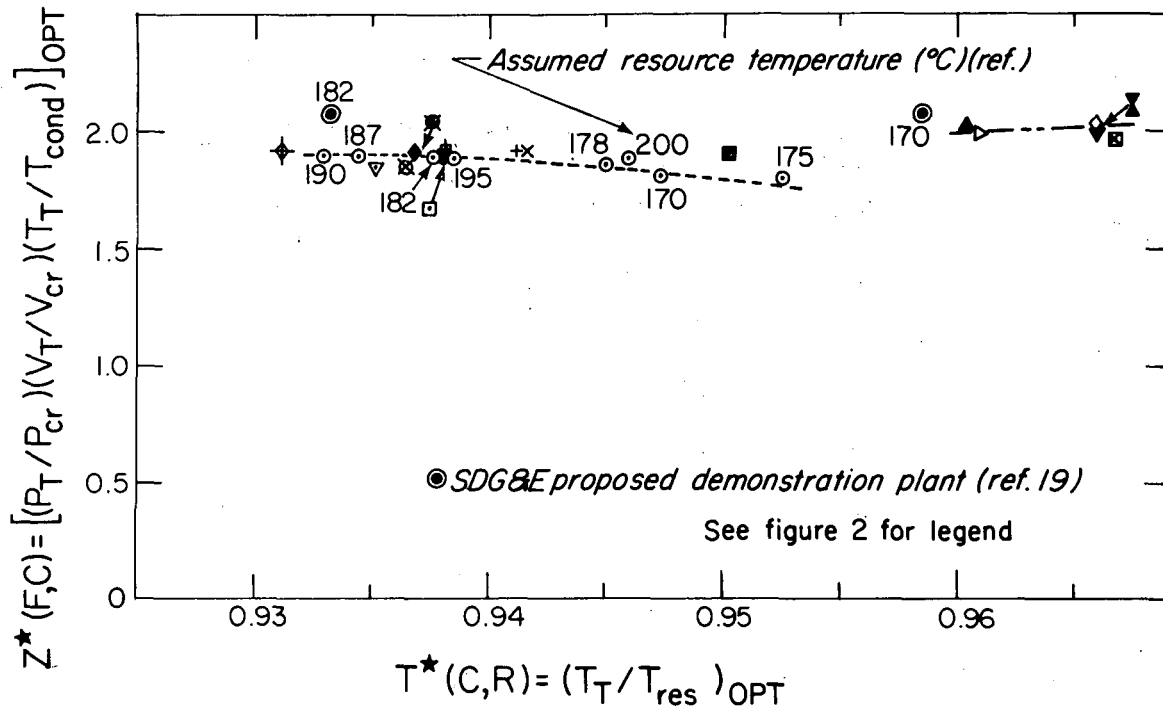
Busbar energy cost design surface contour plot on turbine inlet coordinates for optimized 50 MWe (net) hydrocarbon mixture binary Rankine geothermal power plants. Resource temperature = 182°C, wet bulb temperature = 26.7°C. TPCT line was obtained from Figure 10. Figure reproduced from Reference 28. See corresponding 3-D plot, Figure 9. (Documentation: PADRA37, 11/16/79).



XBL 806-1140

Fig. 12

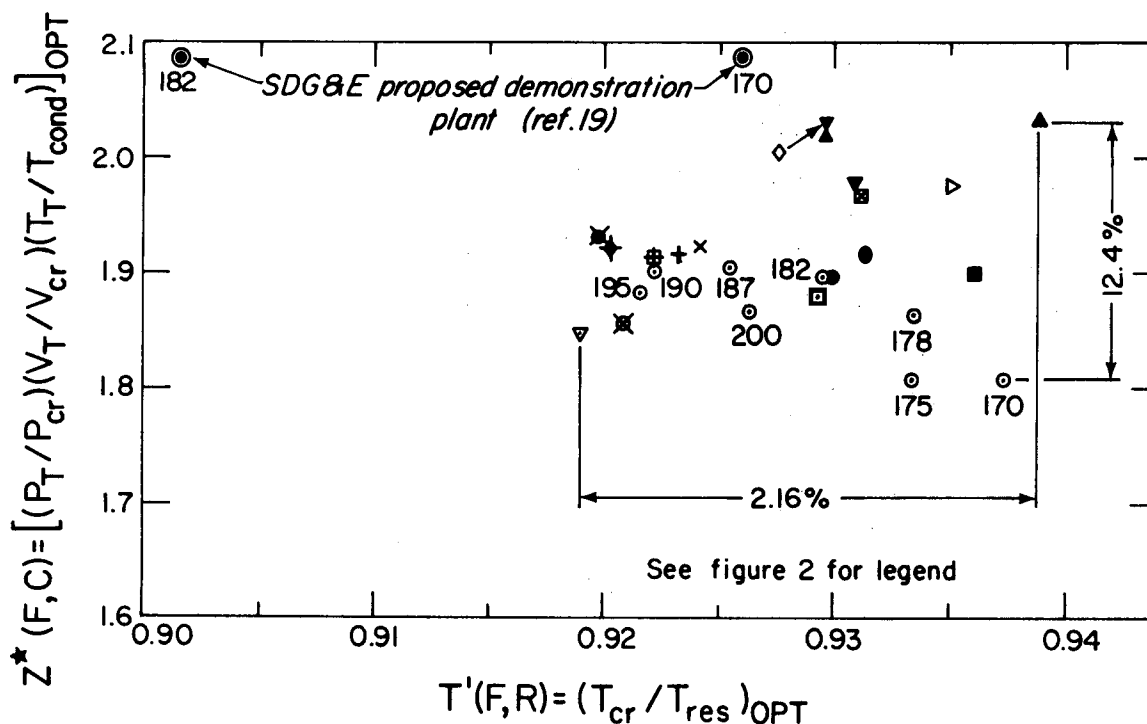
Computed utilization efficiency,  $P_{NET}/\dot{m}_b|\Delta h_b|_s$ , (wellhead to wet bulb) and overall plant net thermodynamic efficiency,  $P_{NET}/Q_{in}$ , as a function of resource temperature for busbar cost optimized iC4/iC5 mixture binary Rankine cycle power plants of Figure 1 for all sub-system cost and efficiency and constraint conditions listed in Figure 2. See Tables 1, 2, 3, and 4 for details.



XBL 806-1165

Fig. 13

Correlation of cycle states:  $T_T$ ,  $P_T$ ,  $T_{cond}$ ; working fluid critical point:  $P_{cr}$ ,  $T_{cr}$ ; and reservoir temperature,  $T_{res}$ , for busbar cost optimized  $iC_4/iC_5$  mixture binary Rankine cycle power plants of Figure 1 for all sub-system cost and efficiency and constraint conditions listed in Figure 2. The group in the upper right corner correspond to GROUP B, 2.) conditions (brine return temperature  $\geq 160^\circ F$ ). Reference 19 plotted points correspond to the SDG & E proposed binary Demonstration Plant process state conditions by our calculations assuming the Starling specific MBWR (Ref. 23). The SDGE point (at  $182^\circ C$ ) suggests that the turbine inlet temperature is inconsistent (too low) with the  $160^\circ F$  brine return constrained group (see also Figure 14).



XBL 806-1141

Fig. 14

This plot depicts the  $Z^*$  function of Figure 13 plotted against the dimensionless temperature ratio,  $(T_{cr})_{opt}/T_{res}$ , for all conditions investigated in this study. The SDG & E point plotted in the upper left hand corner suggests that the Fluor (Ref. 19) selected mixture composition (for  $T_{res} = 182^\circ\text{C}$ ) is clearly inconsistent with other results of this study ( $T_{cr}$  too low).

TABLE 1 Baseline Input Assumptions For All Analyses Except as Noted in Text

| Parameter                                                     | Baseline Assumption                   |
|---------------------------------------------------------------|---------------------------------------|
| Produced Fluid State at Wellhead                              | Saturated liquid at Tres              |
| "Brine" Salinity                                              | 0.0 (Pure H <sub>2</sub> O)           |
| Drawdown Factor (KPa/Kg/sec)                                  | 22.8                                  |
| Well Depth (m)                                                | 1830.                                 |
| Well Friction and Heat Transfer                               | Ignored                               |
| Net Cycle Output Power (MWe)                                  | 50.                                   |
| Plant Capacity Factor                                         | 0.85                                  |
| Design Wet Bulb Temperature (°C)                              | 26.7                                  |
| Make-up Water Temperature (°C)                                | 32.2                                  |
| Hydrocarbon Expander:                                         |                                       |
| Dry Stage Adiabatic Efficiency                                | 0.85                                  |
| Maximum Stage Pressure Ratio                                  | 1.8                                   |
| Stage Efficiency Reduction<br>(% drop per % exhaust moisture) | 1.0                                   |
| Generator Efficiency                                          | 0.98                                  |
| Motor Efficiency (all)                                        | 0.95                                  |
| Pump Efficiency (all surface pumps)                           | 0.80                                  |
| Down-hole Production Pump                                     | 0.50                                  |
| Overall Heat Transfer Coefficients (W/m <sup>2</sup> °K)      |                                       |
| Supercritical Primary H.X.                                    | 1514.                                 |
| Sub-critical Primary H.X.                                     |                                       |
| • Pre-heating                                                 | 1514.                                 |
| • Boiling                                                     | 2422.4                                |
| • Superheating                                                | 1514.                                 |
| Condensing                                                    | 566.7                                 |
| Desuperheating                                                | 237.5                                 |
| Sub-System Costing:                                           | (See also Reference 15, Section 8.2)  |
| Capital Cost Equations                                        | GEOTHM MODEL (Ref. 2)                 |
| Capital Cost Coefficients                                     | Normalized to Ref. 17                 |
| O & M Costs                                                   | Normalized to Ref. 17                 |
| Direct and Indirect Cost Factors                              | Normalized to Ref. 17                 |
| Brine Cost (\$/MBtu)                                          | Normalized to Ref. 17 (See Text)      |
| Water Properties                                              | GEOTHM MODEL (Ref. 2)                 |
| Working Fluid Properties                                      | Starling specific MBWR (1975) Ref. 23 |

TABLE 2 GROUP A - UN-CONSTRAINED BUSBAR COST OPTIMIZATIONS  
ISOBUTANE/ISOPENTANE SYSTEM - CASE STUDY RESULT DETAILS

| Resource           | OPTIMUM WORKING FLUID     |                  |                   |                     | OPTIMUM CYCLE STATES |                |                  |                    | $T_{TPCT}@$<br>$(P_T)_{OPT}$ | $P_{TPCT}@$<br>$(T_T)_{OPT}$ | DOCUMENTATION<br>(DATE) |
|--------------------|---------------------------|------------------|-------------------|---------------------|----------------------|----------------|------------------|--------------------|------------------------------|------------------------------|-------------------------|
| Temp<br>°C<br>(°K) | ISOB.<br>mole<br>fraction | $T_{cr}$<br>(°K) | $P_{cr}$<br>(bar) | $V_{cr}$<br>(cc/gm) | $T_T$<br>(°K)        | $P_T$<br>(bar) | $V_T$<br>(cc/gm) | $T_{COND}$<br>(°K) |                              |                              |                         |
| 170<br>(443.15)    | 0.69905                   | 415.385          | 33.327            | 5.305               | 419.832              | 34.5166        | 6.8814           | 312.153            | 417.839                      | 35.4862                      | PADRA15<br>(4/15/80)    |
| 175<br>(448.15)    | 0.616434                  | 418.459          | 32.784            | 5.312               | 426.858              | 35.7174        | 6.4305           | 311.467            | 424.648                      | 36.7746                      | PADRA38<br>(3/19/80)    |
| 178<br>(451.15)    | 0.554501                  | 421.148          | 32.486            | 5.306               | 426.356              | 33.8603        | 6.9209           | 311.056            | 424.103                      | 34.9147                      | TPCTØØ3<br>(3/19/80)    |
| 182<br>(455.15)    | 0.513135                  | 423.130          | 32.339            | 5.295               | 426.828              | 33.0328        | 7.1534           | 310.791            | 424.637                      | 34.0465                      | TPCTØ1Ø<br>(3/31/80)    |
| 187<br>(460.15)    | 0.460718                  | 425.856          | 32.211            | 5.273               | 430.020              | 33.1000        | 7.0594           | 310.488            | 427.799                      | 34.1198                      | TPCTØØ4<br>(3/19/80)    |
| 190<br>(463.15)    | 0.438665                  | 427.076          | 32.177            | 5.261               | 432.109              | 33.4327        | 6.9250           | 310.701            | 429.825                      | 34.4802                      | TPCTØ11<br>(3/20/80)    |
| 195<br>(468.15)    | 0.364743                  | 431.484          | 32.147            | 5.209               | 439.360              | 34.7056        | 6.4199           | 310.331            | 437.099                      | 35.7406                      | TPCTØ12<br>(3/20/80)    |
| 200<br>(473.15)    | 0.264649                  | 438.261          | 32.321            | 5.106               | 447.612              | 35.7210        | 5.9654           | 309.383            | 445.701                      | 36.5984                      | TPCTØØ4<br>(3/31/80)    |

PERTURBED INPUT ASSUMPTIONS - ALL FOR 182°C RESOURCE TEMPERATURE

|                            |          |         |        |       |         |         |        |         |         |         |                      |
|----------------------------|----------|---------|--------|-------|---------|---------|--------|---------|---------|---------|----------------------|
| FUEL COST<br>UP 5X         | 0.621861 | 418.239 | 32.814 | 5.312 | 425.670 | 35.2815 | 6.5632 | 305.893 | 423.442 | 36.3465 | TPCTØ19<br>(3/21/80) |
| PRIM. RXR<br>\$ UP 2X      | 0.517120 | 422.932 | 32.352 | 5.296 | 427.101 | 33.2926 | 7.0263 | 310.312 | 424.974 | 34.2785 | TPCTØ11<br>(3/25/80) |
| H.C. PUMP<br>\$ UP 5X      | 0.512824 | 423.145 | 32.338 | 5.294 | 426.874 | 33.0437 | 7.1499 | 310.765 | 424.679 | 34.0595 | TPCTØ16<br>(3/20/80) |
| $\eta_r$ DOWN<br>0.8 → 0.6 | 0.497880 | 423.898 | 32.295 | 5.289 | 426.409 | 32.4911 | 7.3238 | 309.678 | 424.328 | 33.4498 | TPCTØ22<br>(3/27/80) |



TABLE 3 GROUP B, 1) RESULT DETAILS  
CONSTRAINED BUSBAR COST OPTIMIZATIONS

4/10/80

— TURBINE INLET ENTROPY  $\geq$  MAXIMUM SATURATED VAPOR ENTROPY —

| RESOURCE<br>TEMP.<br>°C<br>(°K) | OPTIMUM WORKING FLUID     |                  |                   | OPTIMUM CYCLE STATES |                |                  |                    | $T_{TPCT}$ @<br>( $P_T$ ) <sub>OPT</sub> | $P_{TPCT}$ @<br>( $T_T$ ) <sub>OPT</sub> | $(\eta)_{U/WH}^*$<br>(%) | DOCUMENT-<br>TATION<br>(DATE) |
|---------------------------------|---------------------------|------------------|-------------------|----------------------|----------------|------------------|--------------------|------------------------------------------|------------------------------------------|--------------------------|-------------------------------|
|                                 | ISOB.<br>MOLE<br>FRACTION | $T_{cr}$<br>(°K) | $P_{cr}$<br>(bar) | $T_T$<br>(°K)        | $P_T$<br>(bar) | $v_T$<br>(cc/gm) | $T_{COND}$<br>(°K) |                                          |                                          |                          |                               |

FUEL COST INCREASED BY FACTOR OF 5

|                 |         |         |        |         |        |        |         |         |        |        |                      |
|-----------------|---------|---------|--------|---------|--------|--------|---------|---------|--------|--------|----------------------|
| 170<br>(443.15) | 0.71622 | 414.820 | 33.461 | 421.555 | 35.407 | 6.9113 | 306.25  | 418.804 | 36.759 | 49.320 | PADRA38<br>(4/9/80)  |
| 175<br>(448.15) | 0.79073 | 412.660 | 34.134 | 419.679 | 36.357 | 6.7464 | 307.542 | 417.076 | 37.673 | 50.606 | TPCT027<br>(4/2/80)  |
| 178<br>(451.15) | 0.70547 | 415.171 | 33.376 | 420.200 | 34.484 | 7.2038 | 306.671 | 417.449 | 35.825 | 50.581 | TPCT032<br>(4/3/80)  |
| 182<br>(455.15) | 0.61303 | 418.598 | 32.765 | 426.515 | 35.108 | 6.8686 | 305.956 | 423.553 | 36.521 | 50.438 | TPCT000<br>(4/11/80) |

EFFECT OF PRIMARY HEAT EXCHANGER UNIT COST (ALL FOR  $T_{res} = 182^\circ\text{C}$  AND BASELINE FUEL COST)

|                               |         |         |        |         |        |        |         |                 |                 |        |                      |
|-------------------------------|---------|---------|--------|---------|--------|--------|---------|-----------------|-----------------|--------|----------------------|
| BASILINE<br>HXR UNIT<br>COSTS | 0.58804 | 419.652 | 32.636 | 427.040 | 34.678 | 6.9853 | 311.588 | 424.003         | 36.116          | 43.711 | TPCT034<br>(4/3/80)  |
| \$UP 2.0X                     | 0.57447 | 420.245 | 32.572 | 428.400 | 34.965 | 6.8761 | 310.532 | 425.354         | 36.405          | 42.163 | PADRA32<br>(3/27/80) |
| \$UP 2.5X                     | 0.56788 | 420.539 | 32.543 | 428.568 | 34.858 | 6.9080 | 310.536 | 425.492         | 36.310          | 40.986 | PADRA35<br>(3/27/80) |
| \$UP 3.0X                     | 0.50142 | 423.718 | 32.305 | 423.551 | 30.787 | 8.5102 | 309.286 | NA <sup>‡</sup> | NA <sup>‡</sup> | 40.183 | TPCT009<br>(3/31/80) |

\* BRINE INJECTION TEMPERATURE DETERMINED BY SYSTEM ECONOMICS — NOT CONSTRAINED —

‡ NOT APPLICABLE — SUB-CRITICAL CYCLE IS OPTIMUM

TABLE 4 GROUP B, 2.) RESULT DETAILS

4/15/80

CONSTRAINED BUSBAR COST OPTIMIZATIONS

-BRINE RETURN TEMPERATURE  $\geq 344.26$  K (160 F) -

| RESOURCE<br>TEMP.<br>°C<br>(°K) | OPTIMUM WORKING FLUID      |                  |                   | OPTIMUM CYCLE STATES |                |                  |                    | $T_{TPCT}$ @<br>( $P_T$ ) <sub>OPT</sub> | $P_{TPCT}$ @<br>( $T_T$ ) <sub>OPT</sub> | $(\eta_u)_{WH}$<br>$(\eta_c)_{NET}$ | DOCUMENTATION<br>(DATE) |
|---------------------------------|----------------------------|------------------|-------------------|----------------------|----------------|------------------|--------------------|------------------------------------------|------------------------------------------|-------------------------------------|-------------------------|
|                                 | $iC_4$<br>MOLE<br>FRACTION | $T_{cr}$<br>(°K) | $P_{cr}$<br>(bar) | $T_T$<br>(°K)        | $P_T$<br>(bar) | $v_T$<br>(cc/gm) | $T_{COND}$<br>(°K) |                                          |                                          |                                     |                         |
| 170<br>(443.15)                 | 0.79758                    | 412.486          | 34.203            | 428.151              | 39.969         | 6.4607           | 309.919            | 423.842                                  | 42.196                                   | 44.76<br>11.97                      | PADRAØ9<br>(4/7/80)     |
| 175<br>(448.15)                 | 0.56361                    | 420.782          | 32.524            | 430.428              | 35.154         | 7.1153           | 306.742            | 426.358                                  | 37.077                                   | 44.98<br>12.18                      | PADRA23<br>(4/9/80)     |
| 178<br>(451.15)                 | 0.53943                    | 421.853          | 32.428            | 433.510              | 36.213         | 6.6495           | 306.859            | 429.969                                  | 37.885                                   | 44.76<br>12.22                      | PADRAØ7<br>(4/7/80)     |
| 182<br>(455.15)                 | 0.51361                    | 423.106          | 32.341            | 439.630              | 38.138         | 6.3893           | 307.690            | 435.541                                  | 40.082                                   | 44.54<br>12.29                      | TPCTØ18<br>(4/8/80)     |

ASSUMPTIONS:

- 1) BASELINE PLANT CAPITAL COST, EFFICIENCY, AND U-FACTOR ASSUMPTIONS
- 2) PLANT AVAILABILITY FACTOR = 0.70
- 3) FUEL COST UP BY A FACTOR OF 5
- 4) SEE TABLE 1 AND FIGURE 1 FOR OTHER DETAILS

TABLE 5

Effect of Subsystem Cost and Efficiency Assumptions  
on Computed Optimum Mixture Fraction for Cost Optimized System  
( $T_{\text{Res}} = 182^{\circ}\text{C}$ , expander max. stage pressure ratio = 1.8)

| <u>Input Assumption</u>                                                          | <u>Relative Optimum Isobutane<br/>Mole Fraction</u> | <u>Relative Optimum<br/>Busbar Cost (Ref)</u> |
|----------------------------------------------------------------------------------|-----------------------------------------------------|-----------------------------------------------|
|                                                                                  | (Perturbed/Baseline)                                | (Perturbed/Baseline)                          |
| Baseline Costs and<br>Efficiency Factors                                         | 1.0000                                              | 1.000                                         |
| Increased Fuel Cost (\$/MBtu)<br>by Factor of 5                                  | 1.2119                                              | 2.747                                         |
| Increased Primary Heater<br>Unit Cost (\$/m <sup>2</sup> ) by<br>Factor of 2     | 1.0078                                              | 1.078                                         |
| Increased H.C. Circulating<br>Pump and Motor Unit Cost<br>(\$/kW) by Factor of 5 | 0.9994                                              | 1.128                                         |
| Decreased H.C. Circulating<br>Pump Adiabatic Efficiency<br>from 80% to 60%       | 0.9703                                              | 1.058                                         |

TABLE 6

Comparison of Independent Thermodynamic State Parameters  
for Busbar Cost and Constrained Yield Optimized Designs

( $T_{res} = 182^{\circ}\text{C}$ ,  $T_{wet\ bulb} = 26.7^{\circ}\text{C}$ , expander  
maximum stage pressure ratio = 1.8)

| <u>Parameter</u>                    | <u>Busbar Cost Optimization</u><br>(TPCTØ1Ø, 3/31/80) |                 | <u>Constrained Yield Optimization</u><br>(PADRA3Ø, 4/23/80) |                 |
|-------------------------------------|-------------------------------------------------------|-----------------|-------------------------------------------------------------|-----------------|
|                                     | <u>Input</u>                                          | <u>Computed</u> | <u>Input</u>                                                | <u>Computed</u> |
| $T_{TURB}$ ( $^{\circ}\text{K}$ )   | Optimizable                                           | 426.8279        | Optimizable                                                 | 425.7038        |
| $P_{TURB}$ (bar)                    | Optimizable                                           | 33.03275        | Optimizable                                                 | 33.35050        |
| $T_{CON}$ ( $^{\circ}\text{K}$ )    | Optimizable                                           | 310.791100      | Optimizable                                                 | 310.5490        |
| $1_{MCOMP}$                         | Optimizable                                           | 0.513135        | 0.513135                                                    | 0.513135        |
| $2_{DTMIN1}$ ( $^{\circ}\text{K}$ ) | Optimizable                                           | 5.313892        | 5.313892                                                    | 5.313892        |
| $3_{DTMIN2}$ ( $^{\circ}\text{K}$ ) | Optimizable                                           | 4.911490        | 4.911490                                                    | 4.911490        |
| $4_{CTEMP}$ ( $^{\circ}\text{K}$ )  | Optimizable                                           | 304.851500      | 304.85150                                                   | 304.85150       |

Proximity of selected optimum turbine inlet state to TPCT line:

|                                     |        |        |        |
|-------------------------------------|--------|--------|--------|
| $\frac{T_{TPCT}}{(T_{TURB})_{OPT}}$ | 0.9949 | 0.9991 | } Note |
| $\frac{P_{TPCT}}{(P_{TURB})_{OPT}}$ | 1.0307 | 1.0053 |        |

Notes:

- 1: Isobutane mole fraction
- 2: Primary heat exchanger pinch point temperature difference
- 3: Condenser pinch point temperature difference (zero subcooling)
- 4: Cooling tower water exit temperature - i.e.  $T_{wet\ bulb} + \Delta t$  approach

TABLE 7

Detailed Thermodynamic and Economic Performance  
Comparison for Busbar Cost and 3 Parameter Constrained  
Yield Optimized Geothermal Binary Rankine Cycle Systems

( $T_{res} = 182^{\circ}\text{C}$ ,  $T_{wet\ bulb} = 26.7^{\circ}\text{C}$ ,  
Expander stage pressure ratio = 1.8)

| <u>Parameter</u>                              | <u>Busbar Cost Optimized</u><br>(TPCTØ1Ø, 3/31/80) | <u>Constrained</u><br><u>Yield Optimized</u><br>(PADRA3Ø, 4/23/80) |
|-----------------------------------------------|----------------------------------------------------|--------------------------------------------------------------------|
| Independent Parameters                        | 7                                                  | 3                                                                  |
| Brine exit temperature ( $^{\circ}\text{K}$ ) | 340.482 (153.2 $^{\circ}\text{F}$ )                | 332.400 (138.65 $^{\circ}\text{F}$ )                               |
| Brine flow rate $\dot{m}_b$ (Kg/sec)          | 884.01                                             | 857.89                                                             |
| Hydrocarbon flow rate, $\dot{m}_h$ (Kg/sec)   | 992.29                                             | 1081.42                                                            |
| Cooling water flow rate $\dot{m}_w$ (Kg/sec)  | 5873.50                                            | 6828.47                                                            |
| Net plant efficiency (%)                      | 11.624                                             | 11.202                                                             |
| Wellhead Utilization Efficiency (%)           | 43.52                                              | 44.847                                                             |
| Overall Expander Efficiency (%)               | 84.998                                             | 83.501                                                             |
| Exhaust Quality (%)                           |                                                    |                                                                    |
| Stage 1                                       | 99.9                                               | 91.7                                                               |
| Stage 2                                       | dry                                                | 99.6                                                               |
| Stage 3                                       | dry                                                | dry                                                                |
| Stage 4                                       | dry                                                | dry                                                                |
| Relative Fuel Cost                            | 1.000 (@\$.570/10 <sup>6</sup> Btu)                | .9705 (@\$.533/<br>10 <sup>6</sup> Btu)                            |
| Relative Plant Capital Cost                   | 1.000                                              | 1.0573                                                             |
| Relative busbar energy cost                   | 1.000                                              | 1.0146                                                             |

Table 8

Thermodynamic and Economic Performance Difference  
Highlights for Busbar Cost and 4 Parameter Constrained  
Yield Optimized Geothermal Binary Rankine Cycle Systems  
( $T_{res} = 182^{\circ}\text{C}$ ,  $T_{wet\ bulb} = 26.7^{\circ}\text{C}$ ,  
Expander stage pressure ratio = 1.8)

| <u>Parameter</u>                | <u>Busbar Cost<br/>Optimization</u><br>(TPCT 010, 3/31/80) | <u>Constrained<br/>Yield Optimization</u><br>(PADRA45, 4/24/80) |
|---------------------------------|------------------------------------------------------------|-----------------------------------------------------------------|
| No. of Indep. Parameters        | 7                                                          | 4                                                               |
| Relative brine flow rate        | 1.000                                                      | 0.9351                                                          |
| " hydrocarbon flow rate         | 1.000                                                      | 1.0572                                                          |
| " cooling water flow rate       | 1.000                                                      | 1.7734                                                          |
| Isobutane mole fraction         | 0.5131                                                     | 0.777                                                           |
| Relative turbine inlet pressure | 1.000                                                      | 1.401                                                           |
| " net plant efficiency          | 1.000                                                      | 0.9983                                                          |
| " utilization efficiency        | 1.000                                                      | 1.0694                                                          |
| " expander efficiency           | 1.000                                                      | 1.000                                                           |
| " fuel cost                     | 1.000 (@\$0.570/10 <sup>6</sup> Btu)                       | 0.9351 (@\$0.532/10 <sup>6</sup> Btu)                           |
| " plant capital cost            | 1.000                                                      | 1.1833                                                          |
| " busbar energy cost            | 1.000                                                      | 1.0611                                                          |

} note

} note

Proximity of selected optimum turbine inlet state to TPCT line:

|                                                        |        |        |
|--------------------------------------------------------|--------|--------|
| $\frac{ T_{TPCT} - T_{TURB_{OPT}} }{(T_{TURB})_{OPT}}$ | 0.9949 | 1.0052 |
| $\frac{ P_{TPCT} - P_{TURB_{OPT}} }{(P_{TURB})_{OPT}}$ | 1.0307 | 0.9740 |

TABLE 9 POTENTIAL BINARY DEMONSTRATION PLANT PERFORMANCE IMPROVEMENTS

| $T_{RES.}$<br>°C<br>(°K) | COMPUTED OPTIMUM DESIGNS   |                    |                     |                    |          |          | SDGE PROPOSED DEMONSTRATION PLANT |                    |                     |                    |            |            | RELATIVE*<br>BUSBAR<br>COST |
|--------------------------|----------------------------|--------------------|---------------------|--------------------|----------|----------|-----------------------------------|--------------------|---------------------|--------------------|------------|------------|-----------------------------|
|                          | $iC_4$<br>mole<br>fraction | $T_{TURB}$<br>(°K) | $P_{TURB}$<br>(bar) | $T_{COND}$<br>(°K) | $\eta_c$ | $\eta_u$ | $iC_4$<br>mole<br>fraction        | $T_{TURB}$<br>(°K) | $P_{TURB}$<br>(bar) | $T_{COND}$<br>(°K) | $\eta_c^*$ | $\eta_u^*$ |                             |
| 170<br>(443.15)          | (a)<br>0.7925              | 428.431            | 40.054              | 310.114            | 11.905   | 44.528   | (c)<br>0.90                       | 424.817            | 39.645              | 313.650            | 11.255     | 39.708     | 0.9621                      |
| 182<br>(455.15)          | (b)<br>0.5137              | 439.690            | 38.213              | 307.298            | 12.291   | 44.544   | (d)<br>0.90                       | 424.817            | 39.645              | 313.650            | 10.959     | 39.709     | 0.9479                      |

ASSUMPTIONS:

- 1) IDENTICAL GENERIC SYSTEMS INCLUDING PRODUCTION AND INJECTION PUMPING (FIGURE 1)
- 2) ALL 50 MWe (net), 0.7 PLANT AVAILABILITY, EXPANDER EFFICIENCY = 85%, STARLING SPECIFIC MBWR.
- 3) BRINE INJECTION TEMPERATURE  $\geq 344.26$  K (160°F) (REF. 19)
- 4) PLANT SUB-SYSTEM CAPITAL COSTS NORMALIZED TO EPRI ER-301 (REF. 17)
- 5) UNIT BRINE COST UP FACTOR OF 5 (OVER REF. 17)
- 6) PRODUCTION & INJECTION ZONE DEPTH 1830 m (6000 ft),  $K_{DD} = 22.8$  KPa/(Kg/sec)
- 7) DOWNHOLE PRODUCTION PUMP ADIABATIC EFFICIENCY = 50%
- 8) LBL DESIGNS ASSUME  $U_c = 50$  BTU/hr ft<sup>2</sup>°F ( SDGE DESIGNS ASSUME  $U_c = 100$  BTU/hr ft<sup>2</sup>°F)
- 9) MINOR PARASITIC LOSSES IN PRODUCTION AND INJECTION PIPING IGNORED
- 10) SEE TABLE 1 FOR OTHER DETAILS

\* BY OUR ANALYSIS

DOCUMENTATION:

- |                       |                      |
|-----------------------|----------------------|
| (a) SF17000, 4/14/80  | (c) SIMS000, 4/14/80 |
| (b) SF182000, 4/14/80 | (d) SIMS002, 4/14/80 |

TABLE 10 COMPARISON OF SDGE PROPOSED PLANT WITH TPCT CYCLE ALTERNATIVE

| COMPARISON<br>CATEGORY                                | SDGE PROPOSED DEMONSTRATION<br>PLANT (REF. 19) |         | MID-LIFE WORKING FLUID<br>COMPOSITION ALTERNATIVE |         |
|-------------------------------------------------------|------------------------------------------------|---------|---------------------------------------------------|---------|
|                                                       | (182°C)                                        | (170°C) | (182°C)                                           | (170°C) |
| <u>BUSBAR ENERGY COST</u>                             | 1.000                                          | 1.000   | 0.948 *                                           | 0.962 * |
| <u>UTILIZATION EFFICIENCY (%)</u>                     | 39.71                                          | 39.71   | 44.54                                             | 44.53   |
| <u>NET CYCLE EFFICIENCY (%)</u>                       | 10.96                                          | 11.26   | 12.29                                             | 11.91   |
| <u>ISOBUTANE MOLE FRACTION</u>                        | 0.90                                           | 0.90    | ≈ 0.51                                            | ≈ 0.79  |
| <u>PRIMARY HEAT EXCHANGER<br/>RELATIVE AREA REQ'D</u> | 1.00                                           | 1.42    | 1.88                                              | 1.98    |
| <u>- CHANGE (%) -</u>                                 |                                                | +42.0 ‡ |                                                   | +5.5    |
| <u>GROSS TURBINE POWER</u>                            | 1.000                                          |         | 0.945                                             |         |
| <u>MAXIMUM INLET PRESSURE</u>                         | 1.000                                          |         | 1.010                                             |         |
| <u>TURBINE CHARACTERISTICS</u>                        |                                                |         |                                                   |         |
| • SPECIFIC SIZE CHANGE                                |                                                | +1.44 % |                                                   | +4.82 % |
| • SPECIFIC SPEED CHANGE                               |                                                | +1.44 % |                                                   | -0.83 % |

\* CONSERVATIVE ESTIMATE - LBL DESIGNS ASSUME 50% LOWER CONDENSER U FACTOR.

‡ BASED 10°F MINIMUM PRIMARY EXCHANGER PINCH POINT ΔT WHICH GIVES 170°F BRINE EXIT TEMPERATURE. WITH 160°F EXIT TEMPERATURE, CHANGE IS LARGER THAN 42 %.



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