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

Annual Report

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ABSTRACT

The research discussed in this report was performed at the University of Oklahoma during the period January 1, 1978 through December 31, 1978. Efforts were directed to the following tasks: (1) documentation of the GEO4 cycle simulator, (2) modification of GEO4 for fixed heat transfer area, (3) initial comparisons of mixture and pure fluid cascade cycles, (4) development of guidelines for working fluid selection for single boiler cycles, (5) continued evaluation of mixtures as working fluids, (6) specification of commercial isobutane composition requirements for isobutane cycles, (7) identification of working fluid thermophysical property data needs, (8) working fluid thermophysical property correlation and presentation of properties information, (9) effects of using different isobutane thermodynamic correlation parameters in single boiler cycle calculations. Some of the conclusions from this research are: (1) mixture dual boiler cascade cycles can be designed to yield approximately as much work per unit mass of brine as pure fluid triple boiler cascade cycles, indicating mixture cascade cycles are attractive when high brine utilization of low temperature georesources is desired, (2) the specifications by suppliers of a number of presently available commercial isobutanes will perform to yield net plant power for 300°F georesource cycles within one percent of the design value based on pure isobutane as the working fluid, (3) in binary cycles, mixtures have advantages over pure fluids which can be exploited through the use of available heat exchanger types in which essentially countercurrent flow can be maintained, (4) although thermodynamic property data recently obtained by the National Bureau of Standards has lessened data needs for isobutane, the use of other fluids (e.g., isopentane and mixtures) in major geothermal projects will create needs for new experimental work to remove data deficiencies for these fluids, (5) the levels of accuracy of presently available isobutane thermodynamic data and correlations leads to property predictions accurate enough for engineering design calculations for virtually all geothermal binary cycle operating conditions, with the exception of the critical region, which is avoided in most designs due to potential control problems.

TABLE OF CONTENTS

1. INTRODUCTION AND OVERALL SUMMARY

The majority of the work discussed in this report was performed at the University of Oklahoma during the period January 1, 1978 through December 31, 1978. Efforts were directed to the following tasks.

(1) Documentation of the GEO4 Cycle Simulator

(2) Modification of GEO4 for Fixed Heat Transfer Area

(3) Initial Comparisons of Mixture and Pure Fluid Cascade Cycles

(4) Development of Guidelines for Working Fluid Selection for Single Boiler Cycles

(5) Continued Evaluation of Mixtures as Working Fluids

(6) Specification of Commercial Isobutane Composition Requirements for Isobutane Cycles

(7) Identification of Working Fluid Thermophysical Property Data Needs

(8) Working Fluid Thermophysical Property Correlation and Presentation of Properties Information

(9) Effects of using Different Isobutane Thermodynamic Correlation Parameters in Single Boiler Cycle Calculations

A report on the documentation of the GEO4 simulator (task 1) has been prepared. The main contribution to the industry provided by the GEO4 simulator is the capability for considering mixture working fluids. The modification of the GEO4 simulator for fixed heat transfer area calculations (task 2), which corresponds closely to fixed plant simulation will be discussed in the Ph.D. thesis of K.Z. Iqbal. Fixed plant type calculations are of course useful for various calculations, such as changes in working fluids, effects of declining georesource temperature, etc. The initial comparisons of mixtures and pure fluids in cascade cycles (task 3) were summarized in a report to DOE-Idaho project personnel in October, 1978. This work indicates that for low temperature georesources such as Raft River, dual boiler mixture cycles may achieve resource utilization levels as great as triple boiler pure fluid cycles. This area of study is being further addressed in the M.S. thesis research of C.T. Chu. The subject of development of guidelines for working fluid selection for single boiler cycles (task 4) was addressed in a paper* presented at the Thirteenth Intersociety Energy Conversion Conference in San Diego in August, 1978. This work indicated there may be numerous fluids as good or better than isobutane in geothermal binary cycles. A total of 100 fluids now have been considered in the preliminary screening process. This work is being further addressed in the M.S. thesis research of S. Ramaswamy. The evaluation of mixtures as working fluids in geothermal binary cycles (task 5), has been a continuing study at the University of Oklahoma since 1973. This work has indicated that if countercurrent flow can be achieved in heat exchangers, mixtures have definite advantages over pure fluids in subcritical binary cycles. Discussion of thermodynamic, economic and equipment considerations will be presented by K.Z. Iqbal in his Ph.D. thesis. The specification of commercial isobutane composition (task 6) has focused on low temperature resource cycles. The conclusion is that a number of presently available commercial isobutanes will perform to yield net plant power for 300°F georesource cycles within one percent of the design value based on pure isobutane as the working fluid. Specifications are given for the maximum levels of the common impurities which can be tolerated in isobutane for satisfactory plant operation. Task 7 is the continuation of an effort initiated in 1976 to identify working fluid thermophysical property data needs. A cooperative effort among different geothermal research groups led to an identification of isobutane thermodynamic properties data needs which in turn led to an experimental data procurement program at the National Bureau of Standards, in 1978. Our recent efforts have been directed to identifying isopentane thermodynamic data needs and identifying viscosity and thermal conductivity data needs for isobutane and isopentane. Also, thermodynamic property data needs were considered for the isobutane-isopentane mixture originally considered for the proposed Heber plant. Task 8 has involved the determination of correlation parameters for the MBWR equation of state for isobutane and isopentane and parameters in correlations of viscosity and thermal conductivity for isobutane and viscosity for isopentane. Because thermal conductivity data for isopentane in the region of cycle operating conditions are not available, a generalized correlation was used to provide a reasonable means for estimating isopentane thermal conductivity. To provide the geothermal industry with information of immediate use, the data judged to be the best available have been correlated for isobutane and isopentane and a report has been prepared which will be published in 1979. Task 9, a study of the effects of using different isobutane thermodynamic correlation parameters in single boiler cycle calculations was performed because it was found that MBWR (modified Benedict-Webb-Rubin) equation parameters determined for isobutane in 1973 (and used in cycle calculations until 1977) were not as accurate as desired. The availability of accurate vapor pressure and density data recently obtained by the National Bureau of Standards allowed the determination of more accurate MBWR parameters in 1978. Cycle calculations using the 1978 MBWR parameters were then used to determine the regions, magnitudes and causes of uncertainties in calculations using earlier reported modified BWR parameters as well as original BWR equation parameters. The largest errors in the earlier equations occured for cycles operating near the critical region. For conditions well removed from the critical region, the agreement in cycle calcuations was reasonably good and accuracy was sufficient for most engineering design calculations.

2. PREPARATION OF SUMMARY DOCUMENTATION FOR THE CONVENTIONAL BINARY CYCLE SIMULATOR GEO4

A report has been prepared presenting summary documentation for the GEO4 simulator developed at the University of Oklahoma (1). This description of the GEO4 program is presented along the lines previously used for the thermodynamic properties program HSGC (2).

The GEO4 computer program is a design simulation of a single boiler geothermal binary power plant using shell-and-tube heat exchangers. Basically, the computer simulation is an ordered set of calculations which describes the changes in the physical state of the working fluid as it completes the power cycle in order to simulate the design of a geothermal power plant. The documentation consists of a separate section for each subroutine and the main program segment. Each section contains that subroutine's hierarchy, functional description, variable description, algorithm, and step-by-step procedure. The hierarchy lists which subroutines call a particular subroutine and in turn, which subroutines are called. The functional description section gives a brief description of the purpose of the subroutine. The variable description consists of the classification of all variables in each subroutine as one of three types; input variables are those variables which are carried through the common block or block data to the subroutine from the calling subroutine, intermediate variables are those variables which are used only in internal calculations for a particular subroutine, output variables are either returned to the calling subroutine or are printed on paper. The algorithm section consists of two parts; the basis, and a step-by-step procedure. In the basis, all the necessary background information and equations used in the subroutine are discussed. The step-by-step procedure is a synopsis of the major steps taken in each subroutine. A computer generated flow chart of each subroutine, the GEO4 listing and examples of input and output also are presented. To make the report useable as a stand-alone source for making computations, parameters for the modified Benedict-Webb-Rubin equation of state for hydrocarbons, halocarbons, nitrogen, carbon dioxide, hydrogen sulfide, ammonia and water are presented, along with coefficients for the ideal gas heat capacity, enthalpy and entropy polynomials used.

The GEO4 program in its present form can be used for simplified multi-boiler binary cycle simulation as well. In addition, the GEO4 program can be modified to fix the heat transfer surface areas of the brine heat exchanger and condenser in order to make a fixed plant type calculation. The procedure to handle these calculations will be presented in K.Z. Iqbal's Ph.D. thesis. The GEO4 program also can be modified to perform staged flash binary cycle calculations and direct contact brine heat exchanger calculations using the Letan-Kehat method for Elgin type direct contact heat exchangers (3).

Copies of the GEO4 program are on magnetic tape at the University of Oklahoma Computing Center and can be obtained from the Department of Chemical Engineering, 202 West Boyd Street, Norman, Oklahoma 73019.

3. MODIFICATION OF THE GEO4 SIMULATOR FOR FIXED HEAT TRANSFER AREA CALCULATIONS

The motive behind the modification of the GEO4 simulator was to simulate fixed plant operation as closely as possible without modifying the GEO4 design simulator extensively. It was relatively easy to fix the sizes of the heat exchangers while leaving the routines for the remaining equipment units unmodified. The result is not a true fixed plant simulator, but rather a fixed heat transfer area simulator. The essential method by which the fixed heat transfer area calculation is performed is summarized in Appendix I; the details will be given in K.Z. Iqbal's Ph.D. thesis.

The use of the GEO4 simulator in the fixed heat transfer area mode is useful for approximating fixed plant calculations. For example, the effects on an existing power plant of declining (or increasing) resource temperature can be studied. Similarly, the effects of changing the brine flow rate can be studied. Also, the effects of changing the working fluid composition can be studied (so long as changes in the turbine and other equipment sizes are not too great). Thus, the GEO4 simulator in the fixed heat transfer area mode can be used for a number of fixed plant type calculations.

4. INITIAL COMPARISONS OF MIXTURE AND PURE FLUID CASCADE CYCLES

The objective of this initial work was to determine if mixtures can be attractive as working fluids in cascade binary cycles. This initial work was summarized in a report to DOE-Idaho project personnel in October, 1978. Dual boiler cycle calculations were performed for a 290°F georesource (Raft River brine) using as working fluids pure isobutane, pure cis-2-butene, three mixtures of propane and cis-2-butene and a mixture of cis-2-butene and cyclopentane. It was found that the use of one of the mixtures of propane and cis-2-butene yielded approximately 10% more work per unit mass of brine than the isobutane dual boiler cycle. Because the mixture cycle would operate at higher pressures than the isobutane cycle, the capital cost (in $/kw) for the mixture cycle would be less than 10% smaller than the pure fluid cycle, especially in the case of direct contact cycles. However, this initial work does demonstrate the potential for the use of mixtures in cascade cycles. An interesting result is the fact that a dual boiler mixture cycle can yield as much work per unit mass of brine as a triple boiler pure fluid cycle. Therefore, if a mixture cycle with operating pressures near the operating pressures for isobutane cycles can be utilized, it is possible that the mixture dual boiler cycle will be more attractive than the pure fluid dual or triple boiler cycles. The M.S. thesis research of C.T. Chu, which addresses the subject of the use of mixtures in cascade cycles, should help to clarify this issue.

5. DEVELOPMENT OF GUIDELINES FOR SELECTION OF WORKING FLUIDS AND OPERATING CONDITIONS FOR GEOTHERMAL BINARY CYCLES

This effort has involved development of a preliminary screening process which eliminates fluids from consideration as candidate working fluids based on unacceptably high brine heat exchanger pressure, unacceptably low subatmospheric condenser pressure and unacceptably large amount of superheat at either the turbine inlet or exit. The fluids passing this preliminary screening test then can be compared in detailed ways, including cycle simulation. Using this method, it was determined that cis-2-butene is potentially a better working fluid than isobutane (the choice for low georesource temperatures depends on the relative costs of the brine delivery system and power plant). The preliminary screening process and calculations comparing cis-2-butene as a working fluid with isobutane and isobutane-isopentane mixtures was presented at the 13th Intersociety Energy Conversion Conference in San Diego in August, 1978. This paper (4) is given in Appendix II. During the latter part of 1978, additional fluids were considered, bringing the total number of fluids considered in the preliminary screening process to 100. Appendix III presents a table of the pertinent information used in the preliminary screening process. For some of the 100 fluids in Appendix III, available thermodynamic data are not sufficient to complete the preliminary screening. For 21 of the fluids which passed the preliminary screening process, single boiler cycle simulation calculations have been performed for the case of a 300°F georesource temperature. We are in the process of analyzing these results in an attempt to determine if a method for working fluid selection beyond the preliminary screening procedure can be developed. The M.S. thesis of S. Ramaswamy will address the subject of working fluid selection for single boiler geothermal binary cycles for the range of georesource temperatures from 300°F to 500°F.

6. DEFINITION OF THE ADVANTAGES AND DISADVANTAGES OF MIXTURES AS WORKING FLUIDS IN GEOTHERMAL BINARY CYCLES

A considerable amount of work has been carried out at the University of Oklahoma since 1973 on the subject of mixture working fluid geothermal binary cycles. Nevertheless, it was felt that a clear definition of the advantages and disadvantages of mixtures compared to pure fluids was needed. In addition, it was felt that hardware (equipment) needs peculiar to mixtures should be discussed. These topics will be addressed by K.Z. Iqbal in his Ph.D. thesis through a discussion of thermodynamic, economic and equipment considerations. In the present report, the advantages and disadvantages of mixtures compared to pure fluids are summarized in Appendix IV and hardware needs are summarized in Appendix V. It has been concluded that if countercurrent flow can be achieved in heat exchangers, mixtures have definite advantages over pure fluids in subcritical single boiler and cascade binary cycles. For shell-and-tube heat exchangers, the problem of unequal vapor and liquid flow velocities can be virtually eliminated by putting the working fluid on the tube side, which is possible if the brine does not foul or if fluidized bed heat exchangers are used. Essentially countercurrent flow can be achieved by proper configuration of shell-and-tube heat exchangers, by the use of vertical spiral wound heat exchangers such as are used in the liquefied natural gas industry or by the use of direct contact countercurrent flow heat exchangers. Thus, there are numberous equipment options which meet the needs of mixture cycles.

7. SPECIFICATION OF PERMISSIBLE COMMERCIAL ISOBUTANE COMPOSITION RANGE FOR ACCEPTABLE OPERATION OF GEOTHERMAL BINARY CYCLES

It was determined in this study (see Appendix VI) that specifications of commercial isobutane purity by a number of suppliers are sufficient for acceptable operation of low temperature geothermal binary cycles designed considering pure isobutane to be the working fluid. Fixed plant simulation calculations were performed for a single boiler geothermal binary cycle utilizing 275°F brine. Calculations were performed for pure isobutane, binary mixtures of isobutane with normal butane, propane, ethane and normal pentane and ternary mixtures of isobutane, normal butane and propane corresponding to available commercial isobutanes. The GMBWR (generalized modified Benedict-Webb-Rubin) equation of state was used for calculation of working fluid thermodynamic properties. It was found that less than a 1% loss in net power would be expected for commercial isobutanes meeting the following specification of composition, in mole percentages; minimum 95% isobutane, maximum 5% normal butane, maximum 3% propane, maximum 0.1% ethane, maximum 0.1% pentanes. A number of suppliers of commercial isobutane have specifications which are within the above limits. Thus, for working fluid operating conditions well removed form the critical region, plants designed for pure isobutane should suffer less than 1% reduction in net plant power with the use of commercial isobutane meeting these specifications. The details of this study are given in Appendix VI.

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

This task has involved searching the literature for data, evaluation of the available data and recommendation of the data procurement and correlation efforts needed to meet geothermal industry needs. This work has been carried out as a cooperative effort with other centers for geothermal research, principally Brown University, the Idaho National Engineering Laboratory and other Department of Energy offices and laboratories, the National Bureau of Standards and the Electric Power Research Institute. The working fluids considered to be prime candidates for geothermal binary cycles in 1978 were emphasized in this effort. The data considered were themodynamic property data, viscosity data and thermal conductivity data. Two fluids were considered in some detail, isobutane and isopentane. Isobutane is the working fluid for the first 5MW Raft River power plant, while isopentane is a prime candidate working fluid for the second Raft River 5MW power plant, which will use direct contact heat exchange. Some work also was done to evaluate thermodynamic property data needs for an isobutane-isopentane mixture considered as the working fluid in design studies for the 50MW Heber power plant. However, the effort for this mixture was stopped when plans for the Heber plant became uncertain early in 1978.

The effort to identify data needs for these fluids will be presented, along with correlations of the data, in a forthcoming report.

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

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

These correlations of the thermophysical properties of isobutane and isopentane will be presented in a forthcoming report.

10. EFFECTS OF USING DIFFERENT ISOBUTANE THERMODYNAMIC CORRELATION PARAMETERS IN SINGLE BOILER CYCLE CALCULATIONS

This work, which is discussed in detail in Appendix VII, demonstrates that the OBWR (original Benedict-Webb-Rubin) equation and the MBWR (modified Benedict-Webb-Rubin) with various parameter sets yield geothermal cycle simulations in reasonable agreement for operating conditions excluding the critical region. For operating conditions in the critical region, only an extremely accurate correlation can be used for design purposes. Because of the inherent problem of control of power plants, the critical region is avoided by most designers. Therefore, the more accurate available correlations of isobutane thermodynamic behavior should be adequate for most design situations. For a 300°F georesource, the OBWR equation and MBWR equation with 1973, 1977 or 1978 parameters yield essentially equivalent design simulations. For a 350°F resource cycle the predicted bubble point enthalpy in the brine heat exchanger is erroneously high using the 1973 MBWR parameters. For the 400°F georesource, there is again good agreement between the different correlations. The conclusion is that although previous design calculations using the 1973 MBWR parameters were adequate for conditions far removed from the critical point, the more recent MBWR parameters (1977, 1978) are recommended.

11. REFERENCES

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APPENDIX I

MODIFICATION OF THE GEO4 SIMULATOR FOR FIXED HEAT TRANSFER SURFACE AREA CALCULATIONS

Modification of the GEO4 Simulator for Fixed Heat Transfer Surface Area Calculations

The original motive behind this modification of the GEO4 simulator was to simulate as closely as possible fixed plant operation without extensive modification of GEO4. To accomplish this objective, the following parameters for the brine heat exchanger and condenser are specified as input:

(1) Heat transfer surface areas

(2) Shell inside diameter

(3) Number of tubes

For the remaining major equipment units (turbine, pumps, cooling tower) equipment sizes are not fixed, but are determined as if the simulator were in a design mode rather than a fixed plant mode. This does not seriously limit the usefulness of the program in the fixed plant mode for studies of the effects of changes in georesource temperature, working fluid composition and other parameters, when the changes are not too large.

The fixed heat transfer surface area simulation is controlled by the optimization control parameter "NOPT" and the areas. The main features of this modification are as follows:

(1) When NOPT=0 and the areas are input as nonzero, the brine flow rate as specified in input remains unchanged. The approach temperature differences and the fluid pressure drops are then updated iteratively as follows:

$$\mathrm{DTHWO}_{i+1} = \frac{A^{\mathrm{calc}}_{\mathrm{BHE}}}{A^{\mathrm{spec}}_{\mathrm{BHE}}} \mathrm{DTHWO}_{i} \qquad (1)$$

$$\mathrm{DTCWO}_{i+1} = \frac{A^{\mathrm{calc}}_{\mathrm{cond}}}{A^{\mathrm{spec}}_{\mathrm{cond}}} \mathrm{DTCWO}_{i} \qquad (2)$$

and

$$\mathrm{DTCWI}_{i+1} = \frac{A^{calc}_{cond}}{A^{spec}_{cond}}\,\mathrm{DTCWI}_{i} \qquad (3)$$

No fluid pressure drop updating is needed if

$$\left|\frac{DP_{input} - DP_{calc.}}{DP_{calc}}\right| \leq \mathrm{DPSDPW} \qquad (4)$$

The above subscripts i and i+1 refer, respectively, to the iteration numbers i and i+1 and

A^{calc}_{BHE} = Calculated brine heat exchanger area, ft^2

A^{spec}_{BHE} = Input specified value of BHE area, ft^2

A^{calc}_{cond} = Calculated condenser area, ft^2

A^{spec}_{cond} = Input specified value of condenser area, ft^2

DP_{input} = Fluid pressure drops in input, psia

DP_{calc} = Calculated fluid pressure drops, psia

EPSDPW = Convergence criterion for fluid pressure drops

DTHWO
DTCWO = Approach temperature differences in BHE and condenser
DTCWI

(2) When NOPT=-1 and the areas are input as nonzero, the brine flow rate, approach temperature differences, and fluid pressure drops are all updated in order to match the specified net power.

The details of these modifications will be provided elsewhere(1). Option 1 has been successfully tested for some working fluids. Option 2 in its present form takes an excessive amount of time for convergence and has not been used.

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APPENDIX II

HYDROCARBON WORKING FLUID AND OPERATING CONDITIONS SELECTION FOR THE CONVENTIONAL GEOTHERMAL BINARY CYCLE

HYDROCARBON WORKING FLUID AND OPERATING CONDITIONS SELECTION FOR THE CONVENTIONAL GEOTHERMAL BINARY CYCLE

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ABSTRACT

The subject of selecting working fluid and process operating conditions for the geothermal binary power cycle is addressed herein. To examine a wide variety of potential working fluids, a fluid screening procedure was developed. The first phase of the three phase screening procedure permits a quick evaluation, with potential working fluids identified for more detailed analysis. The second phase screening procedure requires more thermophysical property data for the potential working fluid in order to evaluate a series of cycle parameters. The final phase of the screening procedure requires evaluation with the aid of a power cycle simulation program complete with rigorous thermodynamics, process, and economic estimation routines. The working fluids compared in detail herein include isobutane, isopentane, cis-2-butene, R-11, R-114 and an equimolar mixture of isobutane and isopentane. It is concluded that the range of working fluids considered by the industry to date has been too limited.

ALTHOUGH THE SUBJECT OF GEOTHERMAL BINARY CYCLE WORKING FLUID SELECTION has been addressed in a number of literature sources (1-4)*, detailed cycle simulation calculations including cost estimation have been reported for only a few fluids. Herein the results of detailed simulation calculations are summarized for a number of working fluids, including isobutane, isopentane, an isobutane-isopentane mixture, cis-2-butene and the halogenated hydrocarbons, R-11 and R-114. From this study, it is concluded that the range of working fluids considered by the industry to date has been too limited and a detailed new review of working fluids should be performed.

THE IDEAL WORKING FLUID

If an ideal working fluid for geothermal binary cycles existed, it could be characterized reasonably well by the following requirements.

(1) The working fluid dew point pressure at the point of initial condensation (say 43°C) in the condenser should be as low as possible (to minimize condenser cost per unit of heat transfer surface area, which is an increasing function of pressure) but greater than atmospheric pressure (to avoid leakage of air into the system).

(2) The working fluid saturated vapor locus on a temperature-entropy diagram should be nearly vertical, to avoid excessive superheat in heat exchangers and condensation at the turbine exit. Superheat is undesirable because heat transfer coefficients in the superheat region are smaller than in the boiling and condensing regions. Condensation in the turbine is undesirable for axial flow turbines because of the loss of approximately one percent in stage efficiency for each percent of liquid formed (manufacturers claim negligible loss due to condensation in radial inflow turbines). Condensation in the turbine is also undesirable because of the potential maintenance problems created by liquid impingement erosion.

(3) The working fluid specific enthalpy change in the turbine should be large, to maximize cycle thermodynamic efficiency and minimize working fluid flow rate.

(4) The turbine inlet pressure should be reasonably low, to minimize brine heat exchanger cost per unit of heat transfer surface area, which is an increasing function of pressure.

(5) The working fluid specific volume at the turbine exit should be small, to keep turbine size small (the turbine diameter is proportional to the square root of the exhaust volumetric flow rate).

(6) Working fluid liquid and vapor thermal conductivity coefficients should be large, to minimize heat transfer surface area. Correlations of convective heat transfer coefficients show that high values of thermal conductivity and low values of viscosity are desired.

(7) Working fluid liquid and vapor viscosities should be low, to minimize frictional pressure drops and maximize convective heat transfer coefficients.

(8) The working fluid should be stable, essentially nonfouling, noncorrosive, nontoxic and nonflammable. At the very least, underwriter approved safe handling methods must exist.

PRELIMINARY SCREENING OF REAL WORKING FLUIDS

No working fluid exists which simultaneously meets all of the requirements of the ideal working fluid discussed above. Real fluids must be screened to select primary candidate working fluids on a relative basis. A major difficulty in comparing different working fluids without resorting to cost optimization calculations using a simulator is choosing a calculation basis. An effort is in progress at the University of Oklahoma to develop a three-phase methodology for working fluid selection which will be reasonably accurate without the use of complex computer calculations. In this three-phase methodology, a preliminary screening of working fluids is performed in Phase I, candidate working fluids are compared using simplified thermodynamic, equipment sizing and economic calculations in Phase II and detailed cycle simulation calculations to determine optimal operating conditions for minimum energy cost are performed for primary candidate working fluids in Phase III. For present purposes, only Phase I and the initial steps of Phase II in the selection procedure will be discussed, followed by presentation of summaries of the results of detailed Phase III simulation calculations for a few selected working fluids.

To initiate the screening procedure, the characterization parameters such as the molecular weight and critical constants as well as certain thermodynamic data are gathered and tabulated for potential candidate working fluids, as shown for a few representative fluids in Table 1. In Table 1, the I-factor is the following quantity defined by Kihara and Fukunaga (1).

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

For a fluid with a vertical saturated vapor locus on a temperature-entropy diagram, the I-factor is unity. If I<1, the turbine exhaust will be superheated, while for I>1, the turbine exhaust may be wet (depending on the turbine inlet condition).

In the preliminary screening process (Phase I) as shown in Figure 1, candidate working fluids are rejected if (1) the fluid vapor pressure at 38°C is outside of the range 35 kpa to 1379 kpa or (2) the fluid I-factor (at 38°C) is outside of the range 0.50 to 2.00. The remaining fluids then can be ranked on the basis of simplified thermodynamic, equipment sizing and capital cost calculations (Phase II). Simulation calculations then can be performed to distinguish between the primary candidate working fluids (Phase III). Herein, simplified thermodynamic

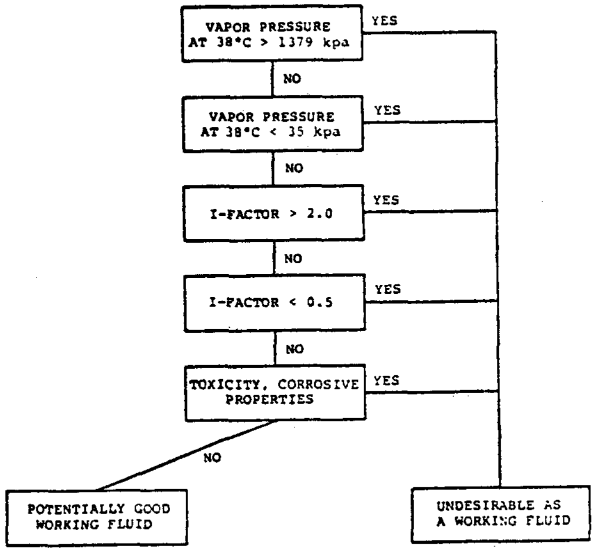

Fig. 1 - Phase one working fluid screening process

Table 1 - Rankine Cycle Working Fluid Thermodynamic Characterization Parameters

Name	Formula	Molecular Weight	Critical Temp., °C	Critical Pressure kpa	Vapor Press., kpa		Spec. Vol., m^3/kg @ 38°C		I-factor
					38°C	93°C	Liquid	Vapor	
Propane	C_3H_8	44.097	96.83	4246	1298.	3985	0.002116	0.0349	1.39
Isobutane	C_4H_{10}	58.124	135.0	3648	496.	1747	0.00186	0.0788	0.81
Isopentane	C_5H_{12}	75.151	187.3	3381	141.	622	0.00166	0.2402	0.77
Normal Hexane	C_6H_{14}	86.178	234.3	3012	34.	207	0.001548	0.8573	0.73
cis-2-Butene	C_4H_8	56.108	162.4	4206	318.	1241	0.001685	0.1326	1.00
Propadiene	C_3H_4	40.065	120.6	5473	1207.	3599	0.001717	0.05263	1.55
1,3 Butadiene	C_4H_6	54.092	151.7	4328	411.	1544	0.001686	0.1049	1.00
Cyclopentane	C_5H_{10}	70.135	237.8	4510	69.	3530	0.001373	0.5173	1.02
Propionaldehyde	C_3H_6O	56.080	222.2	4763	69.	*	*	*	1.2
R-11	$C\ Cl_3F$	137.368	198.9	4408	162.	727	0.000693	0.1102	1.10
R-114	$C_2Cl_2F_4$	170.922	146.1	3304	316.	1230	0.000705	0.0434	0.65
R-142 B	$C_2H_3Cl\ F_2$	100.496	137.2	4125	493.	1864	0.00930	0.0452	1.24
Ammonia	NH_3	17.031	132.4	11282	1466.	5545	0.001716	0.0886	6.89
Water	H_2O	18.015	374.2	22120	6.2	79	0.00100	21.862	3.40

*Data not available.

cycle calculations (Phase II) and detailed simulation calculations (Phase III) are discussed for isobutane, isopentane, cis-2-butene, R-11 and R-114 (the simplified methods for equipment size and capital cost estimation in Phase II are still in development). To keep the presentation concise, only preliminary screening calculations corresponding to a 149°C (300°F) georesource temperature are presented.

The method used here for the preliminary thermodynamic comparison in Phase II is to perform ideal Rankine cycle calculations with fixed boiling and condensing temperatures of 93°C (200°F) and 38°C (100°F), respectively. The definition of the ideal Rankine cycle as used here is a Rankine cycle in which working fluid expansions and compressions are reversible and there are no viscous pressure drops. Thus, for subcritical cycles, the turbine inlet and exit pressures are the vapor pressures at the boiling and condensing temperatures, respectively (at 93°C and 38°C in the illustration).

The turbine inlet condition is first determined. The procedure depends on whether the I-factor is less than or greater than unity and if $I<1$, whether the cycle is subcritical or supercritical. If $I<1$ and the cycle is subcritical, the turbine inlet condition is chosen to be saturated vapor. If $I<1$ and the cycle is supercritical, the inlet entropy is fixed at the maximum entropy along the saturated vapor locus (to avoid condensation in the turbine). If $I>1$, the turbine exit condition is chosen to be saturated vapor, thereby fixing the entropy of the working fluid at both the turbine exit and inlet (entropy at exit equals entropy at inlet for isentropic expansion). The specific turbine work $\underline{W}_T$ then can be calculated as the working fluid enthalpy change in the turbine. The working fluid pump inlet is taken to be saturated liquid at 38°C and the specific pump work, $\underline{W}_P$, is calculated according to the relation

$$\underline{W}_P = v_f \Delta P \qquad (2)$$

where v_f is the liquid specific volume and ΔP is the pressure difference between the pump outlet and inlet (note $\underline{W}_P<0$). The net cycle specific work, $\underline{W}_N$, is then

$$\underline{W}_N = \underline{W}_T + \underline{W}_P \qquad (3)$$

The specific boiler (brine heat exchanger) duty, $\underline{Q}_H$, is then the enthalpy difference between the turbine inlet and pump outlet. The fraction of the boiler duty in the boiling range (versus subcooled or superheated range) is the enthalpy of vaporization h_{fg} divided by $\underline{Q}_H$. The ratio $h_{fg}/\underline{Q}_H$ is used to determine the minimum brine exit temperature, $(T_{HO})_{min}$, by considering a heat exchanger pinch point at the working fluid bubble point,

$$(T_{HO})_{min} = T_{HI} - (T_{HI} - T_{WB})\, \underline{Q}_H/h_{fg} \qquad (4)$$

where T_{HI} is the brine inlet temperature (149°C herein), and T_{WB} is the working fluid bubble point temperature (93°C herein). The cycle thermodynamic efficiency, τ_N, is calculated from the relation

$$\tau_N = \underline{W}_N/\underline{Q}_H \qquad (5)$$

The working fluid required per unit of cycle work, M_W/W_N, then is merely the reciprocal of $\underline{W}_N$. The turbine exhaust volume per unit of work, V_T/W_N is approximated by

$$V_T/W_N = v_g/\underline{W}_N \qquad (6)$$

where v_g is the saturated vapor specific volume at 38°C. Note that V_T/W_N is a reasonable measure of relative turbine size. The brine required per unit of cycle work, M_H/W_N, is calculated from

$$M_H/W_N = (M_H/M_W)(M_W/W_N) \qquad (7)$$

where the brine to working fluid mass ratio is calculated from

$$\frac{M_H}{M_W} = \frac{\underline{Q}_H}{C_{pH}(T_{HI} - T_{HO})} \qquad (8)$$

where C_{pH} is the brine heat capacity (approximated to be 1.0 cal/gm °F herein).

The fluids chosen for illustrating the preliminary screening calculations are isobutane, isopentane, cis-2-butene, R-11 and R-114. All of these fluids except cis-2-butene have been considered previously as working fluids by various investigators, although not simultaneously in a single study. The calculations for cis-2-butene presented herein are given as an illustration of the fact that there remain numerous fluids which potentially can match or exceed the performance of the working fluids considered to date. It can be noted from Table 1 that among the five fluids considered, only cis-2-butene has an essentially vertical saturated vapor entropy locus versus temperature ($I=1.00$); thus, only for cis-2-butene will there be no superheat at either the turbine inlet or exit. Further, it can be noted that the condenser pressure of 318 kpa for the cis-2-butene cycle is very reasonable. With reference to Table 2, it can be noted that cis-2-butene has the largest specific enthalpy drop in the turbine, 60.7 MJ/kg, and net cycle work, 59.1 MJ/kg. Therefore, there is a low working fluid circulation requirement, 0.017 kg per MJ of net cycle work. The turbine for the cis-2-butene cycle is smaller than for all cycles except the isobutane cycle. Although the minimum brine exit temperature is second largest for the cis-2-butene cycle, cost optimized brine exit temperatures usually are considerably greater than the minimum when capital cost is taken into account in detail. Therefore, on the basis of this preliminary screening, cis-2-butene seems a worthy candidate for comparison with isobutane, isopentane, R-11 and R-114 on a more detailed basis.

COMPARISONS BASED ON SIMULATION CALCULATIONS

Simulation calculations were performed to compare the five pure working fluids on a detailed basis, taking into account not only thermodynamics but also equipment considerations and capital cost. Calculations also were performed for a mixture working fluid (equimolar mixture of isobutane and isopentane) to provide perspective regarding mixtures as working fluids.

A direct sequential search optimization routine was used in the process simulation program developed

Table 2 - Summary of Ideal Rankine Cycle Preliminary Screening Calculations

	Isobutane	**Isopentane**	**cis-2-Butene**	**R-11**	**R-114**
Spec. Turbine Work, $\underline{W}_T$, MJ/kg	49.5	56.1	60.7	27.4	19.3
Spec. Pump Work, $\underline{W}_P$, MJ/kg	2.3	0.7	1.6	0.5	0.7
Net Work, $\underline{W}_N$, MJ/kg	47.2	55.3	59.1	27.0	18.6
Spec. Boiler Duty, $\underline{Q}_H$, MJ/kg	387.5	422.9	437.7	200.0	154.7
h_{fg}, MJ/kg	231.0	284.5	307.0	150.7	93.3
$h_{fg}/\underline{Q}_H$	0.60	0.67	0.70	0.75	0.60
Thermodynamic Eff., τ_N	12.2	13.1	13.5	13.5	12.0
Working Fluid Req'd., M_W/W_N, kg/MJ	0.021	0.018	0.017	0.037	0.054
Turbine Exhaust Vol., V_T/W_N, cu.m/MJ	0.00166	0.00435	0.00223	0.00408	0.00233
Min. Brine Exit Temp., °C	56.28	65.94	69.50	75.78	56.28
Min. Brine Req'd., M_H/W_N, kg/MJ	0.021	0.022	0.022	0.024	0.021

at the University of Oklahoma to determine cycle operating conditions. The cycle simulator and the design basis have been discussed elsewhere (5,6) so that only a brief summary is presented here. The objective function for minimization was chosen to be capital cost in dollars per kilowatt for a power generation cycle producing 25 MW net electrical output. The capital cost was calculated using the factored estimate method with specific cost formulas for major equipment and specific factors for other costs. Pure water thermodynamic and transport properties were utilized for the brine model. The material of construction used in estimating the cost of the major heat transfer equipment (counter current flow, shell-and-tube) was carbon steel. The calculations consider the use of axial flow turbines and a wet cooling system. The calculations are based on 26.7°C (80°F) cooling water entering the condenser. Both power plant and well system costs were estimated to obtain the total system cost (in 1976 dollars). The well system cost is based on a well cost of $500,000 for a well flowing at the rate of 63 kg/sec (500,000 lb/hr) and an indirect cost factor of 1.56.

The results of the simulation calculations are summarized in Tables 3, 4 and 5 for the georesource temperatures 260°C (500°F), 204°C (400°F) and 149°C (300°F). The quantities given in Tables 3, 4 and 5 are generally self explanatory. The plant parasitic power requirement is the difference between the gross (turbine) power and the net power. The net work/availability ratio is the ratio of the net plant work per pound of brine to the maximum theoretical work a pound of brine could yield in a change of state between the brine inlet condition and the cooling water inlet condition (26.7°C).

Tables 3-5 indicate that cis-2-butene is the best choice for the geothermal binary cycle working fluid in the georesource range from 149°C to 260°C, although cis-2-butene is only marginally better than iospentane for the higher temperature georesource. Cis-2-butene is a readily available chemical, priced quite favorably at about 8 cents per pound, and is capable of remaining chemically stable in the power cycle operating environment. It should be noted that the halogenated hydrocarbons, R-11 and R-114, have recommended maximum operating temperatures of about 107°C (225°F) and, therefore are not considered for the higher georesource temperatures.

Table 3 - Comparison of Cycle Parameters for the Georesource Temperature of 260°C (500°F)

Fluid / Process Parameters	Iso-Butane	Mixture iC_4H_{10}=50% iC_5H_{12}=50%	Iso-Pentane	Cis-2-Butene
Cycle				
Net Power, MW	25.00	25.00	25.00	25.00
Gross Power, MW	32.72	29.30	27.98	29.27
Net Plant Work, MJ/kg brine	76.83	83.85	82.74	93.46
Net Thermo. Efficiency, %	12.27	13.23	12.68	14.78
Net Work/Availability	0.2803	0.3059	0.3018	0.3410
Power Plant Cost, $/kw	520	457	411	409
Total System Cost, $/kw	720	641	597	574
Brine Delivery				
Brine Flow, kg/kw	46.85	42.95	43.50	38.51
Brine Exit T, °C	108.8	109.1	105.8	109.9
Brine Delivery Cost, $/kw	201	184	186	165
Brine Heat Exchanger (BHE)				
BHE LMTD, °C	65.3	62.8	53.8	57.2
BHE load, ΔH, MJ/kg W.F.	363.3	447.3	449.6	431.0
BHE Cost, $/kw	110	91	90	60
Condenser				
Condenser Superheat ΔT, °C	11.0	24.9	37.3	2.4
Cond. Superheat ΔH, MJ/kg	22.63	49.47	72.87	3.98
Cond. Dew Point P, kpa	733.6	402.6	253.0	466.1
Cond. Dew Point T, °C	54.6	58.9	57.8	51.7
Condenser LMTD, °C	21.0	21.2	24.8	17.0
Cond. Load, ΔH, MJ/kg	318.7	387.9	392.6	367.3
Condenser Cost, $/kw	176	145	89	109
Cooling System				
Cooling Water Flow, kg/kw	493.9	415.4	375.0	376.5
Cooling Cost, $/kw	36	30	27	28
Turbine				
Turbine Inlet P, kpa	4826	3103	1724	3792
Turbine Inlet T, °C	156.7	157.2	145.5	156.1
Turbine ΔH, MJ/kg	54.42	65.13	60.43	70.66
Working Fluid Flow, kg/kw	86.6	64.8	66.7	59.6
W.F. Flow at Turbine Exit T, Cu. cm/sec/kw	1347	1841	2770	1537
Turbine-Generator Cost, $/kw	127	150	176	138
Cycle Pump				
Parasitic Power, MW	5.928	2.69	1.571	2.89
Pump Cost, $/kw	56	28	18	30

Table 4 - Comparison of Cycle Parameters for the Georesource Temperature of 204°C (400°F)

Process Parameters \ Fluid	Iso-Butane	Mixture iC_4H_{10}=50% iC_5H_{12}=50%	Iso-Pentane	Cis-2-Butene	R-11	R-114
Cycle						
Net Power, MW	25.00	25.00	25.00	25.00	25.00	25.00
Gross Power, MW	31.45	29.05	28.18	28.92	27.86	30.59
Net Plant Work, MJ/kg brine	55.54	56.01	53.15	58.94	49.59	52.64
Net Thermo. Efficiency, %	12.74	12.97	12.89	14.42	14.29	12.65
Net Work/Availability	0.3341	0.3369	0.3198	0.3545	0.2983	0.3167
Power Plant Cost, $/kw	620	560	529	507	518	598
Total System Cost, $/kw	897	836	819	768	829	891
Brine Delivery						
Brine Flow, kg/kw	64.82	64.27	67.72	61.10	72.57	68.40
Brine Exit T, °C	94.39	96.94	101.8	103.17	119.61	99.5
Brine Delivery Cost, $/kw	278	275	290	262	311	293
Brine Heat Exchanger (BHE)						
BHE LMTD, °C	39.6	35.6	34.1	35.0	33.2	37.6
BHE load, ΔH, MJ/kg W.F.	379.14	453.6	453.8	449.8	205.5	158.1
BHE Cost, $/kw	173	162	143	155	180	184
Condenser						
Condenser Superheat ΔT, °C	12.9	26.0	36.1	9.6	10.7	26.2
Cond. Superheat ΔH, MJ/kg	25.63	50.82	68.85	15.58	6.51	20.49
Cond. Dew Point P, kpa	612	351	199	401	251	446
Cond. Dew Point T, °C	47.22	53.89	49.44	46.11	52.22	50
Condenser LMTD, °C	14.8	16.5	18.3	13.3	17.9	18.8
Cond. Load, ΔH, MJ/kg	330.8	394.9	395.4	384.9	176.1	138.1
Condenser Cost, $/kw	214	175	120	140	108	182
Cooling System						
Cooling Water Flow, kg/kw	577.0	466.3	512.6	493.5	395.5	582.0
Cooling Cost, $/kw	3792	2413	1379	2758	1724	3103
Turbine						
Turbine Inlet P, kpa	3792	2413	1379	2758	1724	3103
Turbine Inlet T, °C	142.2	141.7	133.3	136.7	140.6	146.1
Turbine ΔH, MJ/kg	55.80	63.43	61.13	69.76	30.70	22.47
Working Fluid Flow, kg/kw	81.15	65.95	66.41	59.69	130.8	196.2
W.F. Flow at Turbine Exit T, Cu. cm/sec/kw	1523	2142	3458	1823	2628	1847
Turbine-Generator Cost, $/kw	133	152	201	140	175	139
Cycle Pump						
Parasitic Power, MW	4.233	2.09	1.206	2.033	1.205	3.354
Pump Cost, $/kw	42	23	14	22	14	34

Table 5 - Comparison of Cycle Parameters for the Georesource Temperature of 149°C (300°F)

Process Parameters \ Fluid	Iso-Butane	Mixture iC_4H_{10}=50% iC_5H_{12}=50%	Iso-Pentane	Cis-2-Butene	R-11	R-114
Cycle						
Net Power, MW	25.00	25.00	25.00	25.00	25.00	25.00
Gross Power, MW	31.07	29.78	29.92	29.64	29.07	31.06
Net Plant Work, MJ/kg brine	26.40	24.52	22.52	22.26	20.98	22.70
Net Thermo. Efficiency, %	10.80	11.02	10.28	10.24	11.18	9.73
Net Work/Availability	0.3161	0.2932	0.2693	0.2664	0.2511	0.2715
Power Plant Cost, $/kw	868	775	819	682	779	870
Total System Cost, $/kw	1453	1405	1504	1374	1514	1549
Brine Delivery						
Brine Flow, kg/kw	136.3	146.9	159.9	161.7	171.5	158.7
Brine Exit T, °C	83.6	89.5	88.9	91.0	98.9	84.9
Brine Delivery Cost, $/kw	584	629	685	692	735	679
Brine Heat Exchanger (BHE)						
BHE LMTD, °C	20.3	19.6	21.0	22.7	19.9	22.7
BHE load, ΔH, MJ/kg W.F.	380.5	440.1	419.1	426.6	199.8	151.2
BHE Cost, $/kw	268	229	191	218	283	258
Condenser						
Condenser Superheat ΔT, °C	14.4	22.1	24.0	7.0	9.12	21.0
Cond. Superheat ΔH, MJ/kg	27.79	42.15	44.12	11.35	5.466	16.17
Cond. Dew Point P, kpa	538	301	163	401	210	377
Cond. Dew Point T, °C	42.22	48.5	42.78	46.11	46.11	43.89
Condenser LMTD, °C	10.3	11.5	12.1	13.0	13.2	12.5
Cond. Load, ΔH, MJ/kg W.F.	339.1	391.5	375.9	382.9	177.4	136.5
Condenser Cost, $/kw	345	278	236	217	194	337
Cooling System						
Cooling Water Flow, kg/kw	802.8	708.5	981.6	766.1	695.3	927.6
Cooling Cost, $/kw	59	52	72	56	51	68
Turbine						
Turbine Inlet P, kpa	2068	1379	689	1379	896	1379
Turbine Inlet T, °C	104.4	112.8	98.9	98.9	107.2	99.4
Turbine ΔH, MJ/kg	45.05	50.94	44.31	45.80	22.96	15.70
Working Fluid Flow, kg/kw	99.24	84.23	97.25	93.21	182.5	284.8
W.F. Flow at Turbine Exit T, Cu. cm/sec/kw	2124	3115	5946	2822	4323	3068
Turbine-Generator Cost, $/kw	146	177	283	155	220	158
Cycle Pump						
Parasitic Power, MW	2.512	1.414	0.845	1.37	0.783	1.956
Pump Cost, $/kw	27	16	10	16	10	21

Figure 2 illustrates the costs of major power plant equipment components as the percentages of the total system cost for the various working fluids for the 149°C georesource. It can be noted that the largest equipment component cost variations occur for the turbine-generator, condenser and brine heat exchanger. To a good extent, the differences in costs are due to differences in thermodynamic behavior from fluid to fluid. For the equimolar mixture of isobutane and isopentane, many aspects of property behavior fall almost linearly between the behavior of isobutane and isopentane. However, the fact that the turbine specific enthalpy drop is greater for the mixture than either pure fluid leads to a lower turbine-generator cost for the mixture than would be predicted from interpolation between the turbine-generator costs for the isobutane and isopentane cycles. This is a major reason for the lower total system cost for the mixture cycle at 149°C relative to the isobutane and isopentane total system costs. Some of the reasons cis-2-butene exhibits good performance on an equipment cost basis correspond to desirable working fluid characteristics discussed previously. (1) Cis-2-butene has a relatively large turbine specific enthalpy drop, leading to a relatively high thermodynamic efficiency and relatively low working fluid flow rate. (2) The cis-2-butene cycle operating pressures are reasonably low, leading to relatively low cost heat transfer surface area (say relative to isobutane). (3) There is little heat transfer in the superheat region for the cis-2-butene cycle, contributing to decreased heat transfer surface area requirements. It should be noted that R-11 would be more attractive on an economic basis if the turbine specific enthalpy drop for R-11 were greater (leading to lower working fluid flow rate and consequently turbine-generator cost). It can be realized from this investigation that a number of features of thermophysical property behavior contribute to making a fluid attractive (or unattractive) as a geothermal binary cycle working fluid. The best working fluids must have minimal unattractive features to make geothermal binary cycles competetive with alternative energy conversion processes.

CONCLUSIONS

A methodology has been developed to quickly identify promising working fluids for the geothermal binary power cycle. Preliminary calculations with a power cycle simulation program have indicated that cis-2-butene has some definite advantages over the working fluids currently considered by the geothermal industry. A detailed evaluation of the accuracy of the cis-2-butene thermophysical properties used in these calculations will be required to substantiate this conclusion. The methodology described herein is being used to expand the list of potential working fluids.

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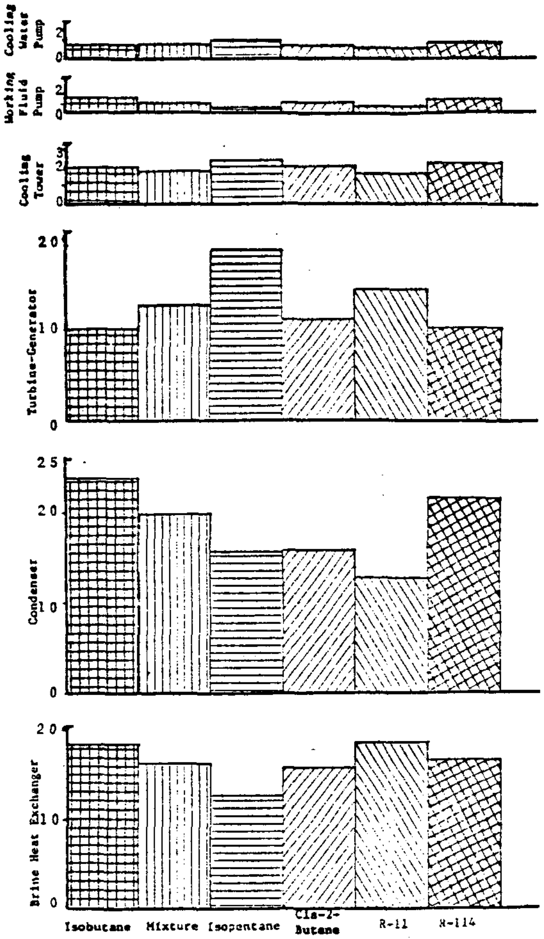

Fig. 2 - Power plant component costs as percentages of total system cost

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APPENDIX III

TABLE OF CHARACTERIZATION PARAMETERS FOR PRELIMINARY SCREENING OF WORKING FLUIDS

TABLE 1

RANKINE CYCLE WORKING FLUID THERMODYNAMIC PROPERTY CHARACTERIZATION PARAMETERS

Fluid	Formula	Molecular Weight	Critical Temperature °F	Critical Pressure psia	Critical Density lb mole/cu ft	Acentric Factor	Vapor Pressure at 100°F psia	ds/dT, Sat Vap at 100°F BTU/lb-°F^2	I-Factor	Specific Volume of Sat Vapor at 100°F Cu ft/lb	Normal Boiling Point °F
Water	H_2O	18.015	705.6	32	1.115	0.348	0.949	-0.0079	3.44	350.22	111.7
Ammonia	NH_3	17.03	270.3	1636	0.86	0.255	212.7	-0.0075	6.89	1.419	-29.5
Sulfur Dioxide	SO_2	64.059	315.5	1145	0.5117	0.246	88	-0.000332	2.68	0.995	13.4
R-13B1	$CBrF_3$	148.9	152.6	575	0.312	-	315.2	-0.000142	1.18	0.0852	-74.8
R-13	$CClF_3$	104.47	83.9	561	0.347	0.18	630	-	-	-	-115
R-12	CCl_2F_2	120.93	233.6	597	0.287	0.176	131.9	-0.0000664	0.91	0.308	-22
R-11	CCl_3F	137.34	390	639.4	0.252	0.184	23.6	-0.0000119	1.05	1.7645	74.6
R-10	CCl_4	153.8	541	661.5	0.226	0.194	4.32	-	-	-	169.5
R-14	CF_4	88	-50	524.4	0.016	0.191	-	-	-	-	-198.6
Carbon Disulfide	CS_2	76.14	585.4	1146	0.3672	0.115	11	-	-	-	114.9
R-22	$CHClF_2$	86.47	205	721.7	0.378	0.225	212.6	-0.0003162	1.12	0.014	-41.7
R-20	$CHCl_3$	119.38	505	793.8	0.261	0.216	3	-	-	-	141.7
Methylene Chloride	CH_2Cl_2	84.9	458	882	0.323	0.193	15	-	-	-	103.4
R-152a	CH_3CHF_2	66.05	236.3	652	0.0345	-	124.6	-0.0002529	1.159	0.061087	-
Methanol	CH_4O	32.04	462.7	1174.5	0.529	0.559	4.3	-	-	-	148
R-500	$(CCl_2F_2+CH_3CF_2)$	99.31	221.9	641.9	0.3121	0.2	156.61	-0.0001146	0.89	0.31	-28.3
R-503	(CHF_3+CClF_3)	87.5	67.1	60.7	-	-	-	-0.000556	-	-	-
R-504	$(CH_2F_2+CHF_2CF_3)$	79.2	151.5	690.4	-	-	376.3	-0.00031	-	0.14	-
R-502	$(CHClF_2+C_2F_5Cl)$	111.56	194	619	0.313	0.158	229.1	-0.000114	1.0	0.175	-
R-114	$C_2Cl_2F_4$	170.94	294.3	473.22	0.213	0.2495	46.4	+0.0000917	0.65	0.695	38.6
R-113	$C_2Cl_3F_3$	187.4	419	495.4	0.192	0.252	10.48	+0.00075	0.78	2.98	117.3
Perchloroethylene	C_2Cl_4	165.8	152	677.7	0.215	-	0.8	-	-	-	249.7
Trichloroethylene	C_2HCl_3	131.4	567.8	712.9	0.244	0.213	3	-	-	-	188.7
Cis,1-2,Dichloroethylene	$C_2H_2Cl_2$	96.95	519.8	850	-	-	7	-	-	-	138.2
Trans,1-2,Dichloroethylene	$C_2H_2Cl_2$	96.95	469.4	800	-	-	10	-	-	-	118

TABLE 1 (continued)

RANKINE CYCLE WORKING FLUID THERMODYNAMIC PROPERTY CHARACTERIZATION PARAMETERS

Fluid	Formula	Molecular Weight	Critical Temperature °F	Critical Pressure psia	Critical Density lb mole/cu ft	Acentric Factor	Vapor Pressure at 100°F psia	ds/dT, Sat Vap at 100°F BTU/lb-°F²	I-Factor	Specific Volume of Sat Vapor at 100°F Cu ft/lb	Normal Boiling Point °F
Acetyl Chloride	C_2H_3OCl	78.5	454.4	852.6	0.306	0.344	12	-	-	-	123.02
R-142b	$C_2H_3ClF_2$	100.5	279	598.2	0.27	-	71.5	-0.000048	1.24	0.724	14
R-140a	$C_2H_3Cl_3$	133.4	623	602.7	-	0.22	4.57	-	-	-	236.4
Acetonitrile	C_2H_3N	41.05	526.4	701.2	0.36	0.321	3	-	-	-	178.6
Chloral	$C_2H_3OCl_3$	147.4	577.4	646	-	-	1.5	-	-	-	207.7
R-150a	$C_2H_4Cl_2$	98.96	479	735	-	0.248	5.71	-	-	-	134.7
Propyl Chloride	$C_2H_4Cl_2$	98.96	549.8	779.1	0.284	0.286	11	-	-	-	134.7
Ethylene Oxide	C_2H_4O	44	384.2	1043	0.446	0.2	39	-	-	-	50.3
Acitic Acid	$C_2H_4O_2$	60.052	609.9	839.4	0.365	0.454	6	-	-	-	243.9
Ethyl Chloride	C_2H_5Cl	64.5	388.7	764.4	0.314	0.19	34	-	-	-	112.9
Ethane	C_2H_6	30.07	89.7	708.54	0.428	0.422	0.098	-	2.23	-	257.8
Ethanol	C_2H_6O	46.06	469	926.1	0.374	0.635	2.2	-	-	-	172.7
Ethanethiol	C_2H_6S	62.13	438.2	796.7	0.302	0.19	18	-	-	-	96
Dimethylsulfide	C_2H_6S	62.13	445.4	802.6	0.311	0.19	15	-	-	-	98.9
Ethylamine	C_2H_7N	45.08	360.8	815.9	0.351	0.284	28	-	-	-	61.5
R-216	$C_3Cl_2F_6$	220.93	355.9	399.5	0.1623	0.31	15.7	-0.0001202	1.42	1.6214	96.24
Propadiene	C_3H_4	40.06	249	793.8	0.391	0.273	156.3	-0.000222	1.55	1.186	-30.3
Propionitrile	C_3H_5N	55	555.9	607.9	0.271	0.318	1.7	-	-	-	206.9
Epichlorohydrin	C_3H_5ClO	92.5	613.4	721	-	-	0.7	-	-	-	208.9
Propylene	C_3H_6	42.08	200	670.3	-	0.14	227.5	-0.000259	1.0	0.465	54.3
1-2,Dichloropropane	$C_3H_6Cl_2$	112.98	588.6	646.8	0.276	0.24	1.8	-	-	-	205.1
Propionaldehyde	C_3H_6O	56	432	690.9	0.279	0.313	10	-	1.2	-	-
Acetone	C_3H_6O	58	456.6	682	0.299	0.309	7	-	-	-	132.9
Allyl Alcohol	C_3H_6O	58	521	829	0.307	0.63	1.0	-	-	-	206
Propylene Oxide	C_3H_6O	58	407.9	714.4	0.336	0.269	17	-	-	-	97.5

TABLE 1 (continued)

RANKINE CYCLE WORKING FLUID THERMODYNAMIC PROPERTY CHARACTERIZATION PARAMETERS

Fluid	Formula	Molecular Weight	Critical Temperature °F	Critical Pressure psia	Critical Density $\frac{\text{lb mole}}{\text{cu ft}}$	Acentric Factor	Vapor Pressure at 100°F psia	$^{ds}/_{dT}$, Sat Vap at 100°F $^{BTU}/_{lb\text{-}°F^2}$	I-Factor	Specific Volume of Sat Vapor at 100°F Cu ft/lb	Normal Boiling Point °F
Ethlyl Formate	$C_3H_6O_2$	74.08	455.1	687.96	0.273	0.283	8	-	-	-	129.3
Methyl Acetate	$C_2H_6O_2$	74.08	452.3	680.6	0.274	0.324	6.3	-	-	-	134.2
Ethylene Dichloride	C_3H_7Cl	78.54	446	663	-	-	2.9	-	-	-	115.7
Propane	C_3H_8	44	206.3	617.4	0.308	0.152	188.3	-0.000108	1.32	0.559	-44
1-Propanol	C_3H_8O	60.09	506	749.7	0.286	0.622	0.8	-	-	-	206.7
Furan	C_4H_4O	68.07	322.4	798.2	0.286	0.204	17	-	-	-	88
1-2,Butadiene	C_4H_6	54.092	340	652.7	0.285	0.255	36	+0.003088	0.99	2.88	51.2
1-3,Butadiene	C_4H_6	54.092	305	627.7	0.282	0.195	59.57	+0.0000247	1.0	1.7	23.7
Vinyl Acetate	$C_4H_6O_2$	86.1	485	632.1	0.236	0.34	4	-	-	-	162.8
Methyl Acrylate	$C_4H_7O_2$	86.1	504.8	617.4	0.236	0.35	3.5	-	-	-	176.3
Butyronitrile	C_4H_7N	69.1	587.96	549.8	0.219	0.35	0.75	-	-	-	243.8
I-Butene	C_4H_8	56.108	295.6	583.5	0.26	0.186	60.9	+0.000025	0.97	1.569	20.4
Cis-2-Butene	C_4H_8	56.108	324.4	610	0.266	0.2023	46.1	-0.000041	1.04	2.125	38.4
Trans-2-Butene	C_4H_8	56.108	311.86	595.35	0.262	0.214	49.8	+0.000014	0.93	2	33.2
Isobutylene	C_4H_8	56.108	292	580.65	0.261	0.19	68	-	0.91	-	19.3
2-Butanone	C_4H_8O	72.1	504.8	602.7	0.234	-	3.4	-	-	-	175
Tetrahydrofuran	C_4H_8O	72.1	512.4	752.6	0.279	-	5.5	-	-	-	150.4
Vinyl Ethyl Ether	C_4H_8O	72.1	395	590.9	0.24	0.363	5	-	-	-	95.8
N-Butyraldehyde	C_4H_8O	72.1	483.2	588	0.225	0.352	4	-	-	-	166.4
Isobutyraldehyde	C_4H_8O	72.1	463.4	602.7	0.228	0.35	6	-	-	-	146.6
Ethyl Acetate	$C_4H_8O_2$	88.1	482	555.7	0.218	0.363	3	-	-	-	170.5
N-Butane	C_4H_{10}	58.124	305.7	551.7	0.244	0.193	51.36	+0.000113	0.75	1.81	30.9
Isobutane	C_4H_{10}	58.124	274.9	529.1	0.237	0.177	72	+0.00013	0.81	1.26	10.3
Ethyl Ether	$C_4H_{10}O$	74.12	380.06	527.7	0.223	0.281	20	-	-	-	97.7
Butanol	$C_4H_{10}O$	74.12	553.2	640.9	0.229	0.59	2	-	-	-	243.6

TABLE 1 (continued)

RANKINE CYCLE WORKING FLUID THERMODYNAMIC PROPERTY CHARACTERIZATION PARAMETERS

Fluid	Formula	Molecular Weight	Critical Temperature °F	Critical Pressure psia	Critical Density $\frac{\text{lb mole}}{\text{cu ft}}$	Acentric Factor	Vapor Pressure at 100°F psia	$^{ds}/_{dT}$, Sat Vap at 100°F $^{BTU}/_{lb\text{-}°F^2}$	I-Factor	Specific Volume of Sat Vapor at 100°F Cu ft/lb	Normal Boiling Point °F
Isobutyl Alcohol	$C_4H_{10}O$	74.12	525.8	623.3	0.279	0.588	1	-	-	-	225.8
Butylamine	$C_4H_{11}N$	73.13	483.2	602.7	0.217	0.396	3	-	-	-	171
Pyridine	C_5H_5N	79.1	656	817.3	0.246	0.24	0.75	-	-	-	239
Ethyl Acrylate	$C_5H_8O_2$	100.118	533.6	543.9	0.195	0.4	1.5	-	-	-	211.4
Cyclopentane	C_5H_{10}	70.135	460	654.1	0.24	0.192	10.04	-0.000058	1.02	8.3	120.3
I-Pentene	C_5H_{10}	70.135	376.9	590	0.208	0.243	19.115	+0.000093	0.76	5.37	85.6
N-Pentane	C_5H_{12}	72.15	385.7	489.6	0.205	0.251	15.6	+0.000231	0.70	5.1	96.6
Isopentane	C_5H_{12}	72.15	369	490.9	0.204	0.227	20.4	+0.00025	0.77	3.85	81.8
Bromobenzene	C_6H_5Br	157.02	746	655.6	0.193	0.249	0.18	-	-	-	312.5
Chlorobenzene	C_6H_5Cl	112.56	678.3	655.6	0.203	0.249	0.4	-	-	-	268.8
Fluorobenzene	C_6H_5F	96.104	550	660	0.23	0.245	2.6	-	-	-	185.3
Benzene	C_6H_6	78.114	551.8	710	0.24	0.212	3.2	-	-	-	175.9
I-Hexene	C_6H_{12}	84.16	450	460.1	0.178	0.285	5.8	+0.000163	0.7	12	145.9
Methyl Isobutyl Ketone	$C_6H_{12}O$	100.16	567.8	474.8	0.168	0.4	0.4	-	-	-	241.3
Isohexane	C_6H_{14}	86.178	435.5	436.6	0.17	0.279	7	-	-	-	140.12
N-Hexane	C_6H_{14}	86.178	453.7	430.9	0.169	0.3	4.95	-0.000031	0.73	13.73	155.4
Propyl Ether	$C_6H_{14}O$	102.17	440	417.5	0.162	0.34	4	-	-	-	157.7
Triethylamine	$C_6H_{15}N$	101.2	471	441	0.16	0.329	2.4	-	-	-	192.7
Toluene	C_7H_8	92.14	605.5	611.5	0.193	0.26	1.0	-0.000065	0.12	62.8	230.8
I-Heptene	C_7H_{14}	98.19	505	411.6	0.142	0.358	1.9	-	-	-	200.2
N-Heptane	C_7H_{16}	100	513	358	0.145	-	1.6	-	-	-	208.9
Ethyl Benzene	C_8H_{10}	106.16	650.8	523.3	0.167	0.301	0.1	-	-	-	276.74
Isooctane	C_8H_{18}	114.23	547.3	360.1	0.128	0.378	1.7	-	-	-	249.9
N-Octane	C_8H_{18}	114.23	563.8	360.15	0.1	0.129	0.4	-	-	-	-127.9
Propylbenzene	C_9H_{12}	120.19	688.9	464.5	0.142	0.344	0.15	-	-	-	318.3

APPENDIX IV

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

Advantages and Disadvantages of the use of Mixtures in Geothermal Binary Cycles

The use of hydrocarbon mixture working fluids in the geothermal binary cycle has been advocated in research carried out at the University of Oklahoma (1,2,3). Isobutane-isopentane binary mixtures were emphasized in the earlier analyses. In the present study, the scope of the investigation has been broadened to include other hydrocarbon systems (eg., binary mixtures of propane, cis-2-butene and cyclopentane). The use of mixtures in cascade (multi-boiler) binary cycles also has been considered. The major conclusions drawn from this work are summarized here as advantages and disadvantages of the use of mixtures in single boiler and cascade binary cycles. A discussion of hardware aspects also is given.

Advantages of Mixtures

(1) For subcritical cycles, binary mixtures yield larger isentropic enthalpy changes per unit mass of the working fluid than do the constituent pure fluids. The larger enthalpy change for the mixture cycle results in increased net thermodynamic cycle specific work, and increased net thermodynamic cycle efficiency. For example, an equimolar mixture of isobutane and isopentane provides approximately 13% and 15% more turbine specific work than isobutane and isopentane single boiler binary cycles for the georesource temperature of 300°F.

(2) The condenser cost per unit area of heat transfer surface is smaller for the isobutane-isopentane mixture cycle than for the pure isobutane cycle because the condensation

pressure (and hence design pressure) for the mixture is lower than for isobutane.

(3) When the design objective is the maximization of the net plant work per unit mass of brine, binary mixtures (of propane and cis-2-butene) provide significantly greater amounts of work than the pure fluids cis-2-butene and isobutane in single boiler and cascade binary cycles.

(4) Because of the nonisothermal mixture condensation profile, a larger cooling water temperature rise can be employed for mixture cycles than for pure fluids cycles, thereby reducing cooling water requirements and reducing cooling tower cost.

(5) The mixture composition can be varied over the plant lifetime to match the changing characteristics of the geothermal resource.

(6) For equal arithmatic MTD's (mean temperature differences), the condenser logarithmic MTD of a mixture cycle is greater than for a pure fluid cycle, leading to smaller heat transfer surface area requirements for a given duty.

(7) For subcritical cycles, the brine heat exchanger logarithmic MTD is greater for the mixture cycle than for the pure fluid cycle for equal arithmatic MTD's, leading to smaller heat transfer surface area requirements for a given duty.

(8) For subcritical cycles, the isobutane-isopentane binary mixture heat transfer coefficients in brine heat exchanger are higher than for pure isobutane or for pure isopentane. This is due to the nonisothermal behavior of the mixture which

increases the average design temperature, which in turn decreases the mixture liquid viscosity and thermal conductivity. The fact that liquid viscosity decreases more rapidly than thermal conductivity coupled with an increase in the heat capacity results in better heat transfer coefficients for mixtures compared to pure fluids.

Disadvantages of Mixtures

(1) Mixture working fluid heat transfer coefficients in the condenser are smaller than for pure fluids because of lower average design temperatures. This is due to the nonisothermal condensation of the mixtures, resulting in lower bubble point temperatures compared to pure fluids (which condense isothermally at constant pressure).

(2) In the design of the brine heat exchanger, the working fluid has been considered to flow on the shell side with brine on the tube side, with an assumption of constant slip. Constant slip means that the slip velocity, V_s, is zero; or in other words, the velocities of gas and liquid phases are identical at each point in the heat exchanger. Since V_s is dependent on the flow regime, turbulence, fluid properties and the length of the conduit (4), the gas velocity is usually higher than the liquid velocity and the working fluid recombined (liquid plus vapor) composition would change as the fluid progresses in the two-phase section. Thus, experimental heat transfer coefficient data will be required to verify the theoretical values of heat transfer coefficients. Moreover, in the comparisons of mixture and pure fluid cycles to date, mass transfer effects, which occur in the two-phase

regions of the brine heat exchanger have been ignored. Thus, only mass transfer effects and the related fact that vapor and liquid in two-phase flow are of different velocity would be expected to detract from the advantages of mixture cycles over pure fluid cycles (5). These disadvantages are expected to be small and the brine heat exchanger for mixtures should be smaller than for pure fluids.

(3) Diffusive mass transfer effects have not been taken into account in simulations to date in the mixture fluid two-phase heat transfer coefficients of the condenser. The resistance to diffusive mass transfer in the vapor phase will cause condensers to be larger than in the absence of diffusive mass transfer. From a preliminary study of diffusive mass transfer effects on condenser performance(5), it was concluded that the net disadvantages of the diffusive effects of mixtures in geothermal cycles seem to be offset by the advantages mixtures indicate with respect to their thermodynamic behavior.

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APPENDIX V

EQUIPMENT CONSIDERATIONS PECULIAR TO GEOTHERMAL BINARY CYCLES USING MIXTURE WORKING FLUIDS

Equipment Considerations Peculiar to Geothermal Binary Cycles Using Mixture Working Fluids

The use of mixture working fluids in geothermal binary cycles has been proposed in research at the University of Oklahoma (1,2). To date, only horizontal, countercurrent, single pass shell and tube exchangers and condensers have been considered in these studies, with the working fluid on the shell side and brine and cooling water on the tube side of heat exchanger and condenser, respectively. In this report other types of heat exchange equipment are considered which may enhance the performance of mixture working fluids in GBC. Other factors taken into consideration are: types of flow, working fluid location, tube arrangement.

Brine Heat Exchanger

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

(1) Only one shell pass and one tube pass, horizontal, counter-current heat exchangers have been considered. Because of non-isothermal heat transfer between the working fluid and the hot brine, counter-current flow will have a distinct thermal advantage over the co-current flow because in co-current flow, the hot fluid (brine in this case) cannot be cooled below the cold fluid (working fluid in this case) outlet temperature (3).

(2) If brine fouling can be controlled, or if there is little fouling, then the working fluid should be on the tube side and the brine on the shell side. The assumption of working fluid phase equilibrium at each point in the two-phase section of the heat exchanger may be adequate if the working

fluid is on the tube side. This would be a good assumption if the slip velocity, $V_s = V_g - V_\ell$, were zero; i.e., both gas and liquid phases travelling at the same velocity. V_s is dependent on the flow regime, turbulance, fluid properties and the length of the conduit (4). The gas velocity, V_g, is usually higher than the liquid velocity, V_ℓ, as the fluid progresses in the two-phase section. The composition of the fluid therefore changes as it flows through the tube. Although the assumption that phase equilibrium exists is not exact, it is the best one that can be made without very tedious calculations. A survey by DeGance does not favor the flow-regime based correlations because of the additional errors associated with the calculation of flow regimes (5).

(3) The liquid/vapor ratio for mixture working fluids can be calculated for tube side flow, if the heat exchanger two-phase section is divided into large numbers of subsections with respect to pressure or temperature. The pressure or temperature interval midpoint may then be used to get the value of L/V by assuming phase equilibrium in each subsection. DeGance recommends Dukler's constant slip method for horizontal flow pressure drop calculations (6). Constant slip would mean that the slip velocity, V_s, is zero; or in other words, the velocities of gas and liquid phase are identical at each point in the heat exchanger. The heat transfer coefficient correlation for two-phase flow used in the GEO4 simulator developed at the University of Oklahoma is probably adequate for design purposes.

(4) Tubes are arranged in triangular, square or rotated square pitch. Triangular tube layouts result in better shell side coefficients and provide more surface area in a given shell diameter (3). The GEO4 simulator uses triangular tube pitch and therefore no change is needed.

(5) Higher circulation rates are possible on the tube side compared to the shell side, and the tube side flow distribution is more uniform. Mixture working fluid heat transfer coefficients are higher than pure fluids when the working fluid is on the shell side. Therefore, even higher heat transfer coefficients may be obtained for mixture working fluids flowing on the tube side because of higher circulation rates.

(6) It has been shown that sand fluidized by the geothermal brine on the shell side prevents scaling and increases the brine side heat transfer coefficient over conventional shell and tube heat exchangers (7). Both horizontal and vertical-tube bundle arrangements are being considered (8). In preliminary evaluations, the vertical tube model appears to have more uniform flow distribution and slightly higher heat transfer coeffifients than horizontal models (at the same cross-sectional velocity); the converse is true when no bed is involved. However, the heat transfer coefficient of the horizontal-tube model can be brought closer to the vertical-tube model by some design improvements. Since fluidized bed heat exchangers could reduce the size of the heat exchanger by as much as 50%, the development of such heat exchangers would definitely enhance the chances of working fluids being used on the tube side. Under such circumstances, the use of fluidized-bed vertical-tube heat exchangers must be given serious consideration.

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

Condenser

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

Many of the same advantages are offered by the vertical vapor-in-shell condensers that use baffles designed to permit condensate to remain on the tube (9). The condensing two-phase heat transfer coefficient can be increased by introducing turbulence, or in other words by increasing Reynolds and Prandtl numbers. This may also be done by employing low-finned tubes or fluted tubes.

The mixture working fluid heat transfer coefficients can also be increased by employing vertical vapor-in-tube condensers. The vertical condenser could be of the conventional shell and tube exchanger type or the spool-wound exchanger shell type. A spool-wound exchanger consists of a layer of spirally wound tubes around a core with a fairly small pitch (10). The working fluid mixture would then condense downward within the tubes, while the cooling water would flow within the shell and between the tubes. This would require technology transfer from the LNG (liquefied natural gas) industry where spool-wound exchangers are used. Reference 10 indicates that mixtures may provide better heat transfer compared to pure fluids for identical pressure drop ratios of liquid and gas phases.

In spool-wound tube exchangers, the tube side working fluid mixture composition in liquid and gas phase at each point in the exchanger subsection can be assumed to be constant if the exchanger is subdivided into a large number of subsections. The vapor phase working fluid enters the exchanger at high velocity. As the mixture fluid condenses, the liquid phase (condensate)flows along with the uncondensed gas phase. Initially the gas and liquid phase velocities are different--the gas phase velocity being on the higher side. However, under the conditions of equilibrium (because of the increased turbulence and increased vapor liquid contact in spool-wound tubes) a constant slip may be assumed. The Dukler correlation applies for vertical tubes, and can be modified for the spool-wound tubes (9,10).

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

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

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APPENDIX VI

SPECIFICATION OF PERMISSIBLE COMMERCIAL ISOBUTANE COMPOSITION RANGE FOR ACCEPTABLE OPERATION OF LOW TEMPERATURE ISOBUTANE GEOTHERMAL BINARY CYCLES

SPECIFICATION OF PERMISSIBLE COMMERCIAL ISOBUTANE COMPOSITION RANGE FOR ACCEPTABLE OPERATION OF LOW TEMPERATURE ISOBUTANE GEOTHERMAL BINARY CYCLES

by

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ABSTRACT

The permissible ranges of impurities in commercial isobutane which can be tolerated for acceptable operation of low temperature geothermal binary cycles were determined. The method used was to consider working fluid mixtures which simulate commercial grade isobutane in geothermal binary cycle simulation calculations. The basis of calculation is a power plant using 275°F brine designed using pure isobutane as the working fluid. The permissible commercial isobutane composition is 95 mol % minimum isobutane, 5 mol % maximum n-butane, 3 mol % maximum propane, 0.1 mol % maximum ethane or pentanes. The compositions of available commercial grade isobutanes are generally within this permissible range. These composition ranges should be acceptable for cycles with working fluid operating conditions which do not approach the critical region.

1. Introduction

Because geothermal binary cycle power plants have been designed to use pure isobutane as the working fluid, it is desired to detect the effect on plant power performance when a commercial grade of isobutane is used as the working fluid instead of pure isobutane. In this study, a number of working fluid mixture compositions which simulate commercial grades of isobutane were utilized in geothermal binary cycle simulation calculations. In order to compare the performance of pure isobutane with readily available commercial isobutanes, a fixed plant type of simulation was performed to carry out the calculations. For the fixed plant type of simulation, the brine temperature at the brine heat exchanger inlet, the brine flow rate, the pressure of working fluid at the turbine inlet, and the total heat transfer areas of both the brine heat exchanger and the condenser are fixed in the calculations. However, the temperature of the working fluid at the turbine inlet will be different due to differences in dew point temperatures for each working fluid composition tested. The results of the simulations of these commercial isobutanes are compared with that of pure isobutane.

2. Fixed Plant Calculations

The conventional geothermal binary cycle simulator, GEO4, developed at the University of Oklahoma, was modified to perform the fixed plant calculations. Since the total heat transfer areas of both the brine heat exchanger and the condenser cannot be automatically fixed in GEO4, a method was developed to let the areas converge to the desired values. To carry out the fixed plant calculation, once thru calculations using GEO4 are performed separately several times. The program control parameter, NOPT, is set equal to zero to perform the once thru calculation. For the first run the parameters which were obtained from a pure isobutane calculation by GEO4 are used as the input data. The parameters of pressure drops in both the brine heat exchanger and condenser are updated for the next calculation from the first simulation result. These parameters are PCW10, PHW12, DPWF35, DPWF81, DPLIQS, DPLIQT, DP2TUB, DP2S, DPC4, and DPCW4. The total heat transfer areas of both the brine heat exchanger and the condenser are also obtained from the first simulation. If the two total areas are not close enough to those of pure isobutane, the pinch point temperature differences, DTHWO and DTCWO, need to be adjusted in the next calculation. Increases of the pinch point temperature differences will increase the log mean temperature differences of the two heat exchangers, and thus decrease the heat transfer area. These parameters, the pressure drops and the pinch point temperature differences, are updated in each calculation until the heat transfer areas approach the desired values.

3. Simulation Results and the Permissible Range of Impurities in Commercial Grade Isobutane

The impurities most commonly found in commercial grade isobutanes are propane and n-butane. Table 1 shows the compositions of commerical grade isobutanes available from three different isobutane suppliers. Hence a range of isobutane-propane and isobutane-n-butane mixtures were studied in cycle simulation calculations. The results of these calculations are compared with that of the pure isobutane. The comparisions are based on the same brine inlet temperature, brine flow rate, working fluid turbine inlet pressure and total heat transfer areas of the brine heat exchanger and condenser. The objective function used for comparison is net plant power output. Table 2 gives the simulation results for a range of isobutane-propane mixtures from 3 to 10 mol. percent propane. The 10 mol % propane cycle yields 24.12 MW net plant power which is almost a 2% decrease when compared with the pure isobutane. From these results, we can say that commercial grades of isobutane may contain up to 5 mol % propane for a net plant power loss below 2%. The effect of n-butane is given in Table 3. The composition range of n-butane is from 5 to 20 mol %. Note that the n-butane shows almost no influence on the net plant power output. This result is to be expected since isobutane and n-butane are isomers. A trace amount of ethane and pentanes are also contained in commercial grade isobutanes. These impurities were also tested. It can be seen from Table 4 that the isobutanes which contain 0.1 mol % ethane and pentane produce no effect on the net plant power. Three available commercial grade isobutanes were also utilized in cycle simulation calculations. Their results are given in Table 5. The net plant power obtained from isobutane A and isobutane B have no significant difference from the pure isobutane. Isobutane C gives 24.92 MW, which is 1.3% greater than the pure isobutane. Also note from Table 1, the isobutane which contains 2-3% propane can yield the highest net plant power.

4. Conclusions

If a 1% decrease in net plant power output is taken as acceptable, a permissible commercial isobutane composition range for acceptable operation of geothermal binary cycles can be specified as follows:

Isobutane	at least 95 mol %
Propane	up to 3 mol %
n-butane	up to 5 mol %
ethane	trace (up to 0.1 mol %)
i-pentane	trace (up to 0.1 mol %)

The compositions of a number of available commercial grade isobutanes are within this range. In other words, using available commercial isobutane as the working fluid instead of pure isobutane in a geothermal power plant should cause no significant effect on the net power output, so long as working fluid operating conditions do not approach the critical region.

Table 1. Compositions & Suppliers of Commercial Grade Isobutane

Supplier		Phillips Co.*	Cities Service Co.*	Williams Energy Co.†
Composition	Isobutane	95% min.	97.12% min.	95% min.
	n-Butane	5% max.	1.57% max.	5% max.
	Propane	1% max.	1.31% max.	3% max.
	Pentanes	0.1% max.	---	0.5% max.

* Mole Percent

† Percent by Liquid Volume

Table 2. Fixed Plant Simulation Results

Working Fluid Process Parameters	100% i-Butane	95% i-Butane 2% n-Butane 3% Propane	95% i-Butane 5% Propane	90% i-Butane 10% Propane
Brine Flow Rate, MMlb/hr	10.505	10.505	10.505	10.505
Brine Inlet Temp., °F	275.0	275.0	275.0	275.0
Brine Exit Temp., °F	184.0	181.5	179.4	175.5
Turbine				
W.F. Inlet T, °F	212.3	210.8	209.1	205.8
W.F. Inlet P, psia	275.0	275.0	275.0	275.0
W.F. Enthalpy Change, BTU/lb	17.70	17.38	16.92	15.79
Gross Power, MW	30.79	31.23	31.29	31.71
Diameter, ft	5.073	5.117	5.143	5.158
Brine Heat Exchanger				
Pinch Point ΔT, °F	9.49	9.20	9.52	10.27
Shell Side ΔP, psia	10.38	12.39	13.11	13.10
Overall Heat Trans. Coeff., BTU/hr-ft^2-°F	213	221	223	226
LMTD, °F	28.8	28.7	29.1	30.0
Total Duty, 10^8BTU/hr	9.616	9.925	10.153	10.562
Total Heat Trans. Area, ft^2	156,368	156,350	156,354	156,375
Condenser				
Pinch Point ΔT, °F	10.41	11.50	12.43	15.96
Shell Side ΔP, psia	2.30	1.43	1.23	1.11
Overall Heat Trans. Coeff., BTU/hr-ft^2-°F	162	163	162	150
LMTD, °F	20.0	20.6	21.2	24.0
Total Duty, 10^8BTU/hr	8.647	8.943	9.085	9.601
Total Heat Trans. Area, ft^2	266,987	267,069	266,991	267,087
Cooling Water Flow Rate, MMlb/hr	43.199	44.679	45.814	47.968
Working Fluid Flow Rate, MMlb/hr	5.936	6.132	6.312	6.640
Cycle Pump Work, MW-hr	2.40	2.46	2.52	2.56
Cooling Water Pump Work, MW-hr	2.26	2.30	2.34	2.41
Brine Pump Work, MW-hr	0.72	0.72	0.72	0.72
Cooling Tower Fan Work, MW-hr	0.81	0.84	0.86	0.90
Net Thermo. Cycle Work, MW-hr	28.39	28.77	28.77	28.15
Net Thermo. Efficiency, %	10.078	9.891	9.673	9.098
Net Plant Work, MW-hr	24.60	24.91	24.85	24.12

Table 3. Fixed Plant Simulation Results

Working Fluid Process Parameters	100% i-Butane	95% i-Butane 5% n-Butane	90% i-Butane 10% n-Butane	80% i-Butane 20% n-Butane
Brine Flow Rate, MMlb/hr	10.505	10.505	10.505	10.505
Brine Inlet Temp., °F	275.0	275.0	275.0	275.0
Brine Exit Temp., °F	184.4	185.5	186.6	189.2
Turbine				
W.F. Inlet T, °F	212.3	213.5	214.6	216.9
W.F. Inlet P, psia	275.0	275.0	275.0	275.0
W.F. Enthalpy Change, BTU/lb	17.70	18.08	18.45	19.22
Gross Power, MW	30.79	30.77	30.73	30.50
Diameter, ft	5.073	5.054	5.034	4.985
Brine Heat Exchanger				
Pinch Point ΔT, °F	9.49	9.25	9.09	8.68
Shell Side ΔP, psia	10.38	10.03	9.68	8.92
Overall Heat Trans. Coeff., BTU/hr-ft^2-°F	213	213	212	210
LMTD, °F	28.8	28.6	28.4	27.8
Total Duty, 10^8BTU/hr	9.616	9.506	9.390	9.116
Total Heat Trans. Area, ft^2	156,368	156,406	156,313	156,338
Condenser				
Pinch Point ΔT, °F	10.41	10.06	9.68	8.85
Shell Side ΔP, psia	2.30	2.34	2.40	2.50
Overall Heat Trans. Coeff., BTU/hr-ft^2-°F	162	164	166	169
LMTD, °F	20.0	19.5	19.0	18.0
Total Duty, 10^8BTU/hr	8.647	8.536	8.420	8.151
Total Heat Trans. Area, ft^2	266,987	266,931	266,940	266,892
Cooling Water Flow Rate, MMlb/hr	43.199	42.646	42.068	42.722
Working Fluid Flow Rate, MMlb/hr	5.936	5.811	5.685	5.416
Cycle Pump Work, MW-hr	2.40	2.36	2.31	2.23
Cooling Water Pump Work, MW-hr	2.26	2.24	2.22	2.18
Brine Pump Work, MW-hr	0.72	0.72	0.72	0.72
Cooling Tower Fan Work, MW-hr	0.81	0.80	0.79	0.76
Net Thermo. Cycle Work, MW-hr	28.39	28.42	28.42	28.27
Net Thermo. Efficiency, %	10.078	10.203	10.330	10.584
Net Plant Work, MW-hr	24.60	24.66	24.69	24.60

Table 4. Fixed Plant Simulation Results

Working Fluid Process Parameters	100% i-Butane	99.9% i-Butane 0.1% Ethane	99.9% i-Butane 0.1% i-Pentane
Brine Flow Rate, MMlb/hr	10.505	10.505	10.505
Brine Inlet Temp., °F	275.0	275.0	275.0
Brine Exit Temp., °F	184.4	184.2	184.5
Turbine			
W.F. Inlet T, °F	212.3	212.2	212.4
W.F. Inlet P, psia	275.0	275.0	275.0
W.F. Enthalpy Change, BTU/lb	17.70	17.66	17.74
Gross Power, MW	30.79	30.79	30.79
Diameter, ft	5.073	5.075	5.072
Brine Heat Exchanger			
Pinch Point ΔT, °F	9.49	9.56	9,49
Shell Side ΔP, psia	10.38	10.40	10.28
Overall Heat Trans. Coeff., BTU/hr-ft^2-°F	213	213	213
LMTD, °F	28.8	28.9	28.8
Total Duty, 10^8BTU-hr	9.616	9.647	9.607
Total Heat Trans. Area, ft^2	156,368	156,325	156,340
Condenser			
Pinch Point ΔT, °F	10.41	10.60	10.41
Shell Side ΔP, psia	2.30	2.27	2.30
Overall Heat Trans. Coeff., BTU/hr-ft^2-°F	162	162	162
LMTD, °F	20.0	20.0	19.9
Total Duty, 10^8BTU/hr	8.647	8.678	8.638
Total Heat Trans. Area, ft^2	266,987	267,020	266,940
Cooling Water Flow Rate, MMlb/hr	43.199	43.354	43.154
Working Fluid Flow Rate, MMlb/hr	5.936	5.953	5.927
Cycle Pump Work, MW-hr	2.40	2.40	2.39
Cooling Water Pump Work, Mw-hr	2.26	2.26	2.26
Brine Pump Work, MW-hr	0.72	0.72	0.72
CoolingTower Fan Work, MW-hr	0.81	0.81	0.81
Net Thermo. Cycle Work, MW-hr	28.39	28.39	28.39
Net Thermo. Efficiency, %	10.078	10.045	10.089
Net Plant Work, MW-hr	24.60	24.60	24.61

Table 5. Fixed Plant Simulation Results

Working Fluid Process Parameters	100% i-Butane	95% i-Butane[*] 4% n-Butane 1% Propane	97.12% i-Butane[**] 1.57% n-Butane 1.31% Propane	94.5% i-Butane[†] 3.1% n-Butane 2.4% Propane
Brine Flow Rate, MMlb/hr	10.505	10.505	10.505	10.505
Brine Inlet Temp., °F	275.0	275.0	275.0	275.0
Brine Exit Temp., °F	184.4	184.3	183.6	182.2
Turbine				
W.F. Inlet T, °F	212.3	212.6	211.8	211.5
W.F. Inlet P, psia	275.0	275.0	275.0	275.0
W.F. Enthalpy Change, BTU/lb	17.70	17.83	17.61	17.55
Gross Power, MW	30.79	30.85	30.87	31.20
Diameter, ft	5.073	5.069	5.081	5.108
Brine Heat Exchanger				
Pinch Point ΔT, °F	9.49	9.45	9.60	9.09
Shell Side ΔP, psia	10.38	10.33	10.56	12.14
Overall Heat Trans. Coeff., BTU/hr-ft^2-°F	213	213	214	221
LMTD, °F	28.8	28.7	28.9	28.5
Total Duty, 10^8BTU/hr	9.616	9.630	9.706	9.848
Total Heat Trans. Area, ft^2	156,368	156,887	156,673	156,356
Condenser				
Pinch Point ΔT, °F	10.41	10.62	10.88	11.21
Shell Side ΔP, psia	2.30	2.21	2.07	1.55
Overall Heat Trans. Coeff., BTU/hr-ft^2-°F	162	163	163	163
LMTD, °F	20.0	19.9	20.2	20.4
Total Duty, 10^8BTU/hr	8.647	8.658	8.735	8.867
Total Heat Trans. Area, ft^2	266,987	266,147	266,090	267,004
Cooling Water Flow Rate, MMlb/hr	43.199	43.258	43.640	43.299
Working Fluid Flow Rate, MMlb/hr	5.936	5.906	5.985	6.068
Cycle Pump Work, MW-hr	2.40	2.38	2.41	2.44
Cooling Water Pump Work, MW-hr	2.26	2.26	2.27	2.29
Brine Pump Work, MW-hr	0.72	0.72	0.72	0.72
Cooling Tower Fan Work, MW-hr	0.81	0.81	0.82	0.83
Net Thermo. Cycle Work, MW-hr	28.39	28.47	28.46	28.76
Net Thermo. Efficiency, %	10.078	10.089	10.008	9.967
Net Plant Work, MW-hr	24.60	24.68	24.65	24.92

[*] Isobutane A; [**] Isobutane B; [†] Isobutane C

APPENDIX VII

ANALYSIS OF THE SENSITIVITY OF CONVENTIONAL GEOTHERMAL BINARY CYCLE SIMULATIONS TO VARIATIONS IN THERMODYNAMIC PROPERTIES CORRELATION PARAMETERS FOR ISOBUTANE

ANALYSIS OF THE SENSITIVITY OF CONVENTIONAL GEOTHERMAL BINARY CYCLE SIMULATIONS TO VARIATIONS IN THERMODYNAMIC PROPERTIES CORRELATION PARAMETERS FOR ISOBUTANE

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ABSTRACT

This work demonstrates that the OBWR (original Benedict-Webb-Rubin) equation and the MBWR (modified Benedict-Webb-Rubin) equation with various parameter sets yield geothermal cycle simulations in reasonable agreement for operating conditions excluding the critical region. For operating conditions in the critical region, only an extremely accurate correlation can be used for design purposes. Because of the inherent problem of control of power plants, the critical region is avoided by most designers. Therefore, the more accurate available correlations of isobutane thermodynamic behavior should be adequate for most design situations. For a 300°F georesource, the OBWR equation and MBWR equation with 1973, 1977 or 1978 parameters yield essentially equivalent design simulations. For the 350°F resource cycle the predicted bubble point enthalpy in the brine heat exchanger is erroneously high using the 1973 MBWR parameters. For the 400°F georesource, there is again good agreement between the different correlations. The conclusion is that although previous design calculations using the 1973 MBWR parameters were adequate for conditions far removed from the critical point, the more recent MBWR parameters (1977, 1978) are recommended.

1. Introduction

In this study, the MBWR (modified Benedict-Webb-Rubin) equation of state with three different sets of parameters for isobutane was utilized in conventional geothermal binary cycle simulation calculations to determine differences in operating conditions and net power output. These three sets of MBWR equation of state parameters, as shown in Table 1, were determined for isobutane in 1973, 1977, and 1978, respectively. Also, using the mixture thermodynamic properties computer program, HSGC, several calculations of isentropic turbine expansions at conditions corresponding to geothermal binary cycles were performed.

2. Isentropic Expansion Calculations

The results of the isentropic expansion calculations are shown in Tables 2, 3, 4 and 5. As a matter of information, results also are given for the OBWR (original Benedict-Webb-Rubin) equation and the GMBWR (generalized modified BWR) equation. However, the GMBWR equation never has been recommended for pure isobutane cycle calculations, rather only for the study of composition effects in mixture cycle studies. Among the four sets of parameters under consideration, the 1977 MBWR parameters gave the highest and the original BWR parameters gave the lowest values in enthalpies, entropies and enthalpy changes across the turbine. The turbine enthalpy drops for the 1973 and 1977 MBWR isobutane parameter sets have maximum differences of 2% from the turbine enthalpy drops for the 1978 MBWR parameter set (which are believed to be the most accurate representation of isobutane thermodynamic properties used herein).

3. Binary Cycle Simulation Design Calculations

The results of using different MBWR parameter sets in single boiler conventional (shell and tube heat exchanger) geothermal binary calculations are summarized for georesource temperatures of 300°F, 350°F, and 400°F in Tables 6, 7 and 8, respectively. For each georesource temperature, the quantities which are fixed are brine flow rate, turbine inlet temperature and pressure, brine heat exchanger approach and pinch temperature differences, condenser approach and pinch temperature differences, and condenser cooling water temperature rise. The major differences in results due to different MBWR parameters are in the brine outlet temperature, the enthalpy change across the brine heat exchanger, the turbine outlet temperature and pressure, the enthalpy change across the turbine, the working fluid flow rate, the gross turbine power and net plant power. It can be noted in Tables 6 and 8 for the georesource temperatures of 300°F and 400°F that all of these quantities are in close agreement with the 1973, 1977 and 1978 MBWR parameters. However, for the 350°F georesource in Table 7, the brine outlet temperature is 188.3°F using the 1973 MBWR parameters, compared to 206.4°F and 202.5°F, respectively using the 1977 and 1978 MBWR parameters. These results occur because the 1973 MBWR parameters predict the bubble point enthalpy at 454.9 psia to be 120.9 BTU/lb, which is about 4.0 Btu/lb too high. The more accurate 1977 and 1978 MBWR parameters predict the isobutane bubble point enthalpy at 454.9 psia to be 116.3 Btu/lb and 116.9 Btu/lb, respectively. Table 9 gives bubble point predictions using OBWR and MBWR parameters. The fact that the bubble point enthalpy predicted by the 1973

MBWR parameters is high leads not only to an erroneously low brine outlet temperature but consequently also to erroneously high values of working fluid flow rate, gross turbine work and net plant work. Pressure-enthalpy diagrams for isobutane obtained by using different equation of state parameters are shown in Figures 1, 2, 3, and 4. A dashed curve in Figure 4 shows the saturation envelope obtained using the 1973 MBWR parameters superimposed on the P-H diagram using the 1978 MBWR parameters. Note that the bubble point enthalpy predicted from 1973 and 1978 MBWR parameters deviate in the region above 400 psia.

4. Simulations with Fixed Heat Transfer Surface Areas

The basis for comparison is a geothermal power plant designed using the 1973 MBWR equation of state parameters. The georesource temperature is 300°F. The brine flow rate is 7.5156×10^6 lb/hr. The working fluid is isobutane. The temperature and pressure of the working fluid at the turbine inlet are 220°F and 300 psia, respectively. The total heat transfer area of the brine heat exchanger is 115,307 ft^2 and that of the condenser is 273,879 ft^2.

The cycle simulation results using different correlation parameters are shown in Table 10. The use of 1973 MBWR parameters in cycle simulation yields the highest net plant work, 25.39 MW-hr. The net plant work obtained using the 1977 MBWR parameters, is 25.06 MW, which is 1.3% less than the highest value.

From these results, we can conclude that using different correlation parameters in the design calculations does not significantly influence net plant work ouput for the case of a 300°F georesource. The conclusions from using the fixed heat transfer surface area simulation in Table 10 are so similar to the conclusions from the design simulation in Table 6 that fixed area simulations were not performed for the higher temperature georesource conditions.

5. Conclusions

(1) The use of different correlation parameters in conventional geothermal binary cycle simulation calculations generally causes only small differences in operating conditions if predicted thermodynamic properties are accurate in the region.

(2) For operating conditions in the critical region there is great sensitivity to differences in correlation parameters, so that the use of a highly accurate correlation is mandatory. On the other hand, due to control problems, operation too close to the critical point should be avoided in the design of power plants.

(3) The working fluid enthalpy change across the turbine is not the major factor in determining the net plnat power output. The net plant power output is heavily influenced by the bubble point temperature in the brine heat exchanger, which in turn affects the brine outlet temperature and working fluid flow rate for given brine flow rate and pinch temperature difference.

TABLE 1

Parameters* of MBWR Equation of State for Isobutane

Year Determined Parameters	1973	1977	1978
B_o	1.87890	2.02615	2.10544
A_o	37264.0	38980.2	39734.2
$C_o \times 10^{-8}$	101.413	106.581	110.858
γ	7.11486	9.21378	9.48772
b	8.58663	6.70763	6.28615
a	47990.7	38864.4	36583.9
α	4.23987	6.87727	7.64696
$c \times 10^{-8}$	406.763	328.220	342.981
$D_o \times 10^{-10}$	85.3176	147.046	152.353
$d \times 10^{-4}$	2168.63	618.303	457.448
$E_o \times 10^{-10}$	8408.60	8981.52	8098.36

* Units: psi, cubic feet, lb moles, °R

TABLE 2

Isentropic Expansion of Isobutane for 300°F Resource Cycle Conditions

		MBWR			
	OBWR	1973	1977	1978	GMBWR
T_1, °F	222	222	222	222	222
P_1, psia	300	300	300	300	300
H_1, Btu/lb	169.778	172.215	172.896	172.089	169.001
S_1, Btu/lb °F	1.1846	1.1878	1.1888	1.1878	1.1838
T_2, °F	130.97	133.39	134.054	133.118	129.82
P_2, psia	78.553	78.553	78.553	78.553	78.553
H_2, Btu/lb	146.810	148.833	149.420	148.793	146.296
S_2, Btu/lb °F	1.1846	1.1878	1.1888	1.1878	1.1838
ΔH, Btu/lb	22.968	23.382	23.476	23.296	22.705
Dev.* in ΔH, %	-1.408	-0.369	0.773	0	-2.537

*Deviation calculated as Dev. = $\left[\frac{\Delta H - \Delta H \text{ (1978 MWWR)}}{\Delta H \text{ (1978 MBWR)}}\right] \times 100\%$

TABLE 3

Isentropic Expansion of Isobutane for 350°F Resource Cycle Conditions

	OBWR	MBWR			GMBWR
		1973	1977	1978	
T_1, °F	260	260	260	260	260
P_1, psia	450	450	450	450	450
H_1, Btu/lb	172.579	175.839	177.079	175.393	168.560
S_1, Btu/lb °F	1.1804	1.1845	1.1862	1.1841	1.1754
T_2, °F	131.08	134.51	136.04	133.756	124.73
P_2, psia	86.3	86.3	86.3	86.3	86.3
H_2, Btu/lb	145.972	148.58	149.606	148.324	142.981
S_2, Btu/lb °F	1.1804	1.1845	1.1862	1.1841	1.1754
ΔH, Btu/lb	26.607	27.259	27.473	27.069	25.579
Dev.* in ΔH, %	-1.707	0.702	1.492	0	-5.504

* Deviation calculated as $\text{Dev.} = \left[\frac{\Delta H - \Delta H\ (1978\ \text{MBWR})}{\Delta H\ (1978\ \text{MBWR})}\right] \times 100\%$

TABLE 4

Isentropic Expansion of Isobutane for 400°F Resource Cycle Conditions

		MBWR			
	OBWR	1973	1977	1978	GMBWR
T_1, °F	288	288	288	288	288
P_1, psia	550	550	550	550	550
H_1, Btu/lb	179.010	181.745	183.061	181.114	174.020
S_1, Btu/lb °F	1.1858	1.1891	1.1909	1.1886	1.1798
T_2, °F	139.64	142.14	143.79	141.176	132.04
P_2, psia	89.107	89.107	89.107	89.107	89.107
H_2, Btu/lb	149.752	151.910	152.984	151.535	146.124
S_2, Btu/lb °F	1.1858	1.1891	1.1909	1.1886	1.1798
ΔH, Btu/lb	29.258	29.835	30.077	29.579	27.896
Dev.* in ΔH, %	-1.085	0.865	1.684	0	-5.690

* Deviation calculated as Dev. = $\left[\frac{\Delta H - \Delta H\ (1978\ \text{MBWR})}{\Delta H\ (1978\ \text{MBWR})}\right] \times 100\%$

TABLE 5

Isentropic Expansion of Isobutane for 500°F Resource Cycle Conditions

	OBWR	MBWR 1973	MBWR 1977	MBWR 1978	GMBWR
T_1, °F	314	314	314	314	314
P_1, psia	700	700	700	700	700
H_1, Btu/lb	178.57	180.667	180.799	179.872	176.277
S_1, Btu/lb °F	1.1817	1.1842	1.1844	1.1834	1.1795
T_2, °F	145.23	146.29	145.87	145.14	142.38
P_2, psia	106.73	106.73	106.73	106.73	106.73
H_2, Btu/lb	150.477	152.145	152.296	151.682	149.072
S_2, Btu/lb °F	1.1817	1.1842	1.1844	1.1834	1.1795
ΔH, Btu/lb	28.093	28.522	28.503	28.190	27.205
Dev.* in ΔH, %	-0.344	1.178	1.110	0	-3.494

* Deviation calculated as $\text{Dev.} = \left[\frac{\Delta H - \Delta H\ (1978\ \text{MBWR})}{\Delta H\ (1978\ \text{MBWR})}\right] \times 100\%$

TABLE 6

Comparison of Conventional Geothermal Binary Cycle Simulations for a 300°F Georesource Using Different Equation of State Parameters for Isobutane

Correlation Parameters	OBWR	MBWR			GMBWR
Process Parameters		1973	1977	1978	
Brine Heat Exchanger					
Brine Inlet T, °F	300	300	300	300	300
Outlet T, °F	178.8	176.4	179.6	179.4	182.4
W.F. Inlet T, °F	109.7	109.9	109.7	109.7	109.8
H, Btu/lb$_m$	5.74	3.86	4.86	4.84	4.00
Outlet T, °F	220	220	220	220	220
H, Btu/lb$_m$	168.43	170.92	171.63	170.80	167.62
W.F. Enthalpy Change Across the BHE, Btu/lb$_m$	162.69	167.06	166.77	165.96	163.62
W.F. Bubble point T, °F	218.0	217.1	218.3	218.7	221.2
H, Btu/lb$_m$	78.63	79.21	79.30	79.53	78.90
Turbine					
W.F. Inlet T, °F	220	220	220	220	220
P, psia	300	300	300	300	300
H, Btu/lb$_m$	168.43	170.92	171.63	170.80	167.62
S, Btu/lb$_m$ °F	1.1827	1.1859	1.1869	1.1859	1.1818
W.F. Outlet T, °F	138.2	140.6	141.3	140.0	134.0
P, psia	83.2	82.9	82.8	82.3	78.6
H, Btu/lb$_m$	149.71	151.79	152.40	151.63	148.26
S, Btu/lb$_m$ °F	1.1878	1.1911	1.1921	1.1911	1.1872
W.F. Enthalpy Change Across the Turbine, Btu/lb$_m$	18.72	19.13	19.23	19.17	19.36
Brine Flow Rate, MMlb$_m$/hr	7.5156	7.5156	7.5156	7.5156	7.5156
W.F. Flow Rate, MMlb$_m$/hr	5.6739	5.6350	5.4986	5.5353	5.4764
Gross Turbine Power, MW	31.11	31.59	30.97	31.10	31.07
Net Plant Power, MW	24.94	25.39	24.91	25.01	25.01
Plant Cost, $/kw	883	885	876	875	858
Total Cost, $/kw	1468	1460	1462	1459	1442

TABLE 7

Comparison of Conventional Geothermal Binary Cycle Simulations for a 350°F Georesource Using Different Equation of State Parameters for Isobutane

Correlation Parameters / Process Parameters	OBWR	MBWR			GMBWR
		1973	1977	1978	
Brine Heat Exchanger					
Brine Inlet T, °F	350	350	350	350	350
Outlet T, °F	204.2	188.3	206.4	202.5	185.6
W.F. Inlet T, °F	117.0	117.3	117.0	117.0	117.1
H, Btu/lb_m	10.50	8.49	9.54	9.52	8.52
Outlet T, °F	260	260	260	260	260
H, Btu/lb_m	172.58	175.84	177.08	175.39	168.56
W.F. Enthalpy Change Across the BHE, Btu/lb_m	162.08	167.35	167.54	165.87	160.04
W.F. Bubble point T, °F	258.0	257.9	258.8	258.9	260.2
H, Btu/lb_m	113.75	120.96	116.34	116.92	119.20
Turbine					
W.F. Inlet T, °F	260	260	260	260	260
P, psia	450	450	450	450	450
H, Btu/lb_m	172.58	175.84	177.08	175.39	168.56
S, Btu/lb_m °F	1.1804	1.1845	1.1862	1.1841	1.1754
W.F. Outlet T, °F	142.1	145.6	147.09	144.4	132.52
P, psia	91.8	91.5	91.4	90.8	86.7
H, Btu/lb_m	150.63	153.30	154.34	152.89	146.63
S, Btu/lb_m °F	1.1864	1.1906	1.1924	1.1903	1.1815
W.F. Enthalpy Change Across the Turbine, Btu/lb_m	21.95	22.54	22.74	22.50	21.93
Brine Flow Rate, MMlb_m/hr	5.1706	5.1706	5.1706	5.1706	5.1706
W.F. Flow Rate, MMlb_m/hr	4.7619	5.1099	4.5359	4.7055	5.4334
Gross Turbine Power, MW	30.62	33.75	30.23	31.03	34.91
Net Plant Power, MW	24.66	27.31	24.48	25.09	28.05
Plant Cost, $/kw	643	658	635	639	671
Total Cost, $/kw	1050	1026	1045	1040	1029

TABLE 8

Comparison of Conventional Geothermal Binary Cycle Simulations for a 400°F Georesource Using Different Equation of State Parameters for Isobutane

Correlation Parameters / Process Parameters	OBWR	MBWR			GMBWR
		1973	1977	1978	
Brine Heat Exchanger					
Brine Inlet T, °F	400	400	400	400	400
Outlet T, °F	205.9	202.2	206.9	204.7	201.9
W.F. Inlet T, °F	120.9	121.2	120.9	120.9	121.1
H, Btu/lb_m	13.08	11.00	12.08	12.07	11.02
Outlet T, °F	288	288	288	288	288
H, Btu/lb_m	179.01	181.74	183.06	188.11	174.02
W.F. Enthalpy Change Across the BHE, Btu/lb_m	165.93	170.74	170.98	169.04	163.00
W.F. Bubble point T, °F	-	-	-	-	-
H, Btu/lb_m	-	-	-	-	-
Turbine					
W.F. Inlet T, °F	288	288	288	288	288
P, psia	550	550	550	550	550
H, Btu/lb_m	179.01	181.74	183.06	181.11	174.02
S, Btu/lb_m °F	1.1858	1.1891	1.1909	1.1886	1.1798
W.F. Outlet T, °F	151.1	153.6	155.3	152.2	140.2
P, psia	94.3	94.0	93.9	93.2	89.1
H, Btu/lb_m	154.72	156.93	158.01	156.38	150.03
S, Btu/lb_m °F	1.1923	1.1958	1.1976	1.1952	1.1678
W.F. Enthalpy Change Across the Turbine, Btu/lb_m	24.29	24.81	25.05	24.73	23.99
Brine Flow Rate, MMlb_m/hr	3.5725	3.5725	3.5725	3.5725	3.5725
W.F. Flow Rate, MMlb_m/hr	4.3056	4.2642	4.1561	4.2522	4.4739
Gross Turbine Power, MW	30.64	31.01	30.50	30.81	31.45
Net Plant Power, MW	24.48	24.89	24.50	24.70	25.00
Plant Cost, $/kw	619	618	613	616	620
Total Cost, $/kw	902	896	897	897	897

TABLE 9

Bubble Point Predictions at 454.9 psia Using Different Equation of State Parameters for Isobutane

		MBWR		
	OBWR	1973	1977	1978
Temperature, °F	258.0	257.9	258.8	258.9
Pressure, psia	454.9	454.9	454.9	454.9
Enthalpy, Btu/lb	113.75	120.96	116.34	116.92
Enthalpy at Evaporator Outlet, Btu/lb	172.58	175.84	177.08	175.39
Heat of Vaporization, Btu/lb	58.83	54.88	60.74	58.47
Brine Exit Temp., °F	204.2	188.3	206.4	202.5

TABLE 10

Comparison of Conventional Binary Cycle Simulations with Fixed Heat Transfer Area Using Different Equation of State Parameters for Isobutane

Correlation Parameters / Process Parameters	OBWR	MBWR			GMBWR
		1973	1977	1978	
Brine Flow Rate, MMlb/hr	7.5156	7.5156	7.5156	7.5156	7.5156
Brine Inlet Temp., °F	300	300	300	300	300
Brine Exit Temp., °F	178.0	176.4	179.1	178.8	180.9
Turbine					
W.F. Inlet T, °F	220	220	220	220	220
W.F. Inlet P, psia	300	300	300	300	300
W.F. Enthalpy Change, BTU/lb	18.73	19.13	19.31	19.25	19.45
Gross Power, MW	31.33	31.59	31.20	31.32	31.54
Diameter, ft	4.868	4.854	4.815	4.838	4.887
Brine Heat Exchanger					
Pinch Point ΔT, °F	14.2	15	14.4	14.6	14.1
Shell Side ΔP, psia	15.4	15.8	14.6	14.7	14.3
Overall Heat Trans. Coeff., BTU/hr-ft^2-°F	221	228	220	220	221
LMTD, °F	36.3	35.9	36.3	36.3	35.6
Total Duty, 10^8 BTU/hr	9.2931	9.4138	9.2109	9.2278	9.0689
Total Heat Trans. Area, ft^2	115,681	115,307	115,344	115,324	115,345
Condenser					
Pinch Point ΔT, °F	10	10	9.6	9.6	9.6
Shell Side ΔP, psia	0.56	0.55	0.57	0.57	0.56
Overall Heat Trans. Coeff., BTU/hr-ft^2-°F	160	161	161	161	162
LMTD, °F	18.9	19.1	18.7	18.7	18.2
Total Duty, 10^8 BTU/hr	8.3113	8.4206	8.2308	8.2441	8.0796
Total Heat Trans. Area, ft^2	274,163	273,879	273,904	274,080	273,515
Cooling Water Flow Rate, MMlb/hr	46.136	46.742	45.687	45.763	44.850
Working Fluid Flow Rate, MMlb/hr	5.712	5.635	5.516	5.554	5.535
Cycle Pump Power, MW	2.58	2.49	2.49	2.50	2.56
Cooling Water Pump Power, MW	2.30	2.32	2.29	2.29	2.26
Brine Pump Power, MW	0.51	0.51	0.51	0.51	0.51
Cooling Tower Fan Power, MW	0.86	0.88	0.86	0.86	0.84
Net Thermo. Cycle Power, MW	28.76	29.10	28.72	28.82	28.98
Net Thermo. Efficiency, %	10.56	10.55	10.64	10.66	10.91
Net Plant Power, MW	25.08	25.39	25.06	25.16	25.36

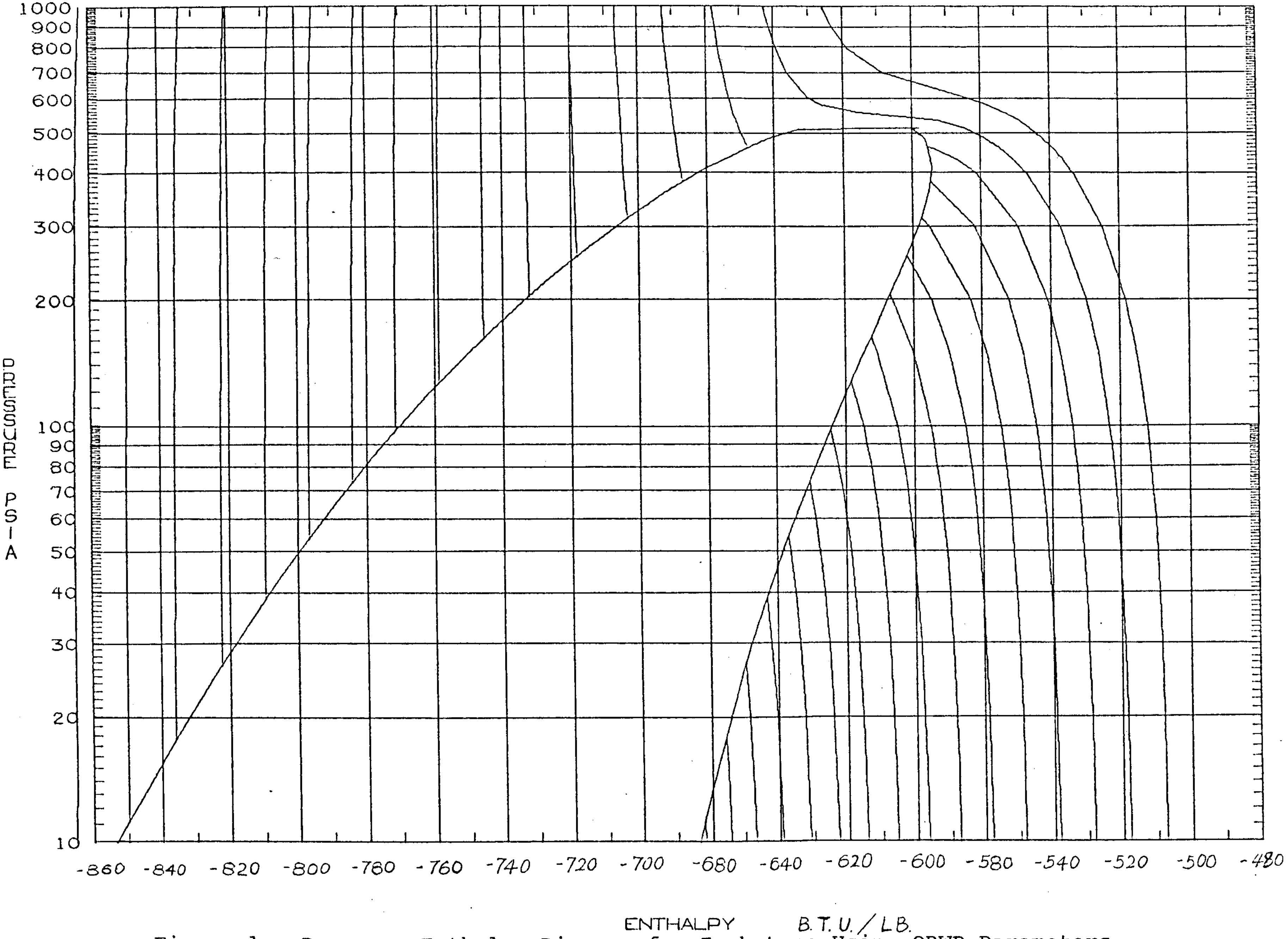

Figure 1. Pressure-Enthalpy Diagram for Isobutane Using OBWR Parameters

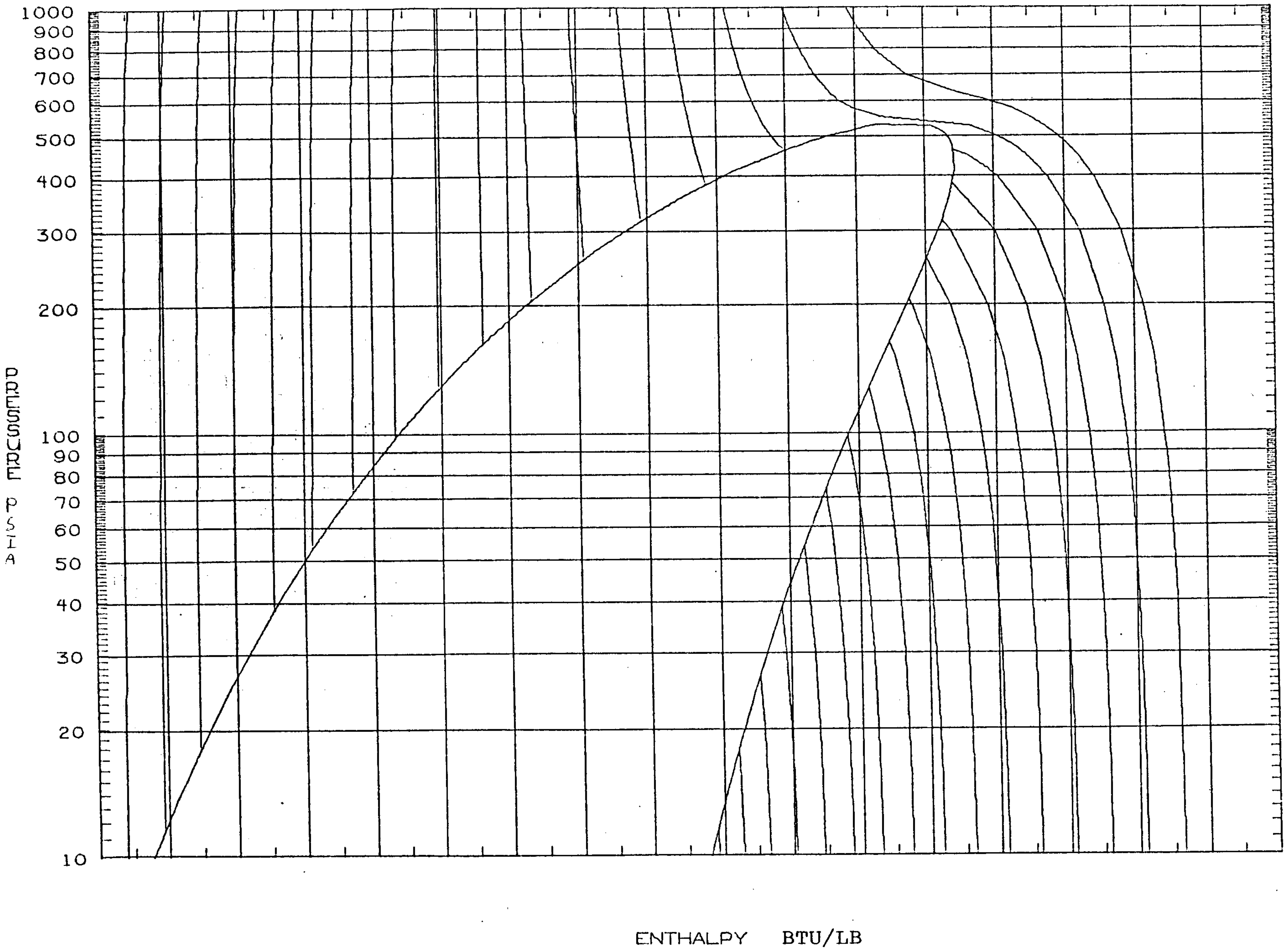

Figure 2. Pressure-Enthalpy Diagram for Isobutane Using 1973 MBWR Parameters

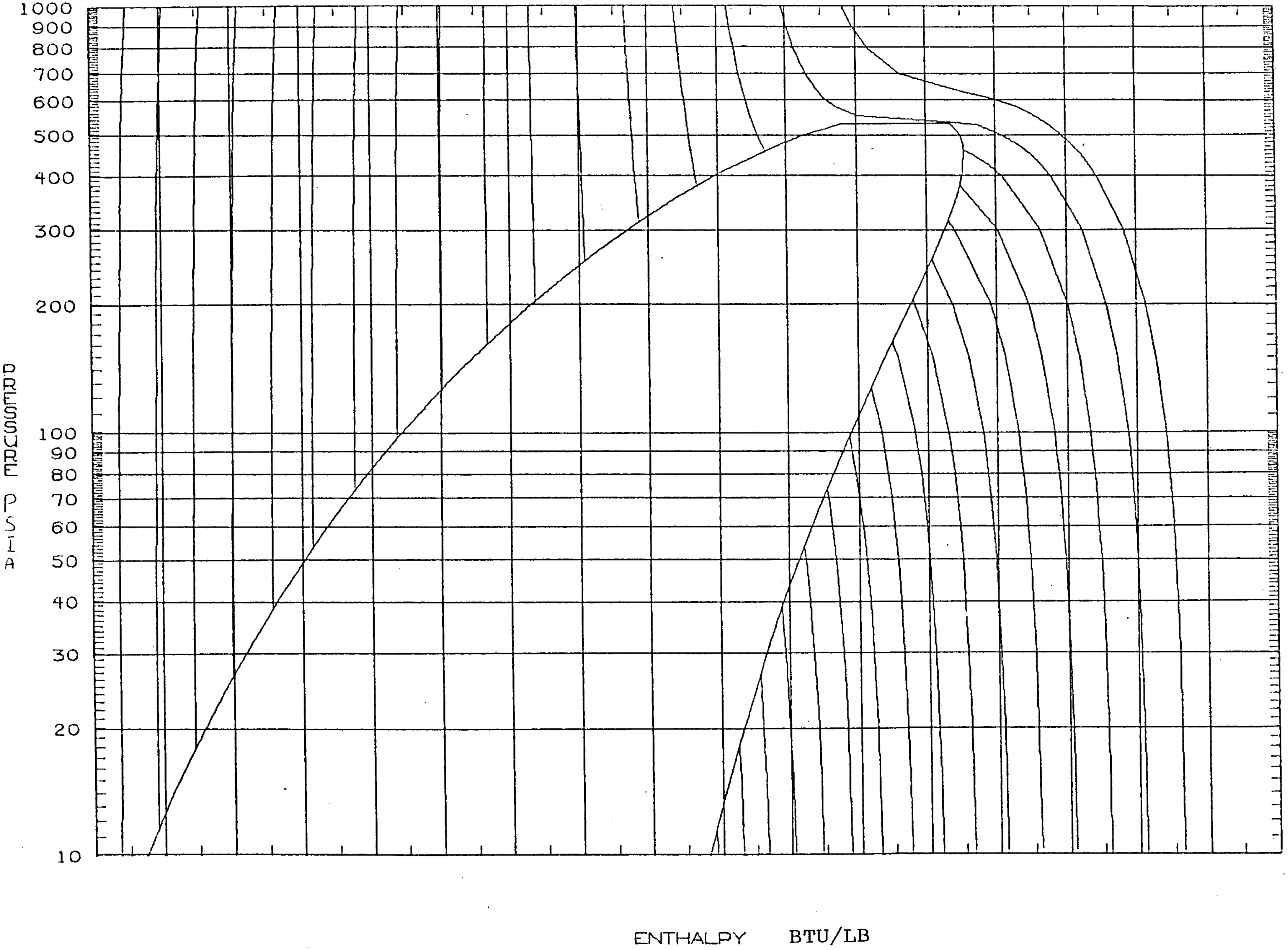

Figure 3. Pressure-Enthalpy Diagram for Isobutane Using 1977 MBWR Parameters

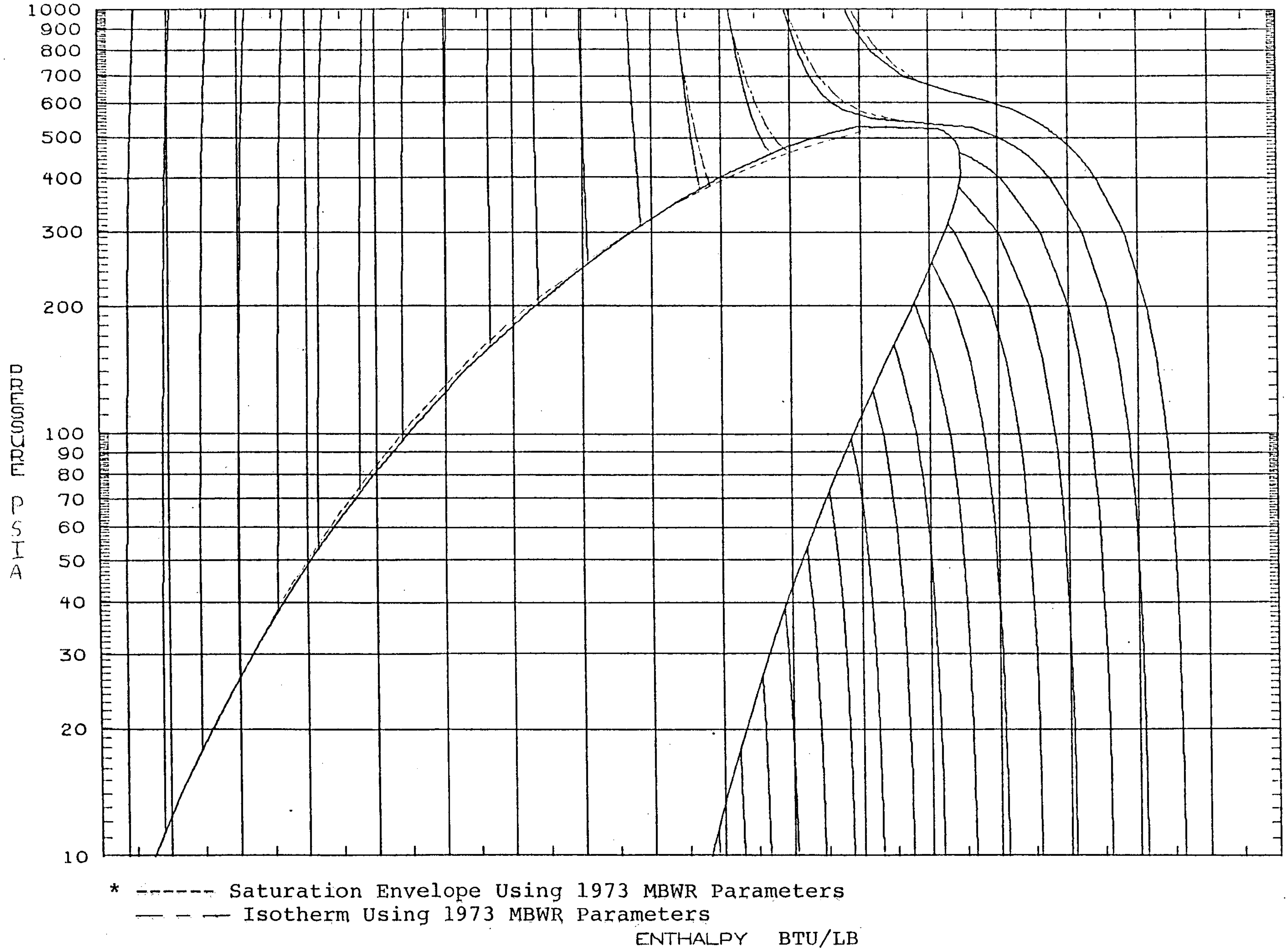

Figure 4. Pressure-Enthalpy Diagram for Isobutane Using 1978 MBWR Parameters

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