DOE/ET 1/1

DOE/ET/28443--T18

DAEDALEAN ASSOCIATES, Incorporated

DE82 004326

FINAL REPORT

MASTER

GLEF DIRECT-CONTACT
HEAT-EXCHANGER PROGRAM

Submitted to:

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AC03-76 ET 28443

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DAI Technical Report
PDV-7902-002-TR

March 1980

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978

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ACKNOWLEDGEMENTS

The authors wish to acknowledge the efforts of Mr. Steven Smith, Mr. Ronald Rowan, and Mr. Russel Beck in the construction and operation of the direct contact heat exchange test loop used in this program.

ABSTRACT

A number of flash-binary cycles for the Salton Sea Known Geothermal Area have been evaluated leading to the identification of the optimum one. The brine coming from the well is separated in a flash tank producing a liquid and a vapor stream. The liquid is used in a direct contact preheater to bring the working fluid (isopentane) to the saturation point. The isopentane is then vaporized in a shell and tube heat exchanger by use of the flashed steam. The isopentane vapor is expanded in a turbine driving the generator, condensed in a surface condenser and finally returned to the preheater by the main feed pump.

A review of the theoretical correlations available for the design of direct contact heat exchangers has been carried out as well as the design and fabrication of a test facility to evaluate the performance of direct contact heat exchangers under various conditions and to verify the cycle analysis.

The results of the test data support the assumptions made in the system analysis and in particular, the terminal temperature differences established in the heat balance have been achieved. The column length calculation procedure has been tested and found satisfactory under the test conditions.

TABLE OF CONTENTS

	<u>Page</u>
ACKNOWLEDGEMENTS	i
ABSTRACT	ii
LIST OF TABLES AND FIGURES	iii
1.0 INTRODUCTION	1
1.1 Program Objectives and Scope	2
2.0 SYSTEM ANALYSIS	7
2.1 Design Parameters	7
2.1.1 Brine	7
2.1.2 Cooling Water System	7
2.1.3 Electrical	8
2.1.4 Temperatures	8
2.1.5 Pressures	8
2.2 Working Fluid Selection	8
2.3 Flash Tank	10
2.4 Evaluation of Alternate Cycles	12
2.4.1 Binary ("Liquid/Liquid") Cycle With Steam Heat Exchanger	13
2.4.2 Direct Contact Cycle	14
2.4.3 Direct Contact Cycle With Hydrocarbon Superheater	17
2.4.4 Direct Contact Cycle With Steam Turbine	17
2.5 Further Improvements of the Binary ("Liquid/Liquid") Configuration	18

	<u>Page</u>
2.6 Concluding Remarks Investigating the Optimum Cycle	19
3.0 THEORETICAL DESIGN OF A DIRECT CONTACT HEAT EXCHANGER	22
3.1 Theoretical Correlations	22
3.1.1 Evaluation of the Column Diameter	23
3.1.2 Evaluation of the Column Length	27
3.1.3 Notation	31
4.0 DESIGN OF THE TEST FACILITY	33
4.1 Introduction	33
4.2 System Description	33
4.2.1 Cooling Water Subsystem	33
4.2.2 Hot Water (Brine) Subsystem	34
4.2.3 Working Fluid Subsystem	35
4.2.4 Makeup Water Subsystem	36
4.2.5 Noncondensibles Subsystem	36
4.2.6 Direct Contact Heat Exchanger	36
5.0 SYSTEM ANALYSIS VERIFICATION	40
5.1 Introduction	40
5.2 Preliminary Design	40
5.3 Test Plan	41
6.0 DATA COLLECTION AND ANALYSIS	44
6.1 Experimental Setup	44

	<u>Page</u>
6.2 Data Collection and Analysis	44
7.0 CONCLUSIONS AND RECOMMENDATIONS	46
REFERENCES	50
APPENDIX A	
Computer Program for the Design of Direct Contact Preheaters	A-1
APPENDIX B	
Engineering Specifications for the Main Equipment Utilized in the Test Facility	B-1
APPENDIX C	
Cycle Evaluation Procedure	C-1
APPENDIX D	
The Design of a Direct Contact Preheater for a Unit Utilizing the Full Flow of One Well at the Niland Reservoir	D-1
Indications on the Direct Contact Preheater to be Installed in the 50 MWe Plant	D-3

LIST OF TABLES AND FIGURES

- TABLE 2.2.1 - COMPARISON OF CYCLES USING ISOPENTANE AND ISOBUTANE
- TABLE 2.4.2 - PRELIMINARY EVALUATION OF THE DIRECT CONTACT CYCLE
- TABLE 6.2.1 - DIRECT CONTACT PREHEATER TEST RESULTS
- FIGURE 2.2.1 - ISOBUTANE CYCLE
- FIGURE 2.2.2 - ISOPENTANE CYCLE
- FIGURE 2.4.1.1 - CYCLE FLOW DIAGRAM
- FIGURE 2.4.1.2 - HEAT BALANCE NO. 1 (BOILING TEMP 325°F)
- FIGURE 2.4.1.3 - STEAM CONDENSATION CURVE FOR HEAT BALANCE #1
- FIGURE 2.4.1.4 - HEAT BALANCE NO. 2 (BOILING TEMP 320°F)
- FIGURE 2.4.1.5 - STEAM CONDENSATION CURVE FOR HEAT BALANCE #2
- FIGURE 2.4.1.6 - HEAT BALANCE NO. 3 (BOILING TEMP 315°F)
- FIGURE 2.4.1.7 - STEAM CONDENSATION CURVE FOR HEAT BALANCE #3
- FIGURE 2.4.1.8 - HEAT BALANCE NO. 4 (BOILING TEMP 310°F)
- FIGURE 2.4.1.9 - STEAM CONDENSATION CURVE FOR HEAT BALANCE #4
- FIGURE 2.4.1.10 - BRINE RATE AND AUXILIARY POWER VERSUS BOILING TEMPERATURE FOR THE BINARY ("LIQUID/LIQUID") CYCLE
- FIGURE 2.4.2.1 - CYCLE FLOW DIAGRAM
- FIGURE 2.4.2.2 - HEAT BALANCE NO. 5 (BOILING TEMP 240°F)
- FIGURE 2.4.2.3 - HEAT BALANCE NO. 6 (BOILING TEMP 250°F)

- FIGURE 2.4.2.4 - HEAT BALANCE NO. 7 (BOILING TEMP 260°F)
- FIGURE 2.4.2.5 - HEAT BALANCE NO. 8 (BOILING TEMP 270°F)
- FIGURE 2.4.2.6 - BRINE RATE VERSUS BOILING TEMPERATURE FOR THE DIRECT CONTACT CYCLE
- FIGURE 2.4.3.1 - CYCLE FLOW DIAGRAM
- FIGURE 2.4.3.2 - CYCLES WITH AND WITHOUT SUPERHEAT
- FIGURE 2.4.3.3 - HEAT BALANCE NO. 9
- FIGURE 2.4.4.1 - CYCLE FLOW DIAGRAM
- FIGURE 2.4.4.2 - HEAT BALANCE NO. 10 (BOILING TEMP 240°F)
- FIGURE 2.4.4.3 - HEAT BALANCE NO. 11 (BOILING TEMP 250°F)
- FIGURE 2.4.4.4 - HEAT BALANCE NO. 12 (BOILING TEMP 260°F)
- FIGURE 2.4.4.5 - HEAT BALANCE NO. 13 (BOILING TEMP 270°F)
- FIGURE 2.4.4.6 - BRINE RATE AND AUXILIARY POWER VERSUS BOILING TEMPERATURE FOR THE DIRECT CONTACT CYCLE WITH STEAM TURBINE
- FIGURE 2.5.1 - HEAT BALANCE DIAGRAM FOR THE LIQUID/LIQUID PREHEATER CYCLE
- FIGURE 3.1.1.1 - COLUMN DIAMETER VERSUS DROPLET SIZE ACCORDING TO THE SIDEMAN AND SAKIADIS-JOHNSON CORRELATIONS FOR ONE WELL FLOW
- FIGURE 3.1.1.2 - COLUMN DIAMETER VERSUS DROPLET SIZE ACCORDING TO THE SIDEMAN AND SAKIADIS-JOHNSON CORRELATIONS FOR 50 MWe PLANT (6 UNITS REQUIRED)
- FIGURE 3.1.2.1 - COLUMN LENGTH VERSUS DROPLET SIZE ACCORDING TO THE SIDEMAN AND LETHAN-KEHAT CORRELATIONS
- FIGURE 4.1.1 - VIEW OF THE DIRECT CONTACT HEAT EXCHANGER TEST LOOP
- FIGURE 4.1.2 - PARTICULAR OF THE CONTROL PANEL

- FIGURE 4.2.1 - GLEF TEST LOOP FLOW DIAGRAM
- FIGURE 5.2.1 - PRELIMINARY DESIGN OF THE DIRECT CONTACT
PREHEATER FOR A 500 KW PROTOTYPE
- FIGURE 5.3.1 - TEMPERATURE PROFILES OF ISOPENTANE AND HOT
WATER IN THE TEST LOOP
- FIGURE 6.1.1 - TEMPERATURE PROFILES OF ISOPENTANE AND HOT
WATER IN THE TEST LOOP (MODIFIED)
- FIGURE 7.1 - DIRECT CONTACT PREHEATER
- FIGURE B.1 - ISOPENTANE DISTRIBUTOR
- FIGURE D.1 - HEAT BALANCE DIAGRAM FOR ONE WELL PLANT
- FIGURE D.2 - COLUMN DIAMETER VERSUS DROPLET SIZE ACCORD-
ING TO THE SIDEMAN AND SAKIADIS-JOHNSON
CORRELATIONS FOR ONE WELL FLOW
- FIGURE D.3 - DIRECT CONTACT PREHEATER

GLEF DIRECT CONTACT
HEAT EXCHANGER PROGRAM

1.0 INTRODUCTION

The successful implementation of the geothermal energy resources in the United States depends upon the engineering development of power production systems capable of effectively and efficiently producing useful work from this source of energy. The Salton Sea Known Geothermal Resource Area (KGRA) is relatively rich with regard to geothermal energy. Therefore, much benefit can be gained from the extraction of the maximum practical heat from this geothermal energy source.

The feasibility of energy extraction from the Salton Sea KGRA was investigated by The Ben Holt Company and Bechtel National, Inc., and was reported to the San Diego Gas and Electric Company in the report titled, "Feasibility and Risk Study of a Geothermal Power Plant at the Salton Sea KGRA." The conclusions reported therein support the utilization of a two-stage flash steam process using unmodified brine and three 50 percent capacity trains of flash vessels. This steam supply system is sized to enable the accommodation of a turbine-generator capable of supplying approximately 57,000 KW gross electrical power output. No minimum injection temperature limit is imposed.

The Ben Holt Company has also conducted an independent study entitled "Feasibility Study: Direct Contact Heat Exchange Process." This report identifies and addresses several direct contact heat exchange concepts.

Different approaches are currently being investigated through sponsored R & D programs to develop the technical feasibility for energy recovery from hot geothermal brines. The primary schemes include: (1) the flash system; (2) the flash-binary cycle; and (3) the binary cycle system using both conventional and direct contact heat exchangers. The flash-binary cycle concept using direct contact heat exchangers offers unique advantages for recovering energy from geothermal sources which are subject to scale formation in the heat exchangers. The increased operation and maintenance costs associated with the scale formation and corrosion of conventional heat exchangers is eliminated through the use of direct contact heat exchangers. Because direct contact heat exchangers are smaller and significantly less costly to manufacture than conventional heat exchangers, their capital cost is greatly reduced.

1.1 Program Objectives and Scope

The objective of this research and development program is to provide the preliminary engineering data for the future evaluation of the technical and economic feasibility

of a flash-binary cycle at the Niland Reservoir using a direct contact heat exchanger.

The program scope consists of performing a system analysis of the primary flow characteristics and establishing a model test loop from which engineering data could be generated and evaluated. The work effort is divided into three specific tasks. The first task, (TASK I) addresses the analytical evaluation for the brine in order to establish an optimum operating cycle for the system. The second task, (TASK II) addresses the generation and evaluation of analysis data from the model test loop. This task is divided into three separate subtasks. Subtask II-A is the design and fabrication of the model test loop; Subtask II-B, the operation of the test loop in order to confirm the system analysis; Subtask II-C, the evaluation of the data generated from the test loop. The third task, (TASK III) of the program is the preparation of the final technical report.

The program approach is to generate test loop data in order to confirm the system analysis and to permit the evaluation of operational parameters that could be factored into a larger system design.

The specific tasks that are necessary for the accomplishment of the program are listed below as follows:

TASK ISystem Analysis

In order that the most efficient direct contact heat exchanger flash-binary cycle could be developed, it is necessary to perform system calculations for the determination of the optimum cycle. Preliminary indications are that a single stage flash system providing input to the direct contact boiler will be the optimum system for use. This concept provides the basic configuration for the system analysis.

The primary parameters to be used in the system analysis, including the brine enthalpy and brine flowrate, are to be varied as a function of temperature and flowrate in the calculations. The analysis for the Niland site system is to be performed based on the assumption that the brine is free flowing from the wells to the flash tank. Cycle optimization and system analysis are to be performed considering the use of isopentane as the secondary system working fluid. Other choices, such as isobutane are also to be explored. The system analysis

effort is to be performed as an independent effort in order to evaluate the model test loop data.

TASK II

Model Test Loop Evaluation

The system analysis performed during the TASK I work effort is to be verified using a model test loop fabricated at the Springlake Research Center. For the performance of this test work, the following subtask efforts will be required.

TASK II-A

Modification of Model Test Loop

In order to meet the wide range of brine input requirements with regard to temperature and pressure, it will be necessary to provide a simulated brine input system that will have capability of supplying hot water at variable temperatures. This simulated brine supply system should also have the capability of injecting noncondensable gases (CO_2) into the direct contact boiler.

TASK II-B

Model Test Performance

In order to verify the system analysis calculations performed in the TASK I work effort,

the calculated system parameters are to be established for the model test loop and the specific cycle operation is to be evaluated. The model test loop data will provide a means of evaluating the TASK I calculations.

TASK II-CTest Loop Data Evaluation

An evaluation of the data collected during the Subtask II-B is to be performed to obtain an experimental configuration of the assumptions taken in the system analysis.

2.0 SYSTEM ANALYSIS

2.1 Design Parameters

The following are the design parameters on which the system analysis has been performed, established during meetings with The Ben Holt Company and San Diego Gas and Electric.

2.1.1 Brine

- Well production: starts 400,000 lb/hr, decreasing to 300,000 lb/hr
- Design downhole temperature: 482°F
- Total dissolved salts: 20% (essentially NaCl)
- Enthalpy of saturated liquid brine: 389.4 Btu/lb at 482°F
- Wellhead pressure: 150 psia
- Noncondensibles: 0.5% by weight (average MW 42)
- Brine is not pumped
- Brine properties are based on the Lawrence Livermore Laboratory report UCID-17406 "Calculation of Brine Properties" by Gerald L. Dittman

2.1.2 Cooling Water System

- Design wet bulb temperature: 77°F
- Mechanically induced draft cooling tower
- Approach: 10°F
- Range: 18°F
- CW circulating pump head: 26 psi

2.1.3 Electrical

- Net power output: 50 MWe
- Turbine-generator efficiency: 80%
- Brine pump efficiency: 70%
- HC pump efficiency: 70%
- Cooling water pump efficiency: 75%

2.1.4 Temperatures

- Condensing temperature: 110°F
- Contactor approach*: 5°F (if controlling for process)
- Vaporizer approach: 20°F

2.1.5 Pressures

- Pressure drop HC pump to contactor: 5 psi
- Pressure drop contactor: negligible
- Pressure drop brine pump to contactor: 5 psi
- Pressure drop condenser (HC side): negligible
- Pressure drop vaporizer (both sides): negligible

2.2 Working Fluid Selection

Based on their low solubility in geothermal brines, two working fluids were considered those being isopentane and isobutane.

One important difference between isobutane and isopentane is that isobutane has a critical temperature that is lower than the isopentane (275°F versus 370°F).

* Approach is defined as the minimum terminal difference on either end of a heat exchanger.

Cursory evaluations of a number of cycles have been carried out for both fluids (for the isobutane the evaluation included some supercritical cycles).

The result of the investigation shows that the optimum operating point is close to the critical point for both fluids. The result of the calculations for the two fluids is summarized in the following Table 2.2.1.

TABLE 2.2.1 - COMPARISON OF CYCLES USING
ISOPENTANE AND ISOBUTANE

	<u>Isopentane</u>	<u>Isobutane</u>
Brine rate (gross) (lb/KWh)	83	82
Boiler pressure (psia)	330	480
Auxiliary power (%)	10	19

Figures 2.2.1 and 2.2.2 show the two optimum cycles for each fluid.

There is no practical difference in the brine rate, but, on the contrary, the boiler operating pressure and the auxiliaries consumption are much in favor of isopentane as a working fluid.

After the completion of the cycle analysis, a check was made on the impact the cycle design parameters would have had on the mechanical design.

As far as the isopentane cycle is concerned, it was determined that at the optimum operating conditions the

rating for flanges, flanged fittings and valves should be the 300 lb class. It is not possible, to utilize the 150 lb class because, to achieve this goal, one would have to decrease the operating pressure so that the cycle efficiency would have been lowered considerably.

As far as the isobutane cycle, all system hardware would be at least 300 lb class and it is possible that, for some parts of the plant, the 600 lb class would have to be adopted.

If the various pieces of equipment subject to internal pressure are considered, it can be seen that in the case of isobutane the wall thicknesses will be approximately 1.5 times that required for the isopentane cycle (the weights will increase accordingly).

For this reason, and those derived from Table 2.2.1, isopentane was selected as the working fluid for the geothermal facility under consideration.

The thermodynamic properties of isopentane at the saturation states have been obtained from the "Journal of Chemical and Engineering Data" Volume 22, No. 1, 1977.

2.3 Flash Tank

The main reason to install a flash tank downstream from the well head is to accomplish the separation of the liquid brine from the steam that is formed due to the partial flash

of the brine along its path from the bottom of the well to the surface.

The operating pressure of the flash tank has been assumed equal to 150 psia. The noncondensable gases present in the incoming stream are about 4.3 percent by weight of the flashed stream. These values are based on figures by The Ben Holt Company. The outcoming streams have the following characteristics (per pound of incoming brine):

Steam Pressure: 150 psia

 Temperature: 357°F

 Flow: 0.122 lb/hr

 N. C.: 0.0048 lb/hr (MW $\approx$ 44)

Liquid Brine Pressure: 150 psia

 Temperature: 372°F

 Flow: 0.888 lb/hr

 TDS: 20%

The possibility of varying the pressure of the flash tank has been investigated to see if the initial assumption (150 psi) was the best choice. An increase in the pressure would be favorable from a thermodynamic efficiency standpoint because doing so would achieve a higher liquid brine temperature; but at the same time would introduce

some inconveniences, such as higher pressure would be maintained in the entire cycle. On the other hand, a decreasing in the flash tank pressure would have the effect of lowering the liquid brine temperature, consequently decreasing the thermodynamic efficiency and increasing the steam flow rate in such a way that it would have been difficult to make use of it without a further decrease in the cycle efficiency. Therefore, the original assumption of 150 psi was correct.

2.4 Evaluation of Alternate Cycles

Four different cycle configurations have been evaluated for the GLEF at the Niland site:

1. A cycle that uses both the streams coming out from the flash tank; the liquid brine preheats the working fluid to the saturation point and the steam boils it.
2. A cycle using only the liquid brine from the flash tank; both the preheating and the boiling of the hydrocarbon are accomplished in the direct contact heat exchanger.
3. A cycle that uses both the streams coming from the flash tank; the preheating and the boiling are both accomplished in the direct contact heat exchanger and the vapor stream out of the flash tank is used to superheat the hydrocarbon vapor.

4. A cycle that uses both the streams coming from the flash tank; the liquid brine is used in a direct contact bottoming cycle and the steam is used in a conventional steam turbine.

2.4.1 Binary ("Liquid/Liquid") Cycle With Steam Heat Exchanger

Figure 2.4.1.1 shows a schematic flow diagram for this configuration. The brine coming from the well is separated in a flash tank, in a liquid stream, and in a vapor stream. The liquid is used in a direct contact preheater to heat up the isopentane working temperature up to the saturation point.

The isopentane is boiled in a tube and shell heat exchanger by use of the flashed steam. The isopentane vapor is expanded in a turbine driving the generator and afterward it is condensed in a surface condenser. The condensate isopentane is finally sent to the preheater by the main feed-pump.

To identify the set of parameters leading to the most efficient cycle a number of heat balances have been carried out with the following assumptions:

1. Isentropic expansion in the turbine.
2. System efficiency equal to 100% (i.e. no auxiliary power)

With this evaluation, it was found that the most attractive cycles employed boiling temperatures between 310°F and 330°F. In this range, four detailed heat balances have been carried out leading to the result that the optimum boiling temperature for the isopentane is 320°F. Four heat balance diagrams are attached for boiling temperatures of 310°, 315°, 320°, 325°F as well as the steam condensation curve for each case (Figures 2.4.1.2 through 2.4.1.9). Figure 2.4.1.10 shows a plot of the brine rate, and auxiliaries consumption versus the isopentane boiling temperature.

The heat balance calculations have been performed with the following assumptions concerning the auxiliary equipment:

1. Circulating water pump efficiency: 75%
2. Brine pump efficiency: 70%
3. Isopentane pump efficiency: 70%
4. Cooling tower and other auxiliaries: 2% of the net output.

2.4.2 Direct Contact Cycle

This alternate has been considered because of the high pressures and high cost involved in the previous cycle configuration. The cycle is simpler and involves a smaller number of equipment subsystems. Figure 2.4.2.1 shows a schematic flow diagram for this configuration.

Both the preheating and the boiling of the isopentane is accomplished in the same direct contact heat exchanger.

Since in this case, due attention should be given to the contribution of the flashed water to the total pressure of the boiler, the cycle concept is entirely different from that of the first alternate. The steam present in the hydrocarbon vapor coming out from the boiler plays a significant role in the work given up along the expansion in the turbine.

The optimization process has been carried out, in this case, in two steps. The first step consisted of a scanning of a wide range of temperatures making the following assumptions:

1. Taking into account the contribution of the brine vapor pressure to the total pressure of the boiler.
2. Assuming an isentropic expansion in the turbine.
3. Assuming a system efficiency equal to 100% (i.e. no auxiliary power).

These calculations have been carried out considering a boiling temperature between 240°F and 320°F. This first evaluation led to the conclusion that the optimum value should have been found in the range of 250° - 270°F as shown in the following table 2.4.2.

TABLE 2.4.2 - PRELIMINARY EVALUATION OF
THE DIRECT CONTACT CYCLE

<u>Boiling Temperature (°F)</u>	<u>Specific Output</u> <u>(Btu/lb of isopentane)</u>
240°	32.4
260°	36.6
280°	36.4
300°	32.1
320°	27.9

The second step consisted of the evaluation of a detailed heat balance for boiling temperatures of 240°F, 250°F, 260°F, 270°F; the respective heat balance diagrams are shown in Figures 2.4.2.2 through 2.4.2.5.

The heat balance calculations have been performed with the following assumptions concerning the auxiliary equipment.

1. Circulating water pump efficiency: 75%
2. Brine pump efficiency: 70%
3. Isopentane pump efficiency: 70%
4. Cooling tower and other auxiliaries: 2% of the net output

If Figure 2.4.2.6 the brine rate and the auxiliaries consumption are plotted versus the boiling temperature.

The minimum brine rate (i.e. the optimum cycle) has been found at 250°F boiling temperature.

2.4.3 Direct Contact Cycle With Hydrocarbon Superheater

Figure 2.4.3.1 shows a schematic flow diagram for this configuration. The only difference with respect to the arrangement described in paragraph 2.4.2 is that in this case we make use of the flashed steam to superheat the hydrocarbon (isopentane) vapor. Doing so, we obtain a small decrease in the brine rate (185 lb/KWh versus 187 lb/KWh). The cost of a shell and tube superheater, however, offsets any advantage that this configuration may give. Furthermore, only 20 percent of the available steam is condensed and the problem of making use of the rest is the same as in section 2.4.2.

Figure 2.4.3.2 shows this cycle superposed on the direct contact cycle, while Figure 2.4.3.3 is a heat balance diagram for this case.

2.4.4 Direct Contact Cycle With Steam Turbine

Another way to use the flashed steam is the traditional use of it in a steam turbine. This way a traditional steam cycle is combined with a hydrocarbon bottoming cycle obtaining a good efficiency. Figure 2.4.4.1 shows a schematic flow diagram for this configuration. The optimum operating point for this cycle is almost the same as that of the alternate described in section 2.4.2 with far better brine

rate. The best value is about 107.1 lb/KWh and the corresponding boiling temperature and pressure are 250°F and 191 psia.

The Figures 2.4.4.2 through 2.4.4.5 show the heat balance diagrams for boiling temperature ranging between 240°F and 270°F.

These are the assumptions used for the heat balance calculations:

1. Circulating water pump efficiency: 75%
2. Brine pump efficiency: 70%
3. Isopentane pump efficiency: 70%
4. Cooling water and other auxiliaries: 2% of the total net output

Figure 2.4.4.6 shows a plot of the brine rate and auxiliaries consumption for this cycle versus the isopentane boiling temperature.

2.5 Further Improvements of the Binary ("Liquid/Liquid") Configuration

The cycle described in section 2.4.1 has been modified mainly to accomplish the cooling of the hot stream coming out from the isopentane boiler. In the previous arrangement this stream came out at 340°F and it was not possible to use it in practice. Then it was arranged that a hydrocarbon preheater, between the main feedpump and the direct contact heat exchanger, was used in order to provide an adequate cooling of the hot stream. Using this heat exchanger, fresh water

is obtained out of the preheater that eventually could be used as makeup water for the cooling tower.

The noncondensable gases are extracted at the end of the hydrocarbon boiler and sent to the appropriate treatment.

The use of this new heat exchanger practically does not affect the efficiency of the cycle (the only difference is a small variation in the feedpump power) allowing a higher reinjection temperature (hence a smaller exploitation of the reservoir) and decreasing the chances of precipitation of the salts present in the brine (hence less scaling is expected in the reinjection pipes).

Figure 2.5.1 shows the heat balance diagram with the addition of the hydrocarbon preheater.

The assumptions concerning the auxiliaries that have been used in this case are the same as in section 2.4.1.

2.6 Concluding Remarks Investigating the Optimum Cycle

All the cycles described in the preceeding sections make use of direct contact heat exchangers and are developed along two different paths:

1. The use of the direct contact heat exchanger as a preheater in which the liquid brine and the isopentane are both maintained in the liquid phase.
2. The use of a direct contact heat exchanger in which both the preheating and the boiling of the isopentane are accomplished.

The cycles that use a direct contact heat exchanger as a preheater and boiler (described in sections 2.4.2 and 2.4.3) have the advantage of a lower working pressure and do not require a separate boiler. On the other hand, there is no possible use, within the cycle, of the steam coming out of the flash tank. In these cases, the only possible use of the steam is to produce mechanical work in a steam turbine (as described in section 2.4.4). In this way, the result is achieved of making use of the available steam (and of its enthalpic content) thus improving remarkably the total cycle efficiency (from 12 percent to 21 percent). The price paid for this improvement is a greater complexity of the cycle, such as: two separate turbines are required, two condensers, and more complicated instrumentation.

The arrangements with a direct contact preheater and a separate boiler have the intrinsic advantage of making use of the available steam within the cycle without equipment complications and have also a greater efficiency compared to the other cycles. In the arrangement described in section 2.5 is obtained the result of producing a substantial amount of fresh water that provides most of the makeup for the cooling tower.

Therefore, it can be concluded that the arrangement described in section 2.5 is the best one among the cycles that have

been evaluated and thus is the one proposed for the Niland geothermal facility.

3.0 THEORETICAL DESIGN OF A DIRECT CONTACT HEAT EXCHANGER

The heat exchange equipment of a geothermal cycle can be of the direct contact type provided the working fluid is immiscible in water. Direct contact heat exchangers can be of many different kinds, among which are spray towers, perforated plate towers, and packed towers. For the conditions that exist at the Niland site (high quantity of dissolved solids), the use of perforated or packed towers is not advisable because of scaling buildup. A good candidate for this application is the spray tower, because of its simple design which is not affected by scaling.

Spray towers have received the closest attention as a direct contact heat exchange device, both for liquid-liquid and for boiling applications due to the simplicity of design. A spray tower consists of a vertical vessel of suitable length and diameter, two inlet distributors and outlet connections for both fluids. The heat transfer takes place along the length of the heat exchanger.

3.1 Theoretical Correlations

The design of a direct contact heat exchanger in which heat is exchanged between two immiscible liquids involves two different types of problems that are, to a certain extent, independent of each other. The first problem is the determination of the column diameter in such a way that the

two phases coexist without unwanted interferences as entrainment of one phase by the other one. The second problem is the determination of a suitable length that allows the two fluids to exchange the prescribed duty.

3.1.1 Evaluation of the Column Diameter

The determination of the diameter in a liquid-liquid direct contact heat exchanger involves considerations very close to the ones that govern the design of a packed tower; in either case the limiting point is the flooding of the tower.

The analysis of the flooding condition in a spray column has been done by many authors. Two models proposed by B. C. Sakiadis - A. I. Johnson and by Sideman have been found interesting.

Sakiadis and Johnson established a theoretical equation relating the flow rates of the phases at the flooding point in the form:

$$1 + C_1 \left[\left(\frac{\rho_i}{\rho_b} \right)^{\frac{1}{2}} \left(\frac{U_i}{U_b} \right)^{\frac{1}{2}} \right] = C_s \left[\frac{U_b^2}{g} \frac{\rho_b}{\rho_b - \rho_i} \mu_b^{\frac{1}{2}} \right]^{-\frac{1}{2}} \quad [3.1]$$

Where:

$$C_1 = 1.8$$

$$C_s = 0.565 d^{\frac{1}{2}}$$

and the subscripts i,b refer to the isopentane and brine phase respectively (See "Notation" at the end of this section).

The superficial velocities can now be expressed in terms of mass flow rates, densities, and column diameter as:

$$U_{b,i} = \frac{4 W_{b,i}}{\pi D^2 \rho_{b,i}} \quad [3.2]$$

Rearranging the equation [3.1] and employing [3.2] the following equation is obtained:

$$U_b = \left\{ \frac{0.565 d^{\frac{1}{4}}}{1 + 1.8 \left[\left(\frac{\rho_b}{\rho_i} \right)^{\frac{1}{4}} \left(\frac{W_i}{W_b} \right)^{\frac{1}{2}} \right]} \right\}^2 \left[\frac{(\rho_b - \rho_i) g}{\rho_b \mu_b^{\frac{1}{4}}} \right]^{\frac{1}{2}} \quad [3.3]$$

and thus:

$$D = 2 \sqrt{\frac{Q_b}{\pi U_b}} \quad [3.4]$$

This is the column diameter at the flooding point for a given set of operating parameters.

The actual diameter of the spray tower will include a design margin of about 10 percent on the cross sectional area on the column.

The second model, proposed by Sideman, uses a simple correlation with the holdup in the tower; the design procedure is as follows:

The first thing to evaluate is the droplet terminal velocity from a balance between the gravity force and the drag force which is expressed by the following formula:

$$f V_T^2 = \frac{4}{3} d \frac{(\rho_b - \rho_i)}{\rho_b} g \quad [3.5]$$

f is the friction factor and is a function of the droplet Reynolds number, defined as:

$$Re_d = \frac{V_T d \rho_b}{\mu_b} \quad [3.6]$$

The functional dependency of the friction factor on the Reynolds number has this form:

$$\begin{aligned} f &= \frac{24}{Re} && \text{if } Re < 0.1 \\ f &= \frac{18.5}{Re^{0.6}} && \text{if } 2 < Re < 500 \\ f &= 0.44 && \text{if } 500 < Re < 2 \times 10^5 \\ f &= 0.2 && \text{if } Re < 2 \times 10^5 \end{aligned} \quad [3.7]$$

The above correlations are acceptable assuming that the droplets behave as solid spheres and their diameter is less than 0.2 inches.

At this point, the next quantity to be evaluated is the holdup at the flooding point, given by:

$$\left[K(1 - R) \right] H_f^2 + \left[(K + 1)R \right] H_f - R = 0 \quad [3.8]$$

Where:

$$R = \frac{Q_i}{Q_b} = \frac{W_i}{W_b} \frac{\rho_b}{\rho_i} \quad [3.9]$$

and K has the following dependence on the drop Reynolds number:

$$\begin{aligned} K &= 4.65 && \text{if } Re < 0.2 \\ K &= 4.35 Re^{-0.03} && \text{if } 0.2 < Re < 1 \\ K &= 4.45 Re^{-0.1} && \text{if } 1 < Re < 500 \\ K &= 1.39 && \text{if } Re > 500 \end{aligned} \quad [3.10]$$

It is not recommended to operate at the flooding point but rather use a 10 percent design margin on the flooding holdup; that is:

$$H = 0.9 H_f \quad [3.11]$$

The superficial velocity of the continuous phase may be calculated from the equation:

$$U_b = \frac{V_T H(1 - H)^K}{R(1 - H) + H} \quad [3.12]$$

and thus the column diameter

$$D = 2 \sqrt{\frac{Q_b}{\pi U_b}} \quad [3.13]$$

A comparison of the two models shows that they are in fairly close agreement. Figure 3.1.1.1 and Figure 3.1.1.2 shows the results of the diameter calculations plotted versus the drop-let size for a single well production and a 50 MWe output respectively. These figures utilize the Sakiadis-Johnson and Sideman models for the conditions at the Niland site.

3.1.2 Evaluation of the Column Length

The calculation of the length of a direct contact heat exchanger is not as well defined as the diameter determination already discussed in the preceeding section. There are many theories on this subject, but none of them is considered completely satisfactory. In this section, some of the available models that have the greatest merit for this application will be presented.

A model that gives an approximation of the column length is the one proposed by Sideman and may be employed for a preliminary sizing of the tower.

This model introduces the so called volumetric heat transfer coefficient as the heat that could be exchanged by the unit volume of the dispersed phase as if it were occupying the entire area of the column. The volumetric heat transfer coefficient is assumed to be constant and equal to 12,000 Btu/hr ft³ °F. In the actual situation when only part of the column is occupied by the dispersed phase we have:

$$h_v = 12,000 H \quad [3.14]$$

From this the length of the direct contact heat exchanger is easily determined considering the total heat involved in the process and the temperature difference under which it is exchanged. Then we have:

$$l = \frac{U_b \rho_b C_p (T_{in} - T_{out})_b}{h_v \text{ LMTD}} \quad [3.15]$$

Where:

$$\text{LMTD} = \frac{(T_{in_b} - T_{out_i}) - (T_{out_b} - T_{in_i})}{\ln \frac{(T_{in_b} - T_{out_i})}{(T_{out_b} - T_{in_i})}} \quad [3.16]$$

A model that appears more accurate than the one already described is the one proposed by R. Lethan and E. Kehat. The heat is transferred from the drop in the following ways: (1) to a wake that forms behind the droplet, while the wake is formed; (2) by shedding and renewal of elements of the wakes in most of the column; and (3) by complete mixing of all the streams at the top of the column.

Lethan-Kehat give some correlations to evaluate the wake growth zone and the wake shedding zone, as well as some criteria to evaluate the other zones of the heat exchanger. However, the most important zone to consider, the one in which most of the heat transfer takes place, is the wake shedding zone. The result of the Lethan-Kehat analysis can be summarized in this dimensionless equation:

$$\theta_d = \frac{T_{i \text{ in}} - T_{b \text{ out}}}{T_{i \text{ in}} - T_{b \text{ out}}} = \left\{ \frac{m}{r} \left[\frac{1+S}{\alpha_1} \left(1 - e^{\alpha_1 z} \right) - \frac{S}{\alpha_2} \left(1 - e^{\alpha_2 z} \right) \right] + 1 \right\} e^{-M/r} \quad [3.17]$$

Where:

$$\alpha_{1,2} = -\frac{m}{2} \left[\left(\frac{1}{M} + \frac{1}{r} - \frac{1}{P} \right) \pm \sqrt{\left(\frac{1}{M} + \frac{1}{r} + \frac{1}{P} \right)^2 - \frac{4}{Mr}} \right] \quad [3.18]$$

$$S = \frac{\alpha_1 + \frac{m}{r} - \frac{rm}{PM} (e^{M/r} - 1)}{\alpha_2 - \alpha_1} \quad [3.19]$$

$$M = 0.83 \quad (\text{ratio of wake volume to drop volume})$$

$$m = 0.03 \quad (\text{volume of wake elements shed per volume of drop and length of column})$$

$$r = \frac{(\rho c_p)_i}{(\rho c_p)_b}$$

$$R = U_i / U_b$$

$$P = \frac{1}{R} + M$$

The equation [3.17] gives the temperature profile of the dispersed phase along the column, and for a set of given temperatures ($T_{d \text{ in}}, T_{d \text{ out}}, T_{c \text{ in}}, T_{c \text{ out}}$), can be solved for z to give the unknown length of the wake shedding zone. To take into account the end effects (distribution section, wake growing, mixing, etc.) the authors suggest to add a length of the order of four to five feet.

The work of many authors (among whom include Steinmeyer and Woodward, Suratt and Hart, Lethan and Kehat) indicates that, no matter which correlation is used to evaluate the length of the column, an important parameter to take into account is the aspect ratio of the column defined as length/diameter. It is recommended to maintain this ratio not lower than 10 to avoid considerable back mixing especially near the dispersed phase distributor.

A comparison between the two models reveals that they are not in close agreement except for the larger droplet sizes as shown in Figure 3.1.2.1. What can be said is that the Sideman model can be adequate for a preliminary sizing

of the equipment while the Lethan-Kehat model can be used for more accurate calculation. Nevertheless, it appears that this correlation is very sensitive to the value R (superficial velocities ratio) and that it is quite accurate only for high values of this parameter (as a rule of thumb greater than 1.5). It has been found that the value of R is related to the geothermal site conditions. That is, when the brine temperature is high (in the order of 300 F - 400 F), the preheating section of the direct contact heat exchanger becomes important; at the same time the value of R (determined by flooding considerations) increases in such a way that the Lethan-Kehat correlation gives a good prediction for the required column length. On the other hand, when the brine temperature is in the low range the preheating duty becomes a small share of the total heat to be exchanged in the direct contact heat exchanger and, therefore, an extremely accurate prediction of the preheating length is not possible.

3.1.3 Notation

- c_p - Specific heat at constant pressure, Btu/lb °F
- D - Column diameter, ft
- d - Droplet diameter, in
- f - Friction factor
- g - Acceleration due to gravity, $4.17 \cdot 10^8$ ft/hr²
- H - Holdup

H_f	- Holdup at flooding
h_v	- Volumetric heat transfer coefficient, Btu/hr ft ³ °F
LMTD	- Log mean temperature difference, °F
l	- Column length, ft
Q	- Volume flow rate, ft ³ /hr
R	- Volumetric flow ratio
Re_d	- Droplet Reynolds number
$T_{b\ i}$	- Brine inlet temperature, °F
$T_{b\ out}$	- Brine outlet temperature, °F
$T_{i\ in}$	- Isopentane inlet temperature, °F
$T_{i\ out}$	- Isopentane outlet temperature, °F
U_b	- Superficial velocity of the brine, ft/hr
U_i	- Superficial velocity of the isopentane, ft/hr
V_t	- Droplet terminal velocity, ft/hr
W	- Mass flow rate, lb/hr
Z	- Column length, cm
μ_b	- Viscosity of the brine, cp
ρ_b	- Brine density, lb/ft ³
ρ_i	- Isopentane density, lb/ft ³

4.0 DESIGN OF THE TEST FACILITY

4.1 Introduction

The direct contact heat exchanger test loop has been intended as a means to simulate the operation of a geothermal facility for power production over a wide range of conditions without the need of an actual geothermal well.

This facility can reproduce almost all of the conditions that could be found throughout the country in liquid dominated geothermal sources, including the possibility of injecting noncondensable gases. Figures 4.1.1 and 4.1.2 depict the test facility.

4.2 System Description

The direct contact heat exchanger test loop, as a whole, can be broken down in the following subsystems:

1. Cooling water
2. Hot water
3. Working fluid (isopentane)
4. Makeup water
5. Noncondensable gases

The attached flow diagram describes the entire test facility (Figure 4.2.1). Appendix B contains the engineering specifications for the main equipment.

4.2.1 Cooling Water Subsystem

The cooling water subsystem provides the means to achieve the final heat rejection of the cycle. It is the "cold source"

and it is composed of a centrifugal pump and the isopentane condenser. The cooling fluid is the water of a nearby source that has a sufficient heat capacity to provide cooling water at a constant temperature.

The cooling water is pumped by a centrifugal pump with a maximum flow of 30 gpm. The cooling water flow is set, depending on the requirement, and its value controlled by a flowmeter. On the discharge of the cooling water pump, there is a connection to the suction of the makeup water pump. The isopentane condenser is shared between the cooling water and the working fluid subsystems.

4.2.2 Hot Water (Brine) Subsystem

The hot water subsystem provides the hot water (the "geothermal brine") to the direct contact heat exchanger. The main components of this subsystem are an electrical water heater, a circulation pump, a settling tank, and the direct contact heat exchanger. The water heater is capable of producing hot water at any temperature between 120°F and 400°F by means of an electric resistance of the power of 20 KW (68,250 Btu/hr). The circulation pump, a two-stage turbine type pump, delivers the water to the direct contact heat exchanger, after which the hot water is sent to a settling tank. This vessel has been intended to provide a way to separate small quantities of isopentane (if present in

the hot water stream) and dissolved gases from the hot water. Anything which gathers at the top of the settling tank is blown down to the isopentane storage tank at convenient intervals. The hot water is eventually returned to the electric heater where it is reheated.

4.2.3 Working Fluid Subsystem

The working fluid (isopentane) subsystem is the one that simulates the power cycle that is used in the geothermal applications. The principle components of this system are a feed pump, the direct contact heat exchanger, a set of expansion valves (that simulate the turbine), the isopentane condenser, a holdup tank, and a storage tank. The feed pump can deliver a flow up to 5 gpm at a pressure up to 600 psi to the direct contact heat exchanger. The isopentane coming out of the direct contact heat exchanger is expanded through a relief valve and then condensed in the plate heat exchanger. The outgoing condensate is accumulated into the holdup tank that provides adequate head on the feed pump suction and compensates the effect of flow fluctuations. The isopentane flow is regulated diverting part of the flow to the condenser, and monitored with a flowmeter. The storage tank (of the capacity of 120 gallons) allows the operation of the system without refilling it for a reasonably long time.

4.2.4 Makeup Water Subsystem

The makeup water system, beside its function of a filling device for the entire test loop, has two functions. To set all of the system at the required pressure and to provide the driving fluid for an ejector that pulls the noncondensable gases out of the condenser. This subsystem works continuously when the test facility is operating and also makes up the water that may be lost in the hot water subsystem for any reason.

4.2.5 Noncondensibles Subsystem

The noncondensable subsystem provides a supply of gas for two purposes. The first is to have capability of injecting a gas into the hot water stream (almost always the geothermal waters contain a considerable amount of gases such as H_2S , CO_2 , N_2). The second is to pressurize the isopentane storage tank so that it is easy to transfer isopentane from storage to the system. Furthermore, the gas station provides a mobile supply (via a flexible hose) that is used, for example, to transfer the isopentane into the storage tank, from the barrels in which it is delivered, as well as to purge lines.

4.2.6 Direct Contact Heat Exchanger

The direct contact heat exchanger is composed of four different sections that make it modular. Each section is

made of a spool of pipe six inches in diameter (schedule XS), two feet long, with flanges at each end. The top section has a high pressure sight glass that allows the observation of the interface at the top of the equipment. The other sections are provided with high pressure view ports (two at each location along the length) to control the behavior of the dispersed phase droplets during the operation.

The operation of the test facility can be outlined in the following steps:

1. Fill the system, except the isopentane subsystem, with water.
2. Turn on the water heater and start circulating the hot water in order to warm up the system to the required hot water temperature.
3. Transfer the isopentane from the storage tank into the system, that is into the holdup drum.
4. When the system has reached the desired temperature, turn on the isopentane feed pump with the recirculation valve completely open so that no isopentane is fed into the boiler. At this point, if necessary, other isopentane could be transferred from storage into the system.
5. Set the cooling water flow rate and the hot water flow rate at the desired value and then shut off

slowly the isopentane recirculation valve until the required flow rate has been achieved.

6. When the system has reached a stable condition, data can be collected from the instruments.
7. Once every 10 - 15 minutes, the condenser pressure should be checked to ascertain if there is any build up of noncondensable gases. In this case, the ejector suction should be opened to extract the noncondensibles. Furthermore, once every 15 - 20 minutes the hot water settling tank should be checked to see if there are gases or separated isopentane on top of it. To do this, open the blow-down valve and keep it open until liquid is seen through the flow indicator.

Once the hot water temperature has been established (depending on the geothermal brine temperature to be simulated), the dispersed phase (isopentane) is fed into the direct contact heat exchanger. Since the vessel length is fixed at nine feet for this application, the isopentane flow rate is decreased (the feed temperature is fixed) to establish a maximum outlet temperature for a given hot water condition.

A further step would be decreasing the hot water flow rate in such a way that the heating curves of the isopentane

and the cooling curve of the hot water are parallel (equal terminal temperature differences). This is the most unfavorable condition for the direct contact heat exchanger because the heat is exchanged between the two fluids under a constant temperature difference.

5.0 SYSTEM ANALYSIS VERIFICATION

5.1 Introduction

The analysis of various configurations has led to the identification of the optimum cycle configuration for the GLEF as described in Section 2.5. In the analysis and consequently in the heat balance calculations, each piece of equipment was considered capable of achieving the required performance, regardless of the actual shape or configuration of each individual equipment component. This approach of considering each piece of equipment as a black box is correct as long as at the end of the heat balance, it is established that there exists some way to predict whether the performance required from a particular piece of equipment is feasible or not. From past engineering experience, the performance of all of the equipment can be accurately estimated with the exception of the direct contact preheater for which the technology is not yet well established.

5.2 Preliminary Design

The optimum cycle configuration for the GLEF at the Niland site requires a direct contact preheater on which there is no actual operating experience. Such a preheater should provide a wide temperature increase in the working fluid with relatively small temperature differences. Hence, the experimental verifications to be performed on the test

loop are concerned mainly with whether the thermal performance of the direct contact preheater for the Niland site could be reasonably expected.

The system analysis established the design conditions for the direct contact preheater (i.e. temperatures and flows). On the basis of the available correlations, it was possible to determine the length required for such a piece of equipment.

As described in Section 3, the length required to achieve a certain thermal performance depends on the ratio of brine and isopentane flow rates as well as on their temperatures.

Therefore, for the conditions at the Niland Reservoir, the effective heat transfer length of the direct contact preheater has been evaluated using a computer program recently developed by DAI (See Appendix A); with consideration to the aspect ratio of the vessel. Such a length has been calculated to be about 26.5 feet. (See Appendix D)

5.3 Test Plan

All the tests performed with the direct contact heat exchanger test facility provide support information for the design of direct contact heat exchangers in real situations; thus, the data collected concerns mainly the "approach" that could be achieved with the direct contact heat exchange mechanism and consequently the heat transfer rates.

What is reproduced in the test loop is a part of the optimum cycle evaluated in Section 2.5. This includes the direct contact preheater, a set of expansion valves, a condenser, the isopentane feed pump, and the hot water supply. This setup is sufficient to test the performance of a direct contact preheater, evidently of smaller size than the one to be installed at the GLEF.

The specific goals of the tests are, first, to check whether the direct contact heat exchanger is capable of achieving the terminal temperature differences established in the heat balance; and, second, to evaluate the actual length needed to exchange a certain heat duty.

A small inconvenience of the actual test loop is that the available length for the heat exchange is only nine feet; therefore, the verification of the cycle should be carried out in steps making a fictitious division of the cycle in two parts--one at low temperature and the other at higher temperature. Figure 5.3.1 shows the temperature profiles of the isopentane and the hot water and where the cut between the two cycles was to be performed.

The test to check the minimum approach that could be achieved in the direct contact preheater was carried out at relatively low temperature since there is no need of pushing the test loop at high temperature due to the fact

that the heat transfer coefficient that could be achieved at high temperatures is equal or better than the one achievable at low temperatures. During this test, the isopentane flow has been decreased from the initial value until the minimum terminal temperature difference is reached (i.e. when there is no more variation in the terminal temperature difference even with a reduction of the isopentane flow rate).

6.0 DATA COLLECTION AND ANALYSIS

6.1 Experimental Setup

Once the test loop was started, it was realized that due to the winter season, there would be limitations in controlling the isopentane inlet temperature to the direct contact heat exchanger. The inlet temperature was too low due to the cold atmospheric temperatures. In fact, the temperature of the cooling water source was in the range of 50°F. Preliminary tests on the condenser performance showed also that it was difficult to control the outlet temperature of the isopentane by varying the cooling water flow rate.

For this reason, it was decided to run the tests differently from the original plan, in such a way that the limits induced from the extremely low temperature of the cooling water could be overcome. This was possible by lowering all the temperatures of the cycle.

Figure 6.1 shows the new temperature profiles compared to the previous ones.

All the tests have been carried out with the distribution plate #1 (see isopentane distributor specification), obtaining a droplet diameter of about 0.25 inch.

6.2 Data Collection and Analysis

The experimental data collected from the tests of the

direct contact preheater are listed in Table 6.2.1.

In the test data there were differences between the heat gained by the isopentane and the heat released by the hot water. This was mainly due to the combination of the errors of the flowmeters and of the thermocouples plus some heat lost to the surrounding ambient by the direct contact heat exchanger. Therefore, the heat transfer calculation was based on the heat gained by the isopentane. For this purpose, the volumetric heat transfer coefficient is as follows:

$$h_v = \frac{Q}{V \cdot \text{LMTD}}$$

Where:

Q - Total heat transferred to the isopentane, Btu/hr

V - Direct contact heat exchanger volume for heat transfer, ft³

LMTD - Log mean temperature difference

7.0 CONCLUSIONS AND RECOMMENDATIONS

The purpose of this program was to provide preliminary engineering data for the future evaluation of the technical and economic feasibility of using a direct contact heat exchanger flash-binary cycle at the Niland Reservoir. The specific work to be accomplished was as follows:

TASK I - To perform a system analysis for the determination of the optimum flash-binary cycle.

TASK II - To verify the optimum cycle established in TASK I using the model test loop at the Springlake Research Center including:

1. Verification of the direct contact heat exchanger's capability of achieving the terminal temperature difference established in the heat balance.
2. Confirmation of the length that was determined with the available correlation as described in Section 5.2.

TASK III - To compile a final report documenting the results of the program.

It was concluded in the system analysis that the optimum cycle for the Niland direct contact heat exchanger application described herein be similar to the configuration shown in Figure 2.5.1 entitled, "Heat Balance Diagram for the Liquid-Liquid Preheater Cycle". The data obtained at the model test loop generally supports the assumptions made in the system analysis. In particular:

1. The final heat balance for the cycle described in Section 2.5 required, for the direct contact preheater, a 25° approach on the cold side (brine outlet temperature minus isopentane inlet temperature) and a 52°F approach on the hot side (brine inlet temperature minus isopentane outlet temperature). Under test conditions a minimum approach of 12°F , was obtained, meanwhile, the majority of the calculated approaches (refer to specific data page) ranged below 30°F .
2. The Lethan-Kehat correlation has been used to evaluate the column length and a 26.5 foot effective heat transfer length has been selected for a prototype system (see Section 5.2). The data collected indicates that such a length is adequate and sometimes even redundant. Figure 7.1 shows the direct contact preheater design, based on Sakiadis-Johnson

correlation for the determination of the column diameter, and the Lethan-Kehat correlation for the determination of the column length, for a unit that utilizes the flow of one well at the Niland Reservoir. Appendix D describes how such a design has been accomplished for a unit that utilizes the flow of one well as well as some preliminary indications for a 50 MWe unit.

The completion of TASK I and II have provided a good foundation for selecting an optimum flash-binary cycle. Future work required to determine overall feasibility of direct contact heat exchangers for binary-cycle applications at liquid-dominated geothermal reservoirs should include the following:

1. Collection of a greater number of data points to provide a broader data bank to use during design efforts.
2. Study of the influence of the surface tension in the heat transfer mechanism as well as in the fluid mechanics of the phenomenon.
3. Study of the end effects in the direct contact heat exchanger and development of suitable distribution and extraction devices to avoid collateral effects such as carryover or carryunder.

4. Test of direct contact preheaters with actual brine, possibly at high temperature and with high salt content to control the accuracy of the laboratory data and of the predictions based on them.
5. Determination of the effect of droplet size and residence length upon the solubility of the dispersed hydrocarbon phase within a continuous phase of geothermal brine.

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TABLE 6.2.1

DIRECT CONTACT PREHEATER TEST RESULTS

W_c	- Continuous phase flow rate, lb/hr
W_d	- Dispersed phase (isopentane) flow rate, lb/hr
T_{ci}	- Continuous phase inlet temperature, °F
T_{co}	- Continuous phase outlet temperature, °F
T_{di}	- Isopentane inlet temperature, °F
T_{do}	- Isopentane outlet temperature, °F
LMTD	- Logarithmic mean temperature difference, °F
Q	- Isopentane heat duty, Btu/hr
ΔT_c	- Brine temperature drop, °F
ΔT_d	- Isopentane temperature rise (observed) °F
h_v	- Volumetric heat transfer coefficient, Btu/hr ft ³ °F

TABLE 6.2.1 continued
DIRECT CONTACT PREHEATER TEST RESULTS

Test	W_c	W_d	T_{ci}	T_{co}	T_{di}	T_{do}	LMTD	Q	ΔT_c	ΔT_d	h_v
2	1000	1085	213.8	141.8	66.2	194.	41.6	81800	72.	127.8	1357
3	750	1085	197.6	138.2	64.4	179.6	39.5	73700	59.4	115.2	1287
4	500	1085	213.8	132.8	62.6	170.6	55.6	69150	81.	108.	858
5	1000	620	192.2	138.2	62.6	179.6	35.1	42800	54.	117.	841
6	500	620	199.4	138.2	62.6	170.6	48.5	39500	61.2	108.	562
7	500	1085	212.	131.	77.	186.8	37.8	70400	81.	109.8	1286
8	750	930	208.4	141.8	75.2	176.	47.5	55400	66.6	100.8	805
9	750	930	199.4	145.4	75.2	174.2	43.9	54100	54.	99.	850
10	500	930	210.2	143.6	71.6	174.2	51.9	56300	66.6	102.6	749
11	500	930	219.2	145.4	84.2	177.8	50.6	51300	73.8	93.6	699
12	500	388	208.4	150.8	84.2	177.8	46.3	21400	57.6	93.6	320
13	500	465	206.6	154.4	93.2	181.4	40.5	24200	52.2	88.2	412
14	1000	620	251.6	197.6	84.2	230.	55.4	53300	54.	145.8	665
15	1000	620	244.4	194.	84.2	226.4	50.7	52000	50.4	142.2	707
16	1000	620	246.2	174.2	86.	217.4	53.	48000	72.	131.4	625
18	2250	310	212.	185.	62.6	194.	54.4	24000	27.	131.4	304
19	2250	310	213.8	188.6	62.6	195.8	55.5	24400	25.2	133.2	303
20	2250	310	212.	179.6	68.	197.6	47.5	23700	32.4	129.6	345
21	2250	465	208.4	179.6	71.6	192.2	48.4	33000	28.8	120.6	471
22	2250	930	212.	158.	73.4	199.4	37.8	69100	54.	126.	1261

SCALE DOUBLES FROM 60 TO 300

ENTHALPY, BTU/LB

FIGURE 2.2.1 ISOBUTANE CYCLE

FIGURE 2.2.2 ISOPENTANE CYCLE

FIGURE 2.4.1.1 CYCLE FLOW DIAGRAM

BRINE RATE (NET) = 98.24 LB/KW

FIGURE 2.4.1.2 HEAT BALANCE NO.1 (BOILING TEMP 325 °F)

FIGURE 2.4.1.3 STEAM CONDENSATION CURVE FOR HEAT BALANCE #1

BRINE RATE (NET) = 94.05 LB/KWh

FIGURE 2.4.1.4 HEAT BALANCE NO. 2 (BOILING TEMP. 320 °F)

FIGURE 2.4.1.5 STEAM CONDENSATION CURVE FOR HEAT BALANCE #2

BRINE RATE (NET) = 97.40 LB/KWh

FIGURE 2.4.1.6 HEAT BALANCE NO. 3 (BOILING TEMP. 315 °F)

FIGURE 2.4.1.7 STEAM CONDENSATION CURVE FOR HEAT BALANCE #3

BRINE RATE (NET) = 99.01 LB/KWh

FIGURE 2.4.1.8 HEAT BALANCE NO. 4 (BOILING TEMP 310 °F)

FIGURE 2.4.1.9 STEAM CONDENSATION CURVE FOR HEAT BALANCE #4

FIGURE 2.4.1.10 BRINE RATE AND AUXILIARY POWER VS. BOILING TEMPERATURE FOR THE BINARY ("LIQUID/LIQUID") CYCLE

FIGURE 2.4.2.1 CYCLE FLOW DIAGRAM

BRINE RATE (NET) = 190.6 LB/KWh

FIGURE 2.4.2.2 HEAT BALANCE NO.5 (BOILING TEMP. 240 °F)

BRINE RATE (NET) = 187.2 LB/KWh

FIGURE 2.4.2.3 HEAT BALANCE NO. 6 (BOILING TEMP 250 °F)

BRINE RATE (NET) = 188.4 LB/KWh

FIGURE 2.4.2.4 HEAT BALANCE NO. 7 (BOILING TEMP. 260 °F)

BRINE RATE (NET) = 214.6 LB/KWh

FIGURE 2.4.2.5 HEAT BALANCE NO. 8 (BOILING TEMP. 270 °F)

FIGURE 2.4.2.6 BRINE RATE VERSUS BOILING TEMPERATURE
FOR THE DIRECT CONTACT CYCLE

FIGURE 2.4.3.1 CYCLE FLOW DIAGRAM

FIGURE 2.4.3.2 CYCLES WITH AND WITHOUT SUPERHEAT

BRINE RATE (NET) = 185.06 LB/KWh

FIGURE 2.4.3.3 HEAT BALANCE NO. 9

P - PSIA
T - °F
H - BTU/LB
M - 10³ LB/HR

FIGURE 2.4.4.1 CYCLE FLOW DIAGRAM

BRINE RATE (NET) = 108.3 LB/KWh

FIGURE 2.4.4.2 HEAT BALANCE NO.10 (BOILING TEMP. 240° F)

P - PSIA
T - °F
H - BTU/LB
M - 10³ LB/HR

BRINE RATE (NET) = 107.1 LB/KWh

FIGURE 2.4.4.3 HEAT BALANCE NO. 11 (BOILING TEMP. 250°F)

P - PSIA
T - °F
H - BTU/LB
M - 10³ LB/HR

BRINE RATE (NET) = 107.5 LB/KWh

FIGURE 2.4.4.4 HEAT BALANCE NO. 12 (BOILING TEMP. 260°F)

P - PSIA
T - °F
H - BTU/LB
M - 10³ LB/HR

BRINE RATE (NET) = 130.0 LB/KWh

FIGURE 2.4.4.5 HEAT BALANCE NO. 13 (BOILING TEMP. 270°F)

FIGURE 2.4.4.6 BRINE RATE AND AUXILIARY POWER VERSUS BOILING TEMPERATURE FOR THE DIRECT-CONTACT CYCLE WITH STEAM TURBINE

FIGURE 2.5.1 HEAT BALANCE DIAGRAM FOR THE LIQUID-LIQUID PREHEATER CYCLE

FIGURE 3.1.1.1 COLUMN DIAMETER VERSUS DROPLET SIZE ACCORDING TO THE SIDEMAN AND SAKIADIS - JOHNSON CORRELATIONS FOR ONE WELL FLOW

FIGURE 3.1.1.2 COLUMN DIAMETER VERSUS DROPLET SIZE ACCORDING TO THE SIDEMAN AND SAKIADIS - JOHNSON CORRELATIONS FOR 50 MWE PLANT (6 UNITS REQUIRED)

FIGURE 3.1.2.1 COLUMN LENGTH VERSUS DROPLET SIZE ACCORDING TO THE SIDEMAN AND LETHAN - KEHAT CORRELATIONS

FIGURE 4.1.1 VIEW OF THE DIRECT CONTACT HEAT EXCHANGER TEST LOOP

FIGURE 4.1.2 PARTICULAR OF THE CONTROL PANEL

FIGURE 4.2.1 GLEF TEST LOOP FLOW DIAGRAM

	CONTINUOUS PHASE	DISPERSED PHASE
FLOW RATE (LB/HR)	.4176E+05	.5784E+05
DENSITY (LB/FT ³)	.6807E+02	.3277E+02
VISCOSITY (CP)	.3000E+00	
SPEC. HEAT (BTU/LB F)	.8200E+00	.6653E+00
INLET TEMP. (F)	.3720E+03	.1100E+03
OUTLET TEMP. (F)	.1360E+03	.3200E+03

STRIKE RETURN TO EXECUTE, TYPE 1 TO STOP

DROPLET SIZE (IN.)

.1

DROP SIZE (IN.)	.1000E+00
DROP VELOCITY (FT/S)	.1098E+00
CONTINUOUS PHASE VELOCITY (FT/S)	.3818E-01
EXCHANGER USEFUL LENGTH (FT)	.2143E+02
EXCHANGER DIAMETER (FT)	.2384E+01

FIGURE 5.2.1 PRELIMINARY DESIGN OF THE DIRECT-CONTACT PREHEATER
FOR A 500 KW PROTOTYPE

FIGURE 5.3.1 TEMPERATURE PROFILES OF ISOPENTANE AND HOT WATER IN THE TEST LOOP

FIGURE 6.1.1 TEMPERATURE PROFILES OF ISOPENTANE AND HOT WATER IN THE TEST LOOP (MODIFIED)

FIGURE 7.1 DIRECT CONTACT PREHEATER

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APPENDIX A

COMPUTER PROGRAM FOR THE
DESIGN OF DIRECT CONTACT PREHEATERS

The design of the preheating section of a direct contact heat exchanger is known to be the most difficult part. Since this part of the calculation involves a great number of operations, a computer program has been developed that enables the selection of the most attractive configuration. This program has the capability of using different mathematical models for the determination of the diameter and the length of the heat exchanger. The diameter is determined using either the Jacob-Boehm or the Sakiadis-Johnson model; the exchanger length is calculated using either the Sideman or the Lethan-Kehat model.

A code allows the user of the program to choose a combination of the above models as required.

The input data for this computer program can be divided as follows:

1. Data to be held constant during the entire execution such as flow rates, physical properties, temperatures, and the mathematical models to use for the evaluation of diameter and length.
2. Data that changes each time the calculation is performed such as the droplet diameter.

To determine the design of a direct contact preheater, one produces a number of possible designs by varying the

droplet size; this means a number of direct contact heat exchangers that satisfy the flooding limitations and that have a length sufficient to exchange the prescribed heat duty. The final choice among these designs will be done by the engineer based on his judgement and experience.

The following pages contain the different designs for a 500 KW prototype system at the Niland site. These designs have been performed using the Sakiadis-Johnson correlation for the determination of the column diameter and the Lethan-Kehat correlation for the column length. Among them, the most attractive design is the one with droplet size of 0.1 inches.

EXEC DCHX2, \. \LENLET
LINK: Loading
[LNKXCT DCHX2 Execution]

CONTINUOUS PHASE INPUT DATA

FLOW RATE(LB/HR) , INLET & OUTLET TEMP.(F)

41760,372,136

DENSITY(LB/FT3) , VISCOSITY(CP) , SPEC.HEAT(BTU/LB F)

68, \. \.07, .3, 0, \. \. \.82

DISPERSED PHASE INPUT DATA

FLOW RATE(LB/HR) , INLET & OUTLET TEMP(F)

57840,110,320

DENSITY(LB/FT3) , SPEC.HEAT(BTU/LB F)

32.77, .6653

CONTINUOUS PHASE

DISPERSED PHASE

FLOW RATE	(LB/HR)	.4176E+05	.5784E+05
DENSITY	(LB/FT3)	.6807E+02	.3277E+02
VISCOSITY	(CP)	.3000E+00	
SPEC. HEAT	(BTU/LB F)	.8200E+00	.6653E+00
INLET TEMP.	(F)	.3720E+03	.1100E+03
OUTLET TEMP.	(F)	.1360E+03	.3200E+03

STRIKE RETURN TO EXECUTE, TYPE 1 TO STOP

DROPLET SIZE (IN.)

.04

DROP SIZE	(IN.)	.4000E-01
DROP VELOCITY	(FT/S)	.6947E-01
CONTINUOUS PHASE VELOCITY	(FT/S)	.2415E-01
EXCHANGER USEFUL LENGTH	(FT)	.2143E+02
EXCHANGER DIAMETER	(FT)	.2998E+01

STRIKE RETURN TO EXECUTE, TYPE 1 TO STOP

1

END OF EXECUTION

CPU TIME: 0.20 ELAPSED TIME: 1:43.10

EXIT

.EXEC DCHX2,LENLET
LINK: Loading
[LNKXCT DCHX2 Execution]

CONTINUOUS PHASE INPUT DATA

FLOW RATE(LB/HR) , INLET & OUTLET TEMP.(F)

41760,372,136

DENSITY(LB/FT3) , VISCOSITY(CP) , SPEC.HEAT(BTU/LB F)

68.07,.3,.82

DISPERSED PHASE INPUT DATA

FLOW RATE(LB/HR) , INLET & OUTLET TEMP(F)

57840,110,320

DENSITY(LB/FT3) , SPEC.HEAT(BTU/LB F)

32.77,.6653

CONTINUOUS PHASE

DISPERSED PHASE

FLOW RATE	(LB/HR)	.4176E+05	.5784E+05
DENSITY	(LB/FT3)	.6807E+02	.3277E+02
VISCOSITY	(CP)	.3000E+00	
SPEC. HEAT	(BTU/LB F)	.8200E+00	.6653E+00
INLET TEMP.	(F)	.3720E+03	.1100E+03
OUTLET TEMP.	(F)	.1360E+03	.3200E+03

STRIKE RETURN TO EXECUTE, TYPE 1 TO STOP

DROPLET SIZE (IN.)

.06

DROP SIZE	(IN.)	.6000E-01
DROP VELOCITY	(FT/S)	.8509E-01
CONTINUOUS PHASE VELOCITY	(FT/S)	.2957E-01
EXCHANGER USEFUL LENGTH	(FT)	.2143E+02
EXCHANGER DIAMETER	(FT)	.2709E+01

STRIKE RETURN TO EXECUTE, TYPE 1 TO STOP

11

END OF EXECUTION

CPU TIME: 0.21 ELAPSED TIME: 1:15.43

EXIT

.EXEC DCHX2,LENLET
LINK: Loading
[LNKXCT DCHX2 Execution]

CONTINUOUS PHASE INPUT DATA

FLOW RATE(LB/HR) , INLET & OUTLET TEMP.(F)
41760,372,136
DENSITY(LB/FT3) , VISCOSITY(CP) , SPEC.HEAT(BTU/LB F)
68.07,.3,.82

DISPERSED PHASE INPUT DATA

FLOW RATE(LB/HR) , INLET & OUTLET TEMP(F)
57840,110,320
DENSITY(LB/FT3) , SPEC.HEAT(BTU/LB F)
32.77,.6653

CONTINUOUS PHASE		DISPERSED PHASE
------------------	--	-----------------

FLOW RATE	(LB/HR)	.4176E+05	.5784E+05
DENSITY	(LB/FT3)	.6807E+02	.3277E+02
VISCOSITY	(CP)	.3000E+00	
SPEC. HEAT	(BTU/LB F)	.8200E+00	.6653E+00
INLET TEMP.	(F)	.3720E+03	.1100E+03
OUTLET TEMP.	(F)	.1360E+03	.3200E+03

STRIKE RETURN TO EXECUTE, TYPE 1 TO STOP

DROPLET SIZE (IN.)
.08

DROP SIZE	(IN.)	.8000E-01
DROP VELOCITY	(FT/S)	.9825E-01
CONTINUOUS PHASE VELOCITY	(FT/S)	.3415E-01
EXCHANGER USEFUL LENGTH	(FT)	.2143E+02
EXCHANGER DIAMETER	(FT)	.2521E+01

STRIKE RETURN TO EXECUTE, TYPE 1 TO STOP

1
END OF EXECUTION
CPU TIME: 0.21 ELAPSED TIME: 1:26.55
EXIT

EXEC DCHX2,LENLET
LINK: Loading
CLNKXCT DCHX2 Execution]

CONTINUOUS PHASE INPUT DATA
FLOW RATE(LB/HR) , INLET & OUTLET TEMP.(F)
41760,372,136
DENSITY(LB/FT3) , VISCOSITY(CP) , SPEC.HEAT(BTU/LB F)
68.07,.3,.83\3\2

DISPERSED PHASE INPUT DATA
FLOW RATE(LB/HR) , INLET & OUTLET TEMP(F)
57840,110,320
DENSITY(LB/FT3) , SPEC.HEAT(BTU/LB F)
32.77,.6653

	CONTINUOUS PHASE	DISPERSED PHASE
FLOW RATE (LB/HR)	.4176E+05	.5784E+05
DENSITY (LB/FT3)	.6807E+02	.3277E+02
VISCOSITY (CP)	.3000E+00	
SPEC. HEAT (BTU/LB F)	.8200E+00	.6653E+00
INLET TEMP. (F)	.3720E+03	.1100E+03
OUTLET TEMP. (F)	.1360E+03	.3200E+03

STRIKE RETURN TO EXECUTE, TYPE 1 TO STOP

DROPLET SIZE (IN.)
.1

DROP SIZE (IN.)	.1000E+00
DROP VELOCITY (FT/S)	.1098E+00
CONTINUOUS PHASE VELOCITY(FT/S)	.3818E-01
EXCHANGER USEFUL LENGTH (FT)	.2143E+02
EXCHANGER DIAMETER (FT)	.2384E+01

STRIKE RETURN TO EXECUTE, TYPE 1 TO STOP

1

END OF EXECUTION
CPU TIME: 0.20 ELAPSED TIME: 1:12.32
EXIT

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.EXEC DCHX2,LENLET
LINK: Loading
[LNKXCT DCHX2 Execution]

CONTINUOUS PHASE INPUT DATA

FLOW RATE(LB/HR) , INLET & OUTLET TEMP.(F)

41760,372,136

DENSITY(LB/FT3) , VISCOSITY(CP) , SPEC.HEAT(BTU/LB F)

68.07,.3,.82

DISPERSED PHASE INPUT DATA

FLOW RATE(LB/HR) , INLET & OUTLET TEMP(F)

57840,110,320

DENSITY(LB/FT3) , SPEC.HEAT(BTU/LB F)

32.77,.6653

CONTINUOUS PHASE

DISPERSED PHASE

FLOW RATE	(LB/HR)	.4176E+05	.5784E+05
DENSITY	(LB/FT3)	.6807E+02	.3277E+02
VISCOSITY	(CP)	.3000E+00	
SPEC. HEAT	(BTU/LB F)	.8200E+00	.6653E+00
INLET TEMP.	(F)	.3720E+03	.1100E+03
OUTLET TEMP.	(F)	.1360E+03	.3200E+03

STRIKE RETURN TO EXECUTE, TYPE 1 TO STOP

DROPLET SIZE (IN.)

.2

DROP SIZE	(IN.)	.2000E+00
DROP VELOCITY	(FT/S)	.1553E+00
CONTINUOUS PHASE VELOCITY	(FT/S)	.5400E-01
EXCHANGER USEFUL LENGTH	(FT)	.2143E+02
EXCHANGER DIAMETER	(FT)	.2005E+01

STRIKE RETURN TO EXECUTE, TYPE 1 TO STOP

1

END OF EXECUTION

CPU TIME: 0.20 ELAPSED TIME: 1:15.87

EXIT

.EXEC DCHX2,LENLET
LINK: Loading
CLNKXCT DCHX2 Execution]

CONTINUOUS PHASE INPUT DATA
FLOW RATE(LB/HR) , INLET & OUTLET TEMP.(F)
41760,372,136
DENSITY(LB/FT3) , VISCOSITY(CP) , SPEC.HEAT(BTU/LB F)
68.07,.3,.82

DISPERSED PHASE INPUT DATA
FLOW RATE(LB/HR) , INLET & OUTLET TEMP(F)
57840,110,320
DENSITY(LB/FT3) , SPEC.HEAT(BTU/LB F)
32.77,.6653

	CONTINUOUS PHASE	DISPERSED PHASE
--	------------------	-----------------

FLOW RATE	(LB/HR)	.4176E+05	.5784E+05
DENSITY	(LB/FT3)	.6807E+02	.3277E+02
VISCOSITY	(CP)	.3000E+00	
SPEC. HEAT	(BTU/LB F)	.8200E+00	.6653E+00
INLET TEMP.	(F)	.3720E+03	.1100E+03
OUTLET TEMP.	(F)	.1360E+03	.3200E+03

STRIKE RETURN TO EXECUTE, TYPE 1 TO STOP

DROPLET SIZE (IN.)

.3

DROP SIZE	(IN.)	.3000E+00
DROP VELOCITY	(FT/S)	.1903E+00
CONTINUOUS PHASE VELOCITY	(FT/S)	.6613E-01
EXCHANGER USEFUL LENGTH	(FT)	.2143E+02
EXCHANGER DIAMETER	(FT)	.1811E+01

STRIKE RETURN TO EXECUTE, TYPE 1 TO STOP

1

END OF EXECUTION
CPU TIME: 0.20 ELAPSED TIME: 1:16.97
EXIT

.
EXEC DCHX2,LENLET
LINK: Loading
CLNKXCT DCHX2 Execution]

CONTINUOUS PHASE INPUT DATA

FLOW RATE(LB/HR) , INLET & OUTLET TEMP.(F)
41760,372,136
DENSITY(LB/FT3) , VISCOSITY(CP) , SPEC.HEAT(BTU/LB F)
68.07,.3,.82

DISPERSED PHASE INPUT DATA

FLOW RATE(LB/HR) , INLET & OUTLET TEMP(F)
57840,110,320
DENSITY(LB/FT3) , SPEC.HEAT(BTU/LB F)
32.77,.6653

CONTINUOUS PHASE DISPERSED PHASE

FLOW RATE	(LB/HR)	.4176E+05	.5784E+05
DENSITY	(LB/FT3)	.6807E+02	.3277E+02
VISCOSITY	(CP)	.3000E+00	
SPEC. HEAT	(BTU/LB F)	.8200E+00	.6653E+00
INLET TEMP.	(F)	.3720E+03	.1100E+03
OUTLET TEMP.	(F)	.1360E+03	.3200E+03

STRIKE RETURN TO EXECUTE, TYPE 1 TO STOP

DROPLET SIZE (IN.)

.4

DROP SIZE	(IN.)	.4000E+00
DROP VELOCITY	(FT/S)	.2197E+00
CONTINUOUS PHASE VELOCITY	(FT/S)	.7636E-01
EXCHANGER USEFUL LENGTH	(FT)	.2143E+02
EXCHANGER DIAMETER	(FT)	.1686E+01

STRIKE RETURN TO EXECUTE, TYPE 1 TO STOP

1

END OF EXECUTION

CPU TIME: 0.21 ELAPSED TIME: 1:12.13

EXIT

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APPENDIX B

ENGINEERING SPECIFICATIONS FOR THE MAIN
EQUIPMENT UTILIZED IN THE TEST FACILITY

The engineering specifications for all of the main equipment installed in the direct contact test facility is attached hereto; they concern:

1. Cooling water pump
2. Makeup water pump
3. Isopentane feed pump
4. Hot water pump
5. Isopentane condenser
6. Water heater
7. Isopentane distributor

COOLING WATER PUMPOperating Conditions

Fluid	:	water
Fluid specific gravity (60/60)	:	1.0
Flow	:	30 gpm
TDH	:	100 feet
NPSH (required)	:	20 feet

Design Conditions

Design pressure	:	75 psig
Design temperature	:	150° F

MAKEUP WATER PUMPOperating Conditions

Fluid	:	water
Fluid specific gravity (60/60)	:	1.0
Flow	:	5 gpm
TDH	:	600 psi
NPSH (required)	:	by vendor

Design Conditions

Design pressure	:	650 psig
Design temperature	:	150° F

ISOPENTANE FEED PUMPOperating Conditions

Fluid	:	isopentane
Fluid specific gravity (60/60)	:	0.62
Flow	:	5 gpm
TDH	:	600 psi
NPSH (required)	:	by vendor

Design Conditions

Design pressure	:	650 psig
Design temperature	:	150° F

HOT WATER PUMP

Operating Conditions

Fluid	:	water
Fluid specific gravity (60/60)	:	0.98
Flow	:	5 gpm
TDH	:	400 psi
NPSH (required)	:	200 psi

Design Conditions

Design pressure	:	600 psig
Design temperature	:	380° F

ISOPENTANE CONDENSEROperating Conditions

Hot fluid	:	isopentane
Hot fluid flow	:	750 lb/hr (vapor)
Hot fluid inlet temperature	:	100° F
Hot fluid outlet temperature	:	100° F
Cold fluid	:	cooling water
Cold fluid flow	:	30 gpm
Cold fluid inlet temperature	:	80° F
Cold fluid outlet temperature	:	90° F
Heat duty	:	100.000 Btu/hr

Design Conditions

Hot side design pressure	:	75 psig
Hot side design temperature	:	250° F
Cold side design pressure	:	75 psig
Cold side design temperature	:	150° F

WATER HEATER

Heat duty	:	20 KW
Maximum operating temperature	:	400° F
Design pressure	:	250 psig
Design temperature	:	400° F

The control system should include a normal operation thermostat with a range 120° - 400° F and a manual reset safety thermostat set at 400° F.

ISOPENTANE DISTRIBUTOR

The isopentane distributor was fabricated with a standard pipe cap connected to the feed line, and two different perforated plates. The seal between the caps and the distribution plate was obtained with an "O" ring. Attached hereto are the details of the distribution plates and a sketch of the distributor assembly.

PLATE #1

Holes diameter 0.125"

Orifice Location

<u>Circle $\emptyset$</u>	<u>Arc</u>
1"	90°
1.75"	45°
2.5"	45°

PLATE #2

Holes diameter 0.0625"

Orifice Location

<u>Circle $\emptyset$</u>	<u>Arc</u>
1.0"	45°
1.5"	30°
2.0"	22° 30'
2.5"	18°
3.0"	15°

FIGURE B.1 ISOPENTANE DISTRIBUTOR

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APPENDIX C

CYCLE EVALUATION PROCEDURE

In this section, the basic steps of the procedure used in evaluating the performance of geothermal cycles for the case of flash binary systems is described.

Flash Binary Cycles

A flash binary cycle is the one that is adopted in the case of geothermal sources presenting a two-phase flow at the well head. This phenomenon may be due to many reasons such as insufficient pressure in the geothermal reservoir and the unavailability of pumping devices.

The two-phase stream coming out from the well is first separated into a flash tank in which a fraction of the non-condensable gases present in the geothermal field are separated from the liquid phase (brine).

Flash Tank

A heat and mass balance around the flash tank gives the amount of steam and of liquid brine which can be obtained once the flash tank pressure has been selected. The input data for this calculation is:

h_s - steam enthalpy @ P_f

h_b - brine enthalpy @ P_f

h_w - downhole enthalpy

Using the principles of conservation of mass and energy we have (considering the unit flow to the flash tank):

$$1 = x_s + x_b \quad [1]$$

$$h_w = h_s x_s + h_b x_b \quad [2]$$

Solving the system of [1] and [2] the steam and liquid brine flow rates respectively are obtained. From physical properties tables, one can evaluate all of the parameters of both streams coming out of the flash tank such as temperatures, pressure, and enthalpy.

Brine Flow and Outlet Temperature

At this point it is possible to conduct the heat balance for the binary cycle making the following assumptions:

1. The hydrocarbon preheater will be kept separate from the boiler and will be a direct contact heat exchanger.
2. The liquid brine will be used to accomplish all the hydrocarbon preheating if possible, but no boiling will be considered into the direct contact device to avoid taking into account the partial pressure of the water.
3. The steam will be used to boil the hydrocarbon and to achieve the final part of the preheating in a shell and tube heat exchanger.
4. A minimum ΔT (in the order of 20°F) will be allowed between the outlet temperature of the condensed

steam and the inlet temperature of the hydrocarbon.

5. The condensation curve for the steam due to the presence of noncondensable gases has been already calculated.
6. All the calculations are performed considering one pound of hydrocarbon.

The hydrocarbon boiling temperature (T_{boil}) is fixed and the following quantities are evaluated from the thermodynamic properties of the working fluid:

P_h - hydrocarbon saturation pressure @ T_{boil}

λ_h - hydrocarbon latent heat of vaporization @ T_{boil}

$Q_{\text{ph}} = h_{\text{in}} - h_{\text{out}}$ hydrocarbon specific duty for the preheating

λ - steam specific duty obtained from the steam condensation curve

Thus, the steam and liquid brine flow rates necessary to boil and preheat the unit weight of working fluid is evaluated:

$$W_s = \frac{\lambda_h}{\lambda_s} \quad [3]$$

$$W_b = W_s \frac{x_b}{x_s} \quad [4]$$

$$W_w = W_b + W_s \quad [5]$$

The brine outlet temperature is given by:

$$T_{bo} = T_{bi} - \frac{W_b}{C_{pb} \cdot Q_{ph}} \quad [6]$$

Turbine Inlet Conditions

The turbine entrance parameters are determined from physical properties tables:

h_i - hydrocarbon enthalpy at T_{boil}

s_i - hydrocarbon entropy at T_{boil}

p_i - admission pressure at T_{boil}

Turbine Outlet Conditions

With an assumed condensation temperature (which can be evaluated knowing the characteristics of the available cooling water), the condenser pressure and the turbine back pressure is determined:

$$P_2 = P_{sat} @ T_{cond} \quad [7]$$

To calculate the expansion line end point, the following parameters are taken from the hydrocarbon thermodynamic properties:

h_v - saturated vapor enthalpy at T_{cond}

h_f - saturated liquid enthalpy at T_{cond}

S_v - saturated vapor entrpoy at T_{cond}

S_f - saturated liquid entropy at T_{cond}

It is now possible to calculate the final enthalpy of the hydrocarbon after the expansion once an expansion efficiency (η_t) is assumed.

$$h_2 = h_i - \frac{(h_v - h_1)}{(s_v - s_1)} (s_i - s_v) \eta_t \quad [8]$$

The turbine outlet temperature will be given by:

$$T_2 = T_{\text{cond}} + \frac{h_2 - h_v}{C_{\text{phv}}} \quad [9]$$

The mechanical work obtained in the turbine is thus:

$$P = h_i - h_2 \quad [10]$$

Condenser

We already know the condensation temperature T_{cond} ; the condenser duty is determined from:

$$Q_{\text{cond}} = h_2 - h_1 \quad [11]$$

and the required cooling water flow from:

$$W_{\text{cw}} = \frac{Q_{\text{cond}}}{T_{\text{wo}} - T_{\text{wi}}} \quad [12]$$

Final Considerations

The flows, pressures, and temperatures in the entire system, as well as the work produced by the turbine, has now

been determined and it is possible to evaluate the power required by the main pumps and other auxiliaries, thus obtaining the net output.

The brine rate, that is an expression of the efficiency of the cycle, is now given by:

$$BR = \frac{W_w}{P_{net}} \quad [13]$$

The entire procedure should be repeated for different boiling temperatures until the optimum set of operating parameters is identified, that is the minimum value of the brine rate.

Notation

- BR - brine rate
- C_{pb} - brine specific
- C_{phv} - hydrocarbon vapor specific heat
- h_b - liquid brine from flash tank enthalpy
- h_l - saturated hydrocarbon liquid enthalpy at condensing temperature
- h_s - steam from flash tank enthalpy
- h_v - saturated hydrocarbon vapor enthalpy at condensing temperature
- h_w - downhole brine total enthalpy
- h_i - turbine inlet enthalpy

h_2	- turbine outlet enthalpy
P_f	- flash tank pressure
P_h	- hydrocarbon boiling pressure
P_1	- turbine inlet pressure
P_2	- turbine outlet pressure
P_{net}	- net power output
Q_{cond}	- condenser heat duty
Q_{ph}	- preheater heat duty
s_l	- saturated hydrocarbon liquid entropy
s_v	- saturated hydrocarbon vapor entropy
s_i	- turbine inlet entropy
T_{bi}	- brine inlet temperature
T_{bo}	- brine outlet temperature
T_{boil}	- hydrocarbon boiling temperature
T_2	- turbine outlet temperature
T_{wi}	- cooling water inlet temperature
T_{wo}	- cooling water outlet temperature
W_b	- brine flow rate to direct contact preheater
W_s	- steam flow rate to hydrocarbon boiler
W_w	- total brine flow rate from well
x_b	- liquid fraction in the flash tank
x_s	- vapor (steam) fraction in the flash tank
η_t	- turbine expansion efficiency
λ_h	- hydrocarbon latent heat @ T_{boil}

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APPENDIX D

THE DESIGN OF A DIRECT CONTACT PRE-
HEATER FOR A UNIT UTILIZING THE FULL
FLOW OF ONE WELL AT THE NILAND RESERVOIR

The production of one well at the Niland site is designated to start at a flow of 400,000 lb/hr and gradually decrease to 300,000 lb/hr during the plant lifetime.

As described in Section 2, there is the need for a flash tank downstream of the wellhead to achieve the separation of the noncondensable gases as well as the steam that has already flashed. With the flash tank pressure set at 150 psia, 88 percent of the incoming flow will come out as a liquid. For the flow of one well, this translates to 355,200 lb/hr of liquid brine available for the power cycle.

The net output of a unit that utilizes the flow of one well, based on the cycle analysis described in Section 2, is estimated to be about 4.25 MWe at the beginning of the operation. It will decrease gradually during the plant lifetime to 3.2 MWe. Figure D.1 shows a flow diagram for the one well unit with 400,000 lb/hr of brine flow.

The design of the direct contact preheater has been performed using the Sakiadis-Johnson correlation for the determination of the diameter and the Lethan-Kehat correlation for the determination of the length.

The results of the diameter calculation are plotted in Figure D.2; a feasible range for the droplet diameter is between 0.0625 inches and 0.25 inches. In this case, a larger droplet size was employed because it leads to vessel dimensions which are easier to fabricate and set up. In particular, a droplet size of 0.2 inches has been selected leading to a column diameter of 6 feet.

The calculation of the column length, using the Lethan-Kehat correlation, leads to a preliminary effective heat transfer length of 19.1 feet for the conditions at the Ni-land Reservoir.

These vessel dimensions, while easy to fabricate and set up, cause a low aspect ratio ($L/D = 3.17$). Aspect ratios of 10 are generally employed in the design of such vessels.

Low aspect ratios can lead to backmixing, so in order to alleviate this possible occurrence, the test vessel should have longitudinal baffles, creating vertical channels inside the straight part of the column. In this case, the equivalent diameter (D_e) of each channel can be used; it is defined as:

$$D_e = \frac{4 A}{P}$$

Where A is the cross sectional area of the channel and P is the wetted perimeter. The installation of two longitudinal

baffles (realizing thus four channels) gives an equivalent diameter for each channel of 2.65 feet. Further, such an arrangement does not significantly add to the maintenance requirements of the vessel. Using this diameter and an aspect ratio of 10, we obtain a column length of 26.5 feet. Additional length should be provided as necessary for the installation of the fluid distribution points.

In conclusion, a suggested design for the direct contact preheater is shown in Figure D.3. The column diameter is 6 feet, its effective heat transfer length is 26.5 feet. Two longitudinal baffles should be installed in the straight part of the heat exchanger as is seen in the figure. Additional length beyond the effective heat transfer length of 26.5 feet will be required, as necessary to allow for the installation of the appropriate piping for the distribution and exit points.

INDICATIONS ON THE DIRECT CONTACT PRE- HEATER TO BE INSTALLED IN THE 50 MWe PLANT

The 50 MWe unit uses a brine flow such that twelve wells are needed (See Section 2.5). For such a flow (4,182,000 lb/hr) it is not feasible to install a single direct contact preheater; therefore, a number of units arranged in parallel will be suggested.

One possible option is to utilize twelve direct contact preheaters like the one used for the single well flow condition. (See Figure D3)

FIGURE D.1 HEAT BALANCE DIAGRAM FOR ONE WELL PLANT

FIGURE D.2 COLUMN DIAMETER VERSUS DROPLET SIZE ACCORDING TO THE SIDEMAN AND SAKIADIS - JOHNSON CORRELATIONS FOR ONE WELL FLOW

FIGURE D.3 DIRECT CONTACT PREHEATER