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FINAL PHASE TESTING AND EVALUATION OF THE 500 KW DIRECT
CONTACT PILOT PLANT AT EAST MESA

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1.0 INTRODUCTION

The Geothermal Direct Contact Heat Exchanger Program was started in late 1977 to evaluate the potential of the direct contact principle and provide design criteria for large commercial plants through the testing of a 500 kW pilot plant. The final phase in evaluation of the direct contact pilot plant was performed during 1982. This report summarizes the work and results obtained in this final phase of evaluation.

Experimental models had proven the direct contact heat exchanger process on a very small scale basis and several economic studies evaluating the potential of large scale plants showed them to be potentially cost effective even though there are system complications introduced by the direct contact approach. The potential of the direct contact approach encouraged the Department of Energy to explore the performance potential further with a 500 kW pilot plant. The plant was configured to evaluate the potential of the direct contact approach at a reasonable level of scale as applied to geothermal energy and provide design criteria for large commercial plants. Figure 1.1 shows the layout of the 500 kW pilot plant at the East Mesa test facility. The plant's direct contact heat exchanger is shown in Figure 1.2.

The objectives of this last phase of the program were to:

- 1) explore further the performance limits of the plant's direct contact heat exchanger, 2) perform an engineering evaluation of methods of recovering working fluid from the spent brine, and 3) develop data on the alternate turbine configuration and evaluate its performance.

The baseline design and the plant testing that was performed prior to this report is covered in LBL reports UCID-80091, LBL-12115 and LBL-13339.

FIGURE 1.1 - 500 kW Direct Contact Pilot Plant

FIGURE 1.2 - Direct Contact Heat Exchanger

2.0 SUMMARY

2.1 DCHX TESTING AND EVALUATION

The testing performed during the last phase of the geothermal direct contact heat exchanger program utilizing the 500 kW pilot plant provided more insight into the capabilities and limits of the direct contact approach and showed that more work needs to be done to understand the inner workings of a large direct contact heat exchanger if they are to be modeled analytically. Testing of the column demonstrated that the performance was excellent and that the sizing criteria is conservative. The system operated smoothly and was readily controlled over a wide range of operating conditions. Performance evaluation showed pinch differentials of 4°F or less and better than predicted heat transfer capability.

Testing during this final phase was directed towards establishing the limits of the column to transfer heat. The working column height was shortened progressively to approximately 16 feet from a design length of 28 feet. The short column performed as well as a full length column and there are indications that the column could have been shortened even more without affecting its ability to transfer heat. The column's ability to perform as well with shortened lengths indicates that the heat transfer coefficients and criteria derived from the small scale tests are very conservative.

Column temperature profiles recorded over the length of the column were not characteristic of a typical counterflow heat exchanger process nor were they predictable, based on previous small scale direct contact experiments. The temperature profiles consistently indicated two regions of low heat transfer, one just above the brine injection point and another about midway up the column. The numerous available theoretical/empirical models of the direct contact heat transfer process did not provide any insight to this phenomenon. It appears the plateau region in the middle of the column is controlled by physical factors with the most plausible explanation being that this is a coalescing zone with large drops being formed that could exhibit poor heat transfer capability.

Attempts made to reach the maximum throughput of the column or the so-called flooding point were not successful. Predicted maximum throughput was at 110% of design flow. The maximum flow limits for the plant hardware were reached at 110% of design rates, but there was no evidence that the throughput limits of the direct contact heat exchanger had been reached. The column temperature profile at the 110% flow condition was different from that at 100% or less but it is not known if this change in temperature profile is significant or if it means that flooding limits were being approached.

2.2 WORKING FLUID RECOVERY EVALUATION

The engineering evaluation of methods for recovering working fluid from spent brine was divided into two segments, one to evaluate techniques for stripping the working fluid from the brine and a second comparing processes for separating the working fluid from the noncondensibles that are also liberated in the stripping process. Four stripping techniques and four separation processes were evaluated. Various combinations of the stripping and separation processes defined the total recovery system.

Of the stripping techniques, the vacuum flash method was the most practical and was evaluated in depth. The gas stream produced from a two-stage flash was used as the feed stream for the various separation processes evaluated. Of the separation methods, three were economically feasible. The compression and condensation approach was optimized because it appeared to have the lowest operating cost due to its simple hardware configuration.

The vacuum flash approach with compression and condensation configurations defined the total recovery system and were optimized to identify the lowest cost approach. Because of the difficulty in separating the isobutane from the noncondensibles, the lowest operating cost was achieved by eliminating the separation process and returning the noncondensibles to the power system with the recovered working fluid. This total return system was coupled with a two-stage flash stripping process. This approach turned out to be not

only cost effective but requires no sophisticated hardware. The penalty associated with the high levels of noncondensibles returned to the system is low. As optimized for the conditions of the 500 kW pilot plant, the simple recovery approach would add 1.4 mils/kW-hr to the cost of power produced.

2.3 ALTERNATE TURBINE DESIGN

Additional data was gathered and evaluated on the alternate turbine design during this last phase of testing. The alternate turbine design was aimed at improving power production efficiency for the pilot plant and incorporating an erosion resistant material in the wheel. The original turbine wheel was estimated to operate at an efficiency of approximately 72% versus the design efficiency of 83%. The alternate design was developed in an earlier phase of the program which was felt to have the potential of achieving approximately 80%.

Testing of the alternate design showed that this configuration was around 80% efficiency, close to the predicted efficiency. This testing also revealed that the gearbox losses were higher than predicted by the supplier and accounting for this higher loss shows that the original turbine wheel was at least 75% efficient.

Plant output with the revised turbine wheel was only marginally higher. The full potential of the new wheel was not achieved due to the higher gearbox losses that are present with the larger thrust bearing required in the gearbox. Further modifications to the gearbox would reduce the high losses and allow the full potential of the 80% turbine to be realized.

2.4 DOWNWELL PUMPING

A significant milestone in geothermal downwell pumps was accomplished in this last phase of testing. Operation of the 500 kW pilot plant required that the wells at East Mesa be pumped to produce the brine flow required by the plant. To pump the East Mesa wells, several Reda 80 hp submersible pumps were used with mar-

ginal success. In October 1981 the Reda pump was installed with a Barber-Nichols oil pressurization system mounted at the surface feeding the motor and keeping it pressurized with oil through a downwell feed tube. This pump operated for approximately 8000 hours without an electrical failure and the unit had to be pulled because of significant wear in the pumping stages which reduced its useful output to an unusable level. When this pump was removed from the well, electrical components, the motor protector assembly and pack-off were in excellent shape and it was estimated by Reda to be of another year's operation. The pumping stages had been eroded by sand in the brine to the point that the thrust balancing was lost and the stages were able to destroy themselves. Electrical tests on the motor indicated it was equivalent to a new, never used unit. The cable however, which was not protected, was developing some weakness.

3.0 DCHX TESTING AND EVALUATION

3.1 INTRODUCTION

Rankine or binary cycle power plants applied to low temperature geothermal resources are susceptible to fouling problems due to high concentration of dissolved solids in the brine. Studies in the mid-70's, supported by laboratory tests and a small scale feasibility test indicated that the direct contact heat exchanger approach could be applied to these Rankine cycle power plants on an economical basis (References 6, 13 and 21). To explore the direct contact approach at a reasonable level of scale as applied to geothermal energy, the Department of Energy contracted with Barber-Nichols Engineering in late 1977 to design, build and test a 500 kW pilot plant utilizing a direct contact heat exchanger (Reference 1.2).

The plant was configured to evaluate the potential of the direct contact approach and provide design criteria for large commercial plants. The exchanger designed and tested under this program is capable of 750 kW gross turbine power. This size was selected because it was felt to be large enough to demonstrate upsized scaling from the models and at the same time large enough to allow con-

fidence in scaling up to commercial-sized installations.

The specific objectives of the program regarding the direct contact heat exchanger principle were to evaluate overall heat transfer performance, correlate design parameters with the results of the small scale modeling and analysis, and provide the basis for future design of commercial-sized plants. In addition, it was felt necessary to demonstrate the stability and controllability of this type of heat exchanger.

This section summarizes the results and the analysis of the testing of the 500 kW pilot plant at East Mesa, California.

3.2 COLUMN DESIGN CONFIGURATION

A direct contact heat exchanger can be designed in several different configurations, all of which attempt to achieve the basic counterflow configuration. The more practical configurations use a working fluid which is lighter than the brine, thus allowing a single column to both preheat and boil the working fluid. In a direct column with the lower density working fluid, brine is injected at the top of the column and flows downward and out the base. The Rankine cycle working fluid is injected near the bottom and moves upward through the brine picking up heat and flashing it into a vapor before it is taken out the top of the column and sent to the expander. The column can have various types of internal configurations which vary from an open column to one that has various heat transfer enhancement devices such as trays or various forms of packing. Because the enhancement devices are subject to scaling and the open column appears to have equivalent performance, the open column was selected.

With the cycle parameters defined and the basic process flows established to achieve the net 500 kW power, the direct contact column was sized. Sizing was based on the "holdup" parameter both in establishing the minimum column diameter and the working volume required for the heat transfer. Holdup is defined as the volume fraction of the column occupied by the working fluid, which for this

system was isobutane.

The column cross section area is the limiting factor for total throughput and was sized so that the plant design point conditions would be at 90% of the predicted flooding limit. Flooding is defined as the condition or the point at which some of the working fluid cannot make its way up the column but is swept out the bottom of the column with the brine. Based on the available predictions for flooding as a function of holdup, a column diameter of 40 inches was selected.

Vessel height, or length, is based on the required volume needed to accomplish the process heat transfer. The length was calculated using parameter values from the Plass-Jacobs-Boehm heat transfer correlation (Reference 18). These were supplemented with data from the DSS tests (Reference 22) as evaluated by Dr. Jacobs from the University of Utah (Reference 6) as well as data from the University laboratory. In consultation with Dr. Jacobs, the following column dimensions were established using the correlation.

Column Diameter	40"
Column Preheater Length	27'
Column Boiling Length	2.5'
Column Superheating Length	0.5'

The lengths were based on a volumetric heat transfer coefficient of 3800 Btu/hr-ft³-°F for the preheater, 9375 Btu/hr-ft³-°F for the boiler and 1620 Btu/hr-ft³-°F for the superheating section. A detailed description of the procedure used in calculating the parameters is presented in Appendix A, Section 3.7.

The bottom end of the column is flared out conically to a larger diameter before being terminated. This conical flare, or the so-called "Elgin Tower" configuration, provides sufficient flow area in the intermediate area of the distributor to achieve low brine velocities and prevent inadvertent carryunder of the working fluid. A demister section was incorporated at the top of the column

with coalescing pads to minimize mist or droplet carryover with the working fluid to the expander. Including the conical section at the bottom and the demister section at the top, the total vessel height is 42' 3" (see Figure 3.1).

The distributor for the isobutane working fluid requires careful design to provide the proper injection velocity for the working fluid which will result in droplets of the correct size to achieve the required heat transfer in the column. The injector configuration selected is a shower head design utilizing 12,900 holes of approximately .078 inch diameter. This shower head results in an average droplet of .125 inch diameter at an injection velocity of 1.96 ft/sec. The droplet size and injection pattern were confirmed by some scale model tests at the University of Utah under the auspices of Dr. Jacobs.

Operation of the column is not sensitive to the brine injection. All that is required is that the brine be injected at the top of the column in a fairly uniform pattern across the column section and with low velocity.

The column, as designed, was to transfer 18.66 million Btu from a brine flow of 97.2×10^3 lbs/hr with an incoming temperature of approximately 340°F . The incoming isobutane flow rate is 95.4×10^3 lbs/hr injected into the column at 95°F , which would result in the brine exiting the column at 148.8°F (Reference 2).

The direct contact column was equipped with some additional mechanical features and instrumentation to aid in understanding the column's performance. Because it was felt that there might be operating problems with a large open column, the column was provided with internal tabs so that various types of internal hardware could be added to either enhance heat transfer or to control flow patterns. These internals were never needed or tested. Additionally, the column was built into equal length sections so that its total height could be reduced in the event that later testing using a working fluid such as pentane was desired.

500 KW DCHX TEST CONFIGURATION

FIGURE 3.1

Instrumentation for the column consisted of pressure and temperature transducers. A means of visually observing the physical phenomena occurring within the column was attempted but no practical means of making visual observations was devised. Some external viewports were placed at the top of the column so that observations could be made in the boiling region but they were unsatisfactory. The primary instrumentation for the column's performance consisted of a series of temperature probes placed over the length of the column which were used to establish temperature profiles and, thus, are used in evaluating heat transfer and other operating peculiarities (see Figure 3.1). The system was equipped with flow and pressure transducers at appropriate points.

3.3 TESTING CHRONOLOGY

Initial testing of the column showed that the column could be started easily and readily controlled. Performance evaluation indicated a low pinch differential and excellent heat transfer capability. The system data indicated, however, that there was a large amount of brine carryover with the working fluid exiting the column. The large amount of brine carryover resulted from improper placement of the brine distributor with respect to the brine level. Adjustment of the brine to a higher level was not possible with the existing configuration and a new brine distributor was configured and installed. Because initial testing indicated that there was more than adequate heat transfer, the new brine injector was set at a height of 24' 9" instead of the original design level of 28 feet. This setting height for the brine distributor allowed control of brine levels of several feet above the injector to evaluate the level effect on brine carryover into the rest of the system. Final setting of the brine injector with brine levels of 14 inches or higher above the injector level proved adequate in controlling the brine carryover.

The data developed on the column with the revised injector indicated that the change in working column height and the change in the injector configuration did not affect the heat transfer capability or general performance. Column heat transfer performance

was still much better than predicted. Considerable data was gathered and evaluated for this configuration both at design flow rates and lower, and at the design operating pressures and lower. Data at all of these conditions indicated, based on heat balance, that pinch points of one to three degrees were achieved and that the heat transfer was approximately 20% better than design. The temperatures being recorded, however, did not fit a typical counterflow exchange profile and showed regions of high and low transfer rates. Long term testing also confirmed an oscillation in the vapor mass flow rate exiting the column.

To further evaluate the temperature profile, heat transfer capability and the observed oscillation, a series of tests was established to vary the brine injection point and the brine injection pattern. An evaluation of the modified brine inlet distributor that was being used showed that the brine injector had a high axial velocity component that could influence the mixing at the top of the column. Tests were run on a model of the injector at the University of Utah by Dr. Jacobs and it was established that the brine injector configuration could cause backmixing for 1.5 diameters below the injection point. A new injector (injector Number 3) was designed to eliminate the strong axial velocity component and produce a low injection velocity profile that was normal to the column axis. Before the testing was started with injector Number 3, the column was opened and inspected visually to discern any plugging or scale buildup in or on the isobutane injector that could influence the injection profile of the working fluid at the bottom of the column. This visual inspection revealed a very clean column and isobutane injector with no signs of plugged injection holes or other abnormalities.

The succeeding series of tests with the Number 3 brine injector covered effective column heights from 24 feet to 16 feet or to almost one-half the original design height by resetting the brine injector and level control. The results of the testing indicate that even further reductions in column height should be evaluated. The 16-foot column was the minimum that could be tested without major hardware changes. See Figure 3.1 for the brine injector settings.

During the last series of tests, the effects of the brine level above the injector on performance were evaluated by running levels 14 to 42 inches above the brine injection point. These brine level effects were evaluated in order to ascertain that at the 16-foot column height configuration, a 42-inch brine level would not have an effect on column performance. Testing at these different effective column heights included operation at flows from 50% to 100% of design and column pressure levels of 80% to 110% of design. During these tests, a traversing temperature probe was installed at Station 12 to see if a cross-sectional temperature profile could identify channeling or large scale separation of the two media in this region of low heat transfer. Evaluation of the data obtained during this testing is covered in the succeeding sections. A test log is presented in Appendix C of Section 3.7.

3.4 TEST DATA AND COLUMN PERFORMANCE

The heat transfer performance of the column was evaluated for many data points with the Number 2 brine injector at the 24' 9" level. Testing of the column covered flow rates of approximately 50% to 100% and operating pressures of 80% and 110% of design. Within this range of operating conditions, column heat transfer performance was consistent with calculated pinch temperatures on the order of 2°F to 4°F achieved versus the 7°F used in the design of the column. With the lower pinch differential and the shortened column, the heat transfer was 20% greater than that allowed for in the initial design. Column pinch differential was calculated using a theoretical temperature profile for the column (see Figure 3.2) which is based on a heat balance. This approach to evaluating the column heat transfer neglects the temperature profile measured during performance runs.

An interpretation of the temperature profiles recorded in the column, assuming they are brine temperatures, indicates that in the first five feet and in a six-foot region in the middle of the column, little heat transfer took place (see Figure 3.3). If this 10 to 11 feet of column was not contributing its share of energy transfer, the other 14 feet was obviously doing the majority of the job.

FIGURE 3.2

FIGURE 3.3

To explore the capability of shorter columns to handle the design heat load, a series of tests was instituted in which the working height of the column was shortened in increments down to a height of 16' 3". The tests with the shortened column were performed using the Number 3 injector configuration. Tests at the 24' injector height showed no difference in either the heat transfer potential of the column or in the temperature profile with the Number 3 injector versus the Number 2 (compare Figures 3.3 and 3.17). At all of the tested heights, column flows and pressures were changed to explore their effect on the column's behavior. The results of this testing was presented chronologically in Figures 3.4 through 3.33.

The characteristics of the temperature profile were consistent as the column height was changed. The regions of "low heat transfer" during the first five feet in the region of 9 to 15 feet were not affected. Figure 3.32 shows temperature profiles with the 24-foot and the 16-foot tests at equivalent conditions. Evaluation of the actual heat transfer at these conditions indicates no loss in capacity and no indication that the minimum pinch differential was affected. The plot Figure 3.34 shows a matrix of column heat transfer data recorded for tests on the column from a height of 24 feet down to the 16-foot level with some of the highest heat transfer values recorded for the tests at the 16-foot height. This data has not been corrected or adjusted for the amount of superheat being produced. The data for the 16-foot height was run with approximately 15° of superheat which is 7° to 8° higher than most of the data shown. It is felt that the column could have been shortened even more but limitations on the hardware prevented testing at reduced heights. In going from a design height of 27 feet to 16 feet, and with the lower pinch differential, the effective column volumetric heat transfer coefficients are at least two times that predicted by earlier experiments and the Plass-Jacobs-Boehms correlation.

The temperature profile of the column is of considerable interest. The profile shown is not one that would be predicted based on a typical counterflow heat exchange process nor is it predicted by the models developed to date with the small scale direct contact

FIGURE 3.4

FIGURE 3.5

TEMPERATURE °F

JAN 25 1982

BRINE DISTRIBUTOR 22' 1"

BRINE LEVEL 24' 5"

Symbol	Time	Brine Flow GPM	IC Liquid Flow GPM	Brine Inlet Temp °F	IC Inlet Temp °F	Column Pressure PSIG
○	13:31	216.7	348.7	332.8	85.6	435.5
□	13:01	213.6	343.1	333.4	84.3	433.8
△	12:31	215.8	345.9	331.9	85.9	433.5

FIGURE 3.6

FIGURE 3.7

FIGURE 3.8

FIGURE 3.9

FIGURE 3.10

FIGURE 3.11

FIGURE 3.12

FIGURE 3.13

FIGURE 3.14

FIGURE 3.15

FIGURE 3.16

MARCH 3, 1982

BRINE DISTRIBUTOR LEVEL 24' 1"

BRINE LEVEL 25' 11"

Symbol	Time	Brine Flow GPM	IC Liquid Flow GPM	Brine Inlet Temp °F	IC Inlet Temp °F	Column Pressure PSIG
●	19:32	207.5	313.4	337.6	89.5	435.6
▲	20:32	208.7	312.8	333.4	88.5	435.5
■	21:32	205.4	314.5	334.6	88.4	434.7

FIGURE 3.17

FIGURE 3.18

JUNE 4, 1987

BRINE DISTRIBUTOR 21.5"

BRINE LEVEL 25.5"

Symbol	Time	Brine Flow GPM	IC ₄ Liquid Flow GPM	Brine Inlet Temp °F	IC ₄ Inlet Temp °F	Column Pressure PSIG
●	20:31	217.4	334.4	337	90.9	434.4
Δ	19:01	216.2	333.2	337	92.8	433.6
□	17:31	219.1	332.5	337	94.5	434.9
◇	16:01	218.8	336.5	336	94.1	434.4

FIGURE 3.19

FIGURE 3.20

JUNE 8, 1982

BRINE DISTRIBUTOR 21-5"

BRINE LEVEL 28-3"

Symbol	Time	Brine Flow GPM	IC Liquid Flow GPM	Brine Inlet Temp °F	IC ₄ Inlet Temp °F	Column Pressure PSIG
●	10:01	174.2	293.0	336	85.3	378.6
△	16:01	177.2	298.7	330	88.0	397.7
□	22:01	176.5	289.3	337	85.5	398.1
◇	23:31	175.4	291.0	337	85.4	378.7

FIGURE 3.21

FIGURE 3.22

JUNE 16, 1982

BRINE DISTRIBUTOR 21.5"

BRINE LEVEL 25.1"

Symbol	Time	Brine Flow GPM	IC ₄ Liquid Flow GPM	Brine Inlet Temp °F	IC ₄ Inlet Temp °F	Column Pressure PSIG
●	9:01	137.7	222.1	334	90	302.5
▲	11:01	178.8	273.1	336	93	432.7
□	16:31	177.8	272.9	337	92.7	433.5
◊	17:31	177.2	274.1	356	88.7	433.2

FIGURE 3.23

JUNE 17, 1962

BRINE DISTRIBUTOR 21'5"

BRINE LEVEL 25'11"

Symbol	Time	Brine Flow GPM	IC ₁ Liquid Flow GPM	Brine Inlet Temp °F	IC ₁ Inlet Temp °F	Column Pressure PSIG
●	19:31	216.6	337.0	337	97.3	433.2
▲	21:31	214.9	334.0	336	95.0	432.8
□	22:31	214.7	337.3	337	93.9	434.9
○	23:31	214.4	337.2	337	93.4	432.1

FIGURE 3.24

FIGURE 3.25

FIGURE 3.26

FIGURE 3.27

SEPTEMBER 22, 1982

BRINE DISTRIBUTOR 16' 3"
BRINE LEVEL 20' 11"

Symbol	Time	Brine Flow GPM	IC Liquid Flow GPM	Brine Inlet Temp °F	IC ₄ Inlet Temp °F	Column Pressure PSIG
●	18:01	145.9	207.1	333.7	98.6	443.3
△	17:46	146.9	205.6	333.9	98.5	444.0
□	17:16	144.6	207.0	333.6	98.3	444.3
◇	16:46	145.2	204.4	333.9	97.6	443.6

FIGURE 3 28

SEPTEMBER 22, 23, 1982

BRINE DISTRIBUTOR 16' 3"
BRINE LEVEL 20' 11"

Symbol	Time	Brine Flow GPM	IC Liquid Flow GPM	Brine Inlet Temp °F	IC Inlet Temp °F	Column Pressure PSIG
●	2:16	139.5	163.6	333.1	95.2	446.8
▲	1:31	135.5	160.5	333.0	94.3	444.6
□	0:16	142.6	163.6	333.9	94.1	444.0
◊	23:01	141.3	160.3	333.6	93.5	445.5

FIGURE 3.29

Symbol	Time	Brine Flow GPM	IC ₄ Liquid Flow GPM	Brine Inlet Temp °F	IC ₄ Inlet Temp °F	Column Pressure PSIG
●	6:46	136.6	100.0	333.0	90.4	444.0
△	6:31	134.8	101.7	332.7	90.6	443.5
□	5:46	141.6	104.0	333.3	89.5	445.4
◇	4:46	129.3	102.0	333.5	89.8	444.9

FIGURE 3.30

FIGURE 3.31

FIGURE 3.32

OCTOBER 20, 1982
 BRINE DISTRIBUTOR LEVEL 19' 3"
 BRINE LEVEL 20' 11"

Symbol	Time	Brine Flow GPM	IC Liquid Flow GPM	Brine Inlet Temp °F	IC ₄ Inlet Temp °F	Column Pressure PSIG
○	9:46	242.0	375.7	344.2	87.1	402.7
△	10:46	239.5	380.4	343.9	87.1	402.0
□	11:30	242.5	386.4	343.8	87.9	402.3
◇	12:30	238.7	374.1	343.7	88.6	402.0
•	13:30	241.9	374.7	343.8	87.6	402.2

FIGURE 3.33

DCHX PERFORMANCE WITH DIFFERENT BRINE DISTRIBUTOR LEVELS AND FLOW RATES

FIGURE 3.34

experiments. The temperature profile indicates two regions of low heat transfer (assuming it represents brine temperature), one right above the injector and the other one approximately midway up the column. These regions of low heat transfer appear consistently in the many runs that were made. During transient conditions, either as a result of startup or due to resetting the column operating point, the temperature profile can vary with some profiles having very interesting shapes. The sensors and the data system were changed and/or calibrated many times to verify that the peculiar profiles were not caused by faulty instrumentation.

Long duration tests were run with constant conditions to establish the potential for the system to have a preferred profile in terms of these plateau regions. The two consistent profiles involved can be seen by looking at Figures 3.19 and 3.7. Figure 3.19 differs from Figure 3.7 only in that at the 11-foot level the temperatures are sometimes at the plateau level and other times indicate heat transfer. Figure 3.7 is the most typical profile.

Figures 3.15, 3.16 and 3.17 show the results of a long-term test in which the plateau region temperature diminished slowly over a 30° range in a period of seven hours without any change in controlling parameters such as flow rate, input temperatures and the flow ratio. The consistency of the input can be recognized by observing that the output temperatures are well within the expected data scatter.

Figures 3.26 and 3.27 show two extremes of the temperature profile as a function of time with steady input conditions. Figure 3.26 shows the temperature profile that includes the startup transient and a full hour period where the profile is still changing. The startup is completed in the first 30 minutes shown with the final flow settings being established. After the first 30 minutes, the column exit temperatures have stabilized, even though the profile is fluctuating considerably. The constant exit temperature indicates that the overall heat transfer is constant. Figure 3.27 is a histogram in which the column has been sitting stable for hours and shows an 8-hour period with relatively small fluctuations in the

temperature at all locations.

The capability of the plant to run at different flow ratios, particularly ones that would allow operation at higher amounts of superheat, was limited and, within this small range of flow ratio, the testing did not indicate that it had a correlatable effect on the temperatures in the plateau region.

The temperature probes used in the column were RTD's with small time constants. The initial set of probes were installed with the probe being inserted approximately 12 inches from the wall. Twenty-four inch probes were installed in selected locations to see if there were any signs of stratification or separation of the fluid within the column. The data showed identical profiles with either set of probes. A traversing probe was installed at Station 12 and scans were made across the diameter of the column. The traversing probe showed no signs that separation or stratification was occurring in this plateau region as the temperature across the section did not vary over $.5^{\circ}$. There is no way to establish whether the probes are measuring the temperature of the continuous phase or an average temperature between the continuous and dispersed phase (IC_4). It is generally believed, based on small scale experience, that the probes are primarily measuring temperatures of the continuous phase or the brine flowing down the column.

During long duration runs, with the Number 2 injector, it was established that the column output would swing approximately $\pm 10\%$ over a 10-minute period with an oscillation that was periodic in character. These oscillations occurred with very stable inlet flows to the column and with a very stable column pressure. The output vapor flow fluctuation was reflected in alternator output which swung the same 10%. The change in injector configuration from Number 2 to Number 3, which eliminated the high downward velocity component from the injector eliminated this oscillation. Any indicated oscillations with the Number 3 injector were very random in nature and were less than $\pm 2\%$. When the oscillating flow was observed in the column with the Number 2 injector, the temperatures in the column and additionally the exit temperature of the working

fluid did not show any symptomatic variations that would tie in with this measured output flow oscillation.

Controllability of the column is good. Once the desired operating point was reached, the column maintained this condition with very little adjustment to the input flows. It was possible to set the flows to the column and set the column pressure and have the column run for many hours without any additional adjustments being made and with a very consistent output.

Testing of the column with high brine levels above the distributor did not appear to affect heat transfer or temperature profiles (see Figures 3.22, 3.23 and 3.24). These high levels did appear to make the system more sluggish in terms of showing response of outlet temperatures to small changes in the mass flow ratio of the working fluid and brine.

Attempts were made to reach or find the maximum throughput limit of the column which would be exhibited by excess carryout of the working fluid in the brine leaving the column. These attempts were made with the brine injector set at the 19' 3" level. The plant operating conditions were adjusted to achieve the maximum brine flow. Column operation pressure was lowered to 400 psig and the sand trap pressure raised to 175 psig. The maximum brine flow achieved was 240 gpm, which is approximately 111% of design. Brine sampling and GC analysis showed the IC_4 levels to be normal in the brine leaving the column, thus indicating that we did not reach the throughput limit. At these high flow rates, the lower five feet of the column shows a large temperature change, but this was the only measured effect (see Figure 3.33).

3.5 PERFORMANCE ANALYSES AND MODELING

The numerous theoretical/empirical models of the direct contact heat transfer process available in the literature were explored with the objective of finding techniques to predict the measured performance of this large scale direct contact column which would explain the temperature profile characteristics observed. Considerable ex-

perimentation and analysis have been done on the direct contact process, but all of it has been concentrated on small scale experiments and ones in which the temperature differences and heat transfer are limited to small values to facilitate analysis. Much of the work to date in this area has been examined and critiqued by Dr. Jacobs, et. al. as documented in References 6 and 14. Based on model tests and compilation of data from various sources, Dr. Jacobs feels that the most promising model for predicting performance in the direct contact process was developed by Letan and Kehat (References 8, 9 and 10).

The basic approach in all the modeling is to assume that a spherical drop is moving steadily under buoyancy or gravity in an infinite body of noncompressible fluid which is otherwise undisturbed. Starting with this assumption, equations are developed to solve the problems of motion, which are then combined with heat transfer equations. Based on the equations of motion, heat transfer has been calculated using equations which give Nusselt numbers that are a function of the drop Reynold's and Prandtl numbers. The shortcoming of this process is that heat transfer and physio chemical processes around the drops not only depend on the velocity field equations of motion, but they also influence the velocity fields. Internal circulation in the drops, oscillation of the drop shape, drop coalescent and breakup, axial dispersion and vorticity associated with the drops all have a strong effect on the heat transfer coefficient.

The analyses are all based on spherical drops that are very small in diameter. Larger diameter drops change their shape from the spherical and eventually oscillation of the drop shape begins, leading to velocities substantially lower than those for spherical drops (References 7 and 24). Drop sizes cannot be estimated accurately since drop formation is influenced by variables such as the degree of turbulence, roughness or deposits of dirt in the nozzle and the flow of continuous phase fluid around the nozzle. The estimate of drop size is essentially an average drop size. In reality, the drops will have a size spectrum with a statistical distribution as discussed in Reference 23. In the 500 kW column tested, there

is sufficient time and length for the drops to coalesce and then break up which could affect velocity and heat transfer distribution in this direct contact heat exchanger.

A mathematical model of the true process is too complex to solve readily and there is presently no theoretical model available which can describe the total heat transfer problem in a large scale direct contact heat exchanger. The theoretical work that has been done to date on the direct contact process is too restricted with simplifying assumptions and ideal conditions with very small dimension compared to the 500 kW direct contact unit.

Jacobs, et. al. did extensive work in evaluating published data and data generated at the University of Utah on the direct contact concepts. The compilation of all this data indicated that the Plass-Jacobs-Boehm correlation could be used for initial sizing of a direct contact open-type column and that further refinement of heat transfer calculations could be achieved by using the Letan-Kehat model (Reference 6). The initial sizing of this column was done using the Plass-Jacobs-Boehm correlation which predicted a volumetric heat transfer coefficient in the preheating zone of somewhere around $3800 \text{ Btu/hr-ft}^3\text{-}^\circ\text{F}$ for the design conditions demonstrated. Based on log-mean temperature differences calculated from the heat balance, the coefficients achieved by the column ranged from 2500 to over 7000. The low values were achieved when the full 24-foot effective column was being used and the high values were achieved when the column was shortened to 16 feet. If one neglects the volume contained in the column in the first five feet where little heat transfer occurs, the 16-foot column tests give an average coefficient in the preheating region of over $10,000 \text{ Btu/hr-ft}^3\text{-}^\circ\text{F}$. None of the small scale experiments produced heat transfer coefficients that even began to approach these numbers.

Temperature profiles for the 24-foot and 16-foot column heights were examined with respect to the Letan-Kehat coefficient to establish the approximate values that are required for the measured profiles. The 24-foot and 16-foot profiles examined were typical and at design flow rates. The profiles analyzed are shown in Figures

3.13 and 3.31. The approach used to develop the coefficients was iterative in that the coefficient M was assumed and then the coefficient m was varied to set end conditions for each distinct slope or zone observed in the profile. The values of M and m recorded in Figures 3.35 and 3.36 give a reasonable fit to the recorded profiles for the distinct zones that occur in the column. Since the values of M and m are not completely independent, some tolerance is involved. The significant fact observed from matching the column profiles is that the coefficient values obtained are significantly different than those presented by Letan-Kehat and that there is no possibility of obtaining a coefficient that matches the total column profile.

An attempt was made, using the recorded temperatures in the column, to create a reasonable explanation for the profiles by assuming that below the mid-region plateau the temperatures being measured reflected the continuous media or brine temperatures, whereas above the plateau the recorded temperatures reflected the dispersed phase of the working fluid temperature. Based on these assumptions, the matching profiles for the other medium were calculated based on heat balance. This approach did not produce reasonable temperature profiles for either media across the length of the column. The first five feet of column are also interesting from the standpoint of modeling. Based on the experimental models, the first five feet should show a high rate of temperature change, which it does not do, and in this first five feet the effective heat transfer coefficient is perhaps only in the 50 to 100 Btu/hr-ft³-°F range. It is obvious, looking at these facts, that it is very difficult to do anything with the Plass-Jacobs-Boehm correlation or the other models in terms of predicting the heat maximum transfer capability of a large direct contact column.

There undoubtedly exist excellent explanations for the observed data, but in all the references cited we were unable to find any combination of techniques for modeling the flow that could be used to explain the measured performance. For example, if the Letan-Kehat coefficients are adjusted to fit heat transfer observed in the first five feet, which is very low, the model cannot then predict the

FIGURE 3.35

FIGURE 3.36

heat transfer for the region above this. The low heat transfer zone at the bottom of the column could be the result of injection velocities that are too high, which delays the formation of drops until the jet has penetrated a considerable distance into the continuous media. Based on References 5, 7 and 19, velocities above .85 ft/sec would result in jets above the injection surface instead of immediate drop formation. The calculated design velocity for injections is 1.96 ft/sec. Tests at low flows which drop the injection velocity below .85 ft/sec do not change the profile (Figures 3.14, 3.21 and 3.30). The tests at higher than design flow rate (Figure 3.31) were the only ones that showed an influence on the heat transfer or profile in this first five feet, so obviously the normal effects of injection velocity and drop formation are not involved.

None of the experimental small scale work shows the temperature plateau in the middle of the heat transfer zone that has been observed with this large column. The temperature plateau always occurs at the same region of the column although at various temperatures. The location of this region was not influenced directly by any of the parameter changes such as flow, flow ratio, column operating pressure or shortening the working column height. Physically this column is much larger than any other test column that has been recorded in literature. Based on the fact that this plateau appears to be controlled by physical factors, the most plausible explanation is that this region is a coalescing zone with very large drops being formed (References 3, 15 and 16). The large drops could reduce the heat transfer observed. The large drops will start oscillating and break up into small drops after this coalescing region (References 4 and 17). After the small drops reform and disperse, the higher heat transfer capability of the column would again exist.

The data on the column performance was supplied to Dr. Jacobs for his review and interpretation. Due to limitations in the quantity of instrumentation and uncertainty as to which media or what sort of average temperature is reflected in the column temperature measurements, Dr. Jacobs could not come to any definitive conclu-

sions regarding modeling techniques for a large column. Dr. Jacobs did provide a report with his interpretation of the phenomena observed in the data and recommendations for further research. His report is included as Appendix B in Section 3.7.

Without the ability to model this large size direct contact heat exchanger, it is difficult to predict how other large units would perform. The tests run prove conclusively that much higher heat transfer rates can be achieved than predicted by the models and the small scale tests. There is no indication that any of the earlier experimenters tried to establish the heat transfer limits of their experimental setups by shortening their columns. Since their configurations were all similarly scaled, they may have missed the fact that the working volume being used is a dependent variable in the heat transfer as is indicated by the change in temperature slope between the five and nine foot levels in the column when the working column height changes from 16 to 24 feet. Based on the data sample of one large column, it is felt that a large direct contact heat exchanger can be designed using the Plass-Jacobs-Boehm correlation and that these coefficients would give a conservative configuration that would meet design goals. It is recommended that any further experimental work done in the direct contact field be aimed at identifying the physical phenomenon taking place in these long columns so that their performance can be understood and later modeled.

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3.7 APPENDIX A - PROCEDURE FOR SIZING DCHX VESSEL (Do the procedure once per region in DCHX (i.e., preheat, boil, superheat))

Step 1- Identify known input parameters from cycle studies and fluid properties.

$C_{p_{\text{brine}}}$	Btu/lb - °F
$\dot{M}_{\text{brine}}$	lb/hr
$\dot{M}_{\text{IC}_4}$	lb/hr
$T_{\text{brine in}}$	°F
$T_{\text{brine out}}$	°F
$\rho_{\text{brine avg}}$	lb/ft ³
μ_{brine}	lb/ft-sec
$T_{\text{IC}_4 \text{ in}}$	°F
$T_{\text{IC}_4 \text{ out}}$	°F
$\rho_{\text{IC}_4 \text{ avg}}$	lb/ft ³
lmtd	°F

Step 2- Identify assumptions and recommendations from earlier investigations.

- a) Column Reynold's number must indicate laminar flow (i.e. $Re_c \leq 2300$).
- b) Maximum droplet diameter = 4 mm
- c) Holdup (design) = 90% of holdup at flooding.
- d) Aspect ratio (l/d) ≥ 10 .

Step 3- Calculate terminal drop velocity from:

$$f V_t^2 = \frac{4}{3} d_p g \left(\frac{\rho_{\text{brine}} - \rho_{\text{IC}_4}}{\rho_{\text{brine}}} \right)$$

V_t = terminal drop velocity in ft/sec.
 f = friction factor calculated below
 d_p = droplet diameter, ft.
 g = gravitational constant, ft/sec²
 ρ_{brine} = brine density from step 1, lb/ft³
 ρ_{IC_4} = IC₄ density from step 1, lb/ft³

Step 4- Calculate drop Reynold's number from:

$$Re_o = \frac{V_T d_p \rho_{\text{brine}}}{\mu_{\text{brine}}}$$

V_T = terminal drop velocity from step 3
 d_p = maximum drop diameter, ft.
 ρ_{brine} = brine density, step 1, lb/ft³
 μ_{brine} = brine kinematic viscosity, lb/ft-sec.

Step 5- Determine f from Reynold's number:

$$f = 24/Re_o \quad \text{if } Re_o < 0.1$$

$$f = 18.5/Re_o^{0.6} \quad \text{if } 2 < Re_o < 500$$

$$f = 0.44 \quad \text{if } 500 < Re_o < 2 \times 10^5$$

$$f = 0.2 \quad \text{if } Re_o > 2 \times 10^5$$

Repeat steps 3, 4 and 5 until there is no longer any change in V_T .

Step 6- Determine m from:

$$m = 3.65 \quad \text{if } Re_o < 0.2$$

$$m = 4.35 \text{ Re}_o^{-0.03} - 1 \quad \text{if } 0.2 < \text{Re}_o < 1$$

$$m = 4.45 \text{ Re}_o^{-0.1} - 1 \quad \text{if } 1 < \text{Re}_o < 500$$

$$m = 1.39 \quad \text{if } \text{Re}_o \geq 500$$

Step 7- Calculate flow rate ratio:

$$R = \frac{\dot{m}_{\text{IC}_4} (\rho_{\text{brine}} / \rho_{\text{IC}_4})}{\dot{m}_{\text{brine}} - \left(\frac{\dot{m}_{\text{IC}_4}}{m} (1-m) \right)}$$

R = flow rate ratio

$\dot{M}_{\text{IC}_4}$ = mass flow rate IC_4 , lb/hr from step 1

$\dot{M}_{\text{brine}}$ = mass flow rate brine, lb/hr from step 1

ρ_{brine} = brine density, lb/ft³ from step 1

ρ_{IC_4} = IC_4 density, lb/ft³ from step 1

m from step 6

Step 8- Calculate flooding holdup, ϕ_f :

$$\phi_f = \frac{-(m+2)R + \sqrt{(m+2)^2 R^2 + R(m+1)(1-R)4}}{2(m+1)(1-R)}$$

m from step 6

R = flow rate ratio, step 7

Step 9- Calculate design holdup, ϕ :

$$\phi = 0.9 \phi_f$$

Step 10- Calculate brine superficial velocity:

$$V_c = \frac{V_T \phi (1-\phi)^{m+1}}{R(1-\phi) + \phi}$$

V_T = terminal drop velocity, ft/sec, step 3

ϕ = design holdup, step 9
 m from step 6
 R = flow ratio from step 7

Step 11- Calculate vessel diameter:

$$V_c = \frac{4 \left(\dot{m}_{\text{brine}} - \left(\frac{\dot{m}_{\text{IC}_4}}{m} (1 - m) \right) \right)}{\rho_{\text{brine}} (3600) \pi \left(\frac{D}{12} \right)^2}$$

V_c = brine superficial velocity, ft/sec, step 9
 $\dot{m}_{\text{brine}}$ = brine mass flow rate, lb/hr, step 1
 $\dot{m}_{\text{IC}_4}$ = IC_4 mass flow rate, lb/hr, step 1
 m from step 6
 ρ_{brine} = brine density from step 1, lb/ft³

Step 12- Calculate volumetric heat transfer coefficient:

$$U_v = \begin{cases} \phi 12000 & \text{if } \phi \leq 0.05 \\ 45000 (\phi - 0.05) \exp(-0.57R) + 600 & \text{if } \phi > 0.05 \end{cases}$$

Step 13- Calculate length from:

$$L = \frac{[V_c \rho_{\text{brine}} C_{p\text{brine}} (T_{\text{brine in}} - T_{\text{brine out}})]}{U_v \ln \Delta T}$$

V_c = superficial brine velocity, ft/sec, step 9
 ρ_{brine} = brine density, lb/ft³, step 1
 $C_{p\text{brine}}$ = brine specific heat capacity, Btu/lb-°F, step 1
 $T_{\text{brine in}}$ = brine temp. (inlet), °F, step 1
 $T_{\text{brine out}}$ = brine temp. (out), °F, step 1

U_v = volumetric heat transfer coefficient,
Btu/lb-ft³°F, step 12

lmtd = log mean temperature difference, °F,
step 1

3.7 APPENDIX B

REVIEW OF THE PERFORMANCE
OF THE DIRECT CONTACT HEAT EXCHANGER
FOR THE 500 KWE DIRECT CONTACT
GEOTHERMAL POWER PLANT
AT
EAST MESA, CALIFORNIA

CONDUCTED FOR BARBER-NICHOLS ENGINEERING
AND THE
UNITED STATES DEPARTMENT OF ENERGY
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INTRODUCTION

The concept of a binary power cycle utilizing direct contact heat exchangers was first proposed by Jacobs and Boehm in 1973⁽¹⁾ for use with geothermal brines. This concept was proposed primarily to overcome difficulties associated with the fouling and scaling nature of many moderate temperature brines. However, thermodynamic analyses and subsequent economic analyses clearly point to the possible economic advantages over conventional binary cycles even with non-fouling, non-scaling brines.

For a direct contact binary power plant to be economically attractive it is necessary that a small pinch point be obtainable so that a maximum amount of power can be obtained per unit mass flow of geothermal brine. Since the working fluid comes in direct contact with the brine it must be immiscible with the brine, low in cost, and, if part of it goes into solution in the brine, easily recoverable. The 500 kWe direct contact heat exchanger test facility installed at East Mesa was originally designed to evaluate techniques to insure economical operation.

The choice of the East Mesa site as a first location to evaluate the direct contact heat exchanger system placed additional requirements on the system. The brine at East Mesa was at low pressure requiring the use of downhole pumps. The selection of isobutane as a working fluid required increasing the pressure of the brine. The high amount of dissolved CO_2 in the brine required that it be preflashed to prevent the carryover of CO_2 gas through the turbine and into the condenser which would adversely affect the system performance. All of these problems have been admirably met by the system designer and operator, Barber-Nichols Engineering. Further, problems with isobutane turbine design, supposed state-of-the-art, were encountered and resolved. Thus, most of the objectives and targets of opportunity associated with operating a direct contact heat

exchanger geothermal binary power plant at East Mesa were encountered and solved.

As has been pointed out in reports by Barber-Nichols Engineering to the U.S. D.O.E., the direct contact heat exchanger met, with minimum modification of the system, all the design requirements. In fact, it appears, upon evaluation of the data in the Barber-Nichols August 1983 report, that the direct contact heat exchanger was conservatively designed. The degree of conservativeness in the design is difficult to ascertain, especially, in view of the limited instrumentation available on the direct contact heat exchanger. Further, it is not certain that the information that has been obtained has been properly interpreted.

The purpose of this report is to shed some light on the interpretation of the data. Due to insufficient data, much of what follows in this report is speculative and merely points out further investigations that may be necessary if less conservative design practices are to be achieved than were used in the design of the East Mesa direct contact heat exchanger.

DIRECT CONTACT HEAT EXCHANGER TEST EVALUATION REVIEW

The direct contact heat exchanger at East Mesa was designed primarily using the procedures outlined in Section 4.2 of the "Sourcebook on the Production of Electricity from Geothermal Brines"⁽¹⁾. The exception of this was in finding the length of the preheater. The length of the preheater was determined utilizing the following correlation for the volumetric heat transfer coefficient:

$$U_v = 45,000 (\Phi - 0.05) e^{-0.57R} + 600 \frac{\text{Btu}}{\text{hr ft}^3 \text{ } ^\circ\text{F}} \quad (1)$$

This correlation was published by S. B. Plass, H. R. Jacobs and R. F. Boehm in the paper "Operational Characteristics of a Spray Column Type Direct Contact Preheater" in the AIChE Symposium No. 189, Vol. 75, Heat Transfer - San Diego, 1979⁽²⁾. The correlation is dependent upon the holdup, Φ and the ratio of volumetric flow rates,

$$R = \frac{\text{volume flow rate dispersed}}{\text{volume flow rate continuous}}$$

The correlation was based upon new data acquired in a six inch diameter spray column using insulating oil as the distributed phase and water as the continuous phase and all of the pre-existing data in the literature including that of DSS Engineers using isobutane as the distributed phase and East Mesa brine as the continuous phase.

A model developed by Letan and coworkers ^(3,4) based on the observed physics of the problem was not utilized since it was limited to application to constant property fluids. This model states that the heat transfer over most of the length of a column is governed by the hydrodynamics of the drops of the dispersed phase. These characteristics are described by the wake shedding frequency of the drops which are dependent upon drop rise velocity, drop spacing, etc., all characteristics that can be related to the ratio of volumetric flow rates and holdup. For constant properties for the two phases, the Letan model predicts a constant value of the volumetric heat transfer coefficient. Appropriate values of two experimental parameters in the Letan model yield essentially the correlation given in Equation (1).

The design criteria used in designing the direct contact heat exchanger preheater section was based upon laminar considerations. Essentially the column is designed for disperse packing of the drops, that is, the distance between drops is much larger than their diameter. However, a dense packing of drops is possible. This can lead to modification of drop wake size and drop shedding frequency. This dense packing normally would increase the heat transfer; however, it is difficult to predict. Dense packing of drops can also lead to conglomeration of the drops resulting in drops of variable size and thus variable drag and resultant variable holdup and drop rise velocity. This again has not proven predictable. If considerable conglomeration occurs, it is possible for the dispersed phase to become the continuous phase. This fact was demonstrated to B/N and U.S. D.O.E. personnel at a visit to the University of Utah in 1979. Using kerosene as the dispersed phase at the bottom of the

column and water as the continuous phase, it was possible at hold-ups near the flooding condition to operate in a steady-state mode where the water was the dispersed phase in the top half of the column and kerosene the dispersed phase in the bottom half of the column. With all of the above facts in mind, we can examine the data acquired by B/N for operation of the direct contact heat exchanger for the 500 kWe facility.

The direct contact heat exchanger is shown schematically, in Figure 1, based on its original design proposed in September 1978 by Barber-Nichols Engineering. This design was essentially the final configuration. It was modified in terms of the brine injector to one where the brine was injected through the top of the vessel. By lengthening the pipe to the injection nozzle, the point along the column where the brine was injected could be altered. In addition, within reason, the level control could be adjusted by relocating the controller as shown. If the top spool of the column were reversed top for bottom, additional shortening of the fluid height could be obtained. However, the latter was not done due to the costs involved. A further shortening of the column could be achieved by removing the bottom spool. This also was not accomplished.

The isobutane injection nozzle was designed as a single plate as is common practice in small diameter Elgin spray towers. The exhausting brine had to make its way around the isobutane injection nozzle. This could well lead to a dead, no-brine flow region directly above the isobutane injector with higher velocities surrounding it. The flared Elgin tower bottom is designed to prevent unduly high brine velocities in the annular region surrounding the injector. The success of this design is clearly pointed out by the low carry-under of isobutane in the discharged brine. What effect could the design of the bottom of the column have had on the heat transfer? This question can be answered in part by examining Figure 2. The brine flowing down the tower at very low speed will sense the obstruction of the injector and must speed up and move annularly around it. Thus a dead flow region must exist within the dashed lines. As there is no resistance due to a downward flow of water in this center

Figure 1 - 500 kw DCHX

Figure 2

region, the drops of isobutane will rise faster over the center of the injector plate. The drops at the periphery of the injector can rise straight up into the high flow annular region or more toward the center of the column, thus causing a necking down of the region through which the drops rise. The drops rising through the stagnant water will only be slightly heated due to radial conduction through the water until they move into the moving brine. Depending on the extent of this brine region in the vertical direction, the heat transfer could be reduced. The design of the Elgin tower base is supposed to be such as to significantly reduce this effect. It appears successful in small diameter columns; however, no data exists to confirm this fact for large columns. The only data is that of the temperature measurements in the direct contact heat exchanger at East Mesa; however, it is unknown whether the temperatures are that of the continuous phase or the dispersed phase or some sort of average. In addition, the first temperature measurement is at five feet above the distributor plate. In order to determine whether the heat transfer is low in this region would require additional temperature probes in this region and the installation of a multiple injection head manifold which would allow for some flow directly downward eliminating or vastly decreasing the height of stagnant water regions. Such an injection nozzle has been designed for the Solar Energy Research Institute (SERI) two-foot diameter direct contact column at the University of Utah. (This column could more easily be modified than the East Mesa direct contact heat exchanger and is available after March 1984 for testing).

Figure 3 illustrates a typical set of data from 11:01 A.M., November 12, 1980. For this experiment B/N reports two sets of measurements of the flow rates of the brine and IC_4 . For the brine, the two flow measurements indicate 76,231 lb/hr and 85,840 lb/hr while the IC_4 measurements are 91,434 lb/hr and 90,989 lb/hr. Although the two IC_4 measurements are within 0.5% of each other, the difference in the two brine flows are more than 10%. This error makes it quite difficult, i.e., impossible, to carry out an accurate heat balance. Assuming that the IC_4 measurements are accurate at the top of the column as well as assuming the column pressure and vapor exit temperatures are accurate, allows one to calculate the energy

Figure 3. Temperature observed in the DUX
12 Nov 1980 (11:01 AM)

given up by the brine to vaporize the IC_4 , superheat it and to evaporate a small portion of the water in the brine. Using these numbers to calculate the entrance temperature of the brine to the preheater section of the direct contact heat exchanger indicates that the lower, orifice-measured, flow rate of the brine would lead to temperatures below the saturation temperature of the brine. Thus, the higher flow rate should be used and it has been generally accepted as accurate by Barber-Nichols in their calculations. However, it is believed that experimental error of at least 1-2% can exist with all flow measurements, as repeated regular calibrations were not carried out. The temperatures recorded by the resistance probes are also subject to error. Calibration in the environment used was not made and the instruments are at best accurate to a few degrees. Consider, for example, the temperatures recorded by the upper resistance thermometers, variations from 248.7 to 251.2°F were recorded. Although they might indicate eddies in the flow, their slow response time of several seconds would indicate that their accuracy was at best $\pm 2^\circ\text{F}$. Thus, predictions based upon their measurement more accurate than this should not be made. Typically, the pressure measurement instruments used were also accurate to only within 1-2% of full scale and are thus reliable only to within $\pm 3-4$ psi. This also can affect the accuracy of any predictions made.

With the above facts in mind, let us examine Figure 3. On this figure, the experimental temperatures for the experiment at 11:01 A.M., November 12, 1980 are shown as previously mentioned. The calculated preheater entrance temperature is 251°F. The IC_4 boiling point is calculated to be 244.5°F. If we assume constant heat transfer per unit length of the preheater and assume the brines temperature unchanged from the IC_4 distributor to the exit, the temperatures of the brine and IC_4 would follow the large dash-dashed lines shown. The measured column temperatures generally lie between these two lines. At the bottom of the column, the temperatures are half way between the two lines; however, at about 12 ft above the IC_4 distributor, the observed temperatures shift over to the line repre-

senting the IC_4 and stay there until the vapor exits the column. One interpretation of the data might be that at about this level, the approximate middle of the column, there is a shift from the brine being the continuous phase to the IC_4 being the continuous phase. Since IC_4 liquid has a much lower thermal conductivity than water, it could perhaps insulate the probes from the water. The volumetric flow rate of IC_4 is, at entrance, 327.9 gpm as compared to 192.0 gpm of brine. This scenario is highly possible as the holdup is close to flooding at about this point in the column. It has been observed at the University of Utah and elsewhere that equilibrium operation of a column can occur with half of the column having one fluid as the continuous phase and the other half having the other fluid as the continuous phase.

For the experiment cited, knowledge of the IC_4 and brine temperatures entering and exiting the preheater, its length and diameter and the fluid properties allow us to calculate the preheater U_v for the column. Given $Q_{\text{preheat}} = 10.477 \times 10^6$ Btu/hr, the preheater length of 23 feet, diameter of 40 inches and the temperatures shown on Figure 3,

$$U_v = \frac{Q}{\frac{\pi D^2 L}{4} \text{LMTD}} = \frac{10.477 \times 10^6 \text{ Btu/hr}}{\frac{\pi (40)^2 \text{ ft}^2 (23 \text{ ft}) (21.8 \text{ }^\circ\text{F})}{4}} = 2390 \text{ Btu/hr ft}^3 \text{ }^\circ\text{F} \quad (2)$$

The calculated U_v from Equation 1 is 2718 Btu/hr ft³ based on the calculated holdup of 22.672% and R of 2.24. Thus the experimental value is only 12% lower than the correlation would predict. This is within acceptable limits for an experimentally determined heat transfer formula.

If we use this constant value of U_v and assume average values of specific heat for the IC_4 and for the brine, the results shown on Figure 4 are found. If we assumed average values of the specific heats over smaller temperature increments*, the curves would shift to the right somewhat at the bottom and to the left at the top. However, it is clear that all of the data lies between the isobutane

* Instead of the whole range of temperature, a change to three or four increments might be used.

Figure 4. Temperature observed and calculated based on constant U_v in the DCHX, 12 Nov 1980 (HWA)

and the brine curves. Further it is noted that the data provides a less steep slope than the model close to the bottom, a flat portion in the middle and a steeper portion at the top. If one assumed no heat transfer during the first five feet and then predicted heat transfer for the next eight feet at a U_v of 2390 Btu/ft² °F, the experimental temperatures would lie between the isobutane and brine theoretical curves. A higher heat transfer coefficient would be required above this level in order to complete the heat transfer in the experimentally determined 23 feet above the distributor. Note that Equation 1 would yield a significantly higher heat transfer rate for $R = 0.44$, the value when the isobutane is the continuous phase.

The postulates described above indicate a strong possibility that the design of the distributor for the isobutane has decreased the heat transfer at the bottom of the tower. The flat portion at the middle of the column still suggests a change in the fluid in the dispersed and continuous liquids. No data exists for water as the dispersed phase in a hydrocarbon; however, it could well be that for such a situation, the volumetric heat transfer coefficient would be higher than when water is the continuous phase. (If the holdup is the same, Equation 1 would predict higher heat transfer if it is applicable). All experimental data in the literature purportedly has water as the continuous phase with either a light hydrocarbon or kerosene as the dispersed phase. These facts cannot be determined however, without significant modifications to the direct contact heat exchanger or the use of another column. It should be pointed out that a column of appreciable size should be used.

The data obtained using brine injection at lower height in the column as was accomplished in 1982 provide additional information. They seem to indicate that the column can be shortened considerably. However, none of the data is conclusive as far as the heat transfer rates are concerned. All of the data indicates (see B/N report "Final Phase Testing and Evaluation of the 500 kWe DCHX Pilot Plant at East Mesa")⁵ a level portion in the column when the brine level is high, 24+ feet, and the conclusion and hypothesis drawn on the 1980 data, as mentioned above, still hold. However, there is an

occasional exception such as for March 3, 1983 when it appears that most of the heat transfer occurs in the first ten feet of the column with the profile of experimental temperature flat for almost ten feet. That this occurred for a several-hour period, although at high brine flow rates (210 gpm), is remarkable. This was at a high brine distributor level, 24 ft 1 inch and high brine level 25 feet 11 inches. It soon adjusted to the "normal" distribution with two primary zones of heat transfer.

When the brine distributor was lowered to 16 ft 3 inches with a brine level of 20 ft 11 inches, the plateau of temperature in the middle of the column disappeared (see Figures 5 and 6). In Figure 5 the flow rates of IC_4 and brine were only 207 gpm and 145.9 gpm respectively as compared to the 327.9 gpm and 192 gpm values shown in Figures 3 and 4. Calculation of the volumetric heat transfer coefficient from the experiment yields a value of only 1440 Btu/hr ft^3 $^{\circ}F$ which is consistent with the low flow rates and low value of holdup, $\emptyset$.

In Figure 6 the flow rates of the isobutane and brine are consistent with the high brine injector position, $V_{IC_4} = 337.2$ gpm and $V_{brine} = 193.5$ gpm. Because of missing experimental temperatures between the 11 foot position and the 19 foot position, it is unclear where the top of the preheater is located. It could, perhaps, be anywhere between 12 and 19 feet as indicated on the figure. The energy balance based on the flow rates and the reported temperatures and pressure indicate that the preheater temperature of the brine lies between 251 $^{\circ}F$ and 230 $^{\circ}F$. Since the boiling point is 242 $^{\circ}F$, the high brine flow is assumed correct; thus, the temperature is probably 251 $^{\circ}F$. The U_v can be calculated to be between 4467 Btu/hr ft^3 $^{\circ}F$ at $L_{preheater} = 16$ ft and 2821 Btu/hr ft^3 $^{\circ}F$ at $L_{preheater} = 19$ ft. (The holdup is slightly higher than for the case of November 12, 1980 as is the value of R ; thus, Equation 1 would predict U_v of approximately the same, 2800 Btu/hr ft^3 $^{\circ}F$ when water is the continuous phase). This would indicate that by lowering the brine injector, and in effect shortening the column length, the column could be shortened from 21.8 ft to between 12 and 19 feet. Thus, in effect, saving a length of 10 to 45%. There is no indication whether further short-

FIGURE 5. DCHX heat transfer, 23 Dec 1987

FIGURE 6. DCIX Heat Transfer, 14 Oct 1982

ening might be accomplished or what mechanism allows for it.

The heat transfer correlation of Equation 1 was accomplished by using data from a variety of fixed size, length and diameter column in which flow rates were changed. There is no reason to suspect that these were optimized. Data used to generate the Plass-Jacobs-Boehm correlation for sizing preheaters shows considerable scatter and in fact, values more than 25% higher than the correlation curve as well as 25% lower were used. Thus, the shortening of the column and its successful operation clearly indicates that the correlation is conservative, a fact that was known by its originators.

The peculiarity of the temperature profile for the long column and the regular nature of the short column are unresolved. In the tall column, the data suggests that the top of the column operates with isobutane as the continuous phase and brine as the continuous phase at the bottom. The higher heat transfer normally seen at the top of the tall column could mean that when the column is shortened, that isobutane is forced to be the continuous phase even down into the lower part of the column. This eliminates the flat temperature profile in the center of the column. This argument is only heuristic. No data exists to verify it or any other explanation of the column temperatures. However, it should be noted that assuming the same drop size for the water, Equation 1 would predict $U_v = 6765$ Btu/hr ft³ °F if the isobutane were the continuous phase. This indicates that it might be possible to achieve even shorter column heights.

The only real conclusion that can be drawn, based on the East Mesa experiments, is that the correlation given in Equation 1 will successfully design an IC₄ brine columns preheater section and that it will be conservative using only the end point temperatures.

In regards to the boiler section, the data acquired from this testing program sheds no new light on its design. The design practice suggested at the beginning of the program is still the best means of "guesstimating" the volume required.

RECOMMENDATIONS

Further studies are needed if improvements to spray column direct contact heat exchanger designs are to be made. These studies should: (1) concentrate on determining the minimum length of column that can achieve counterflow operation; (2) determine the effects of brine and working fluid injector design on column performance. The above studies should be carried out on relatively large scale equipment so as to provide information practical to design of operational systems. Two such systems are available. These are the East Mesa tower (40-inch diameter) and the new (24-inch diameter) tower constructed for Solar Energy Research Institute tests by the University of Utah.

REFERENCES

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3.7 APPENDIX C - 500 KW DIRECT CONTACT HEAT EXCHANGER TESTING LOG
FOR 1982

Date	Configuration	Conditions	Comments
1-13-82	Brine Distributor @ 22' 1"	Startups and Shutdowns	Revised Brine Distributor Installed
1-14-82	"	"	Shutdowns due to DCHX Level Control
1-15-82	"	"	Brine Level @ 23' 3" to 24' 5"
1-18-82	"	"	
1-19-82	"	"	
1-20-82	"	IC ₄ Flow = 335 gpm Brine Flow = 210 gpm P = 435 psi	
1-21-82	"	IC ₄ Flow = 225 gpm Brine Flow = 150 gpm P = 435 psi	
1-22-82	"	IC ₄ Flow = 325 gpm Brine Flow = 210 gpm P = 435 psi	
1-25-82	"	IC ₄ = 335 gpm Brine Flow = 210 gpm P = 435 psi	Recovery System Test
1-26-82	"	Plant Startup and Turbine Problem	
1-27-82	Brine Distributor @ 22' 1"	Max. Brine Flow = 217 gpm P = 400 psi	
1-28-82	"	Brine Flow = 226 gpm IC ₄ Flow = 345 gpm P = 375 psi	
2-8-82	"	Brine Flow = 182 gpm IC ₄ Flow = 434 gpm P = 435 psi	
2-17-82	"	Brine Flow = 180 gpm IC ₄ Flow = 267 gpm P = 470 psi	

3.7 APPENDIX C - 500 KW DIRECT CONTACT HEAT EXCHANGER TESTING LOG
FOR 1982

Date	Configuration	Conditions	Comments
2-18-82	"	Brine Flow = 175 gpm IC ₄ Flow = 266 gpm P = 435 psi	
3-1-82	Brine Distributor @ 24' 9"	IC ₄ Flow = 280 gpm Brine Flow = 190 gpm P = 375 psi	Brine Level @ 25' 11"
3-2-82	"	Plant running at different flow rates and pressures as required for the test map	
3-3-82	"	"	
3-4-82	"	"	
3-8-82	"	"	Brine Level @ 27' 1"
3-9-82	"	"	
4-22-82	"	Brine Flow = 202 gpm IC ₄ Flow = 281 gpm P = 435 psi	Brine level @ 27' 1"
4-23-82	"	"	
4-24-82	"	"	
4-26-82	"	Plant not running	Repairs on recovery compressor and turbine
4-27-82	"	Brine Flow - 180 gpm IC ₄ Flow = 277 gpm P = 400 psi	
4-28-82	"	Brine Flow = 200 gpm IC ₄ Flow = 308 gpm P = 435 psi	
4-29-82	"	"	
5-6-82	"	Brine Flow = 214 gpm IC ₄ Flow = 378 gpm P = 340 psi	Recovery system with all of the CO ₂ returned to the condenser, installed and tested

3.7 APPENDIX C - 500 KW DIRECT CONTACT HEAT EXCHANGER TESTING LOG
FOR 1982

Date	Configuration	Conditions	Comments
5-7-82	"	Startups and Shutdowns	Recovery system with all of the CO ₂ returned to the condenser, installed and tested
5-9-82	"	"	
5-10-82	"	"	
5-11-82	"	Brine Flow = 217 gpm IC ₄ Flow = 343 gpm P = 430 psi	
5-12-82	"	"	
5-13-82	"	Brine Flow = 216 gpm IC ₄ Flow = 360 gpm P = 435 psi	
5-14-82	"	"	
5-15-82	"	"	
5-17-82	"	Brine Flow = 217 gpm IC ₄ Flow = 350 gpm P = 435 psi	Recovery system test
5-18-82	"	"	"
5-19-82	"	"	
5-20-82	"	"	
5-21-82	"	"	
6-3-82	Brine Distributor @ 21' 5"	Brine Flow = 216 gpm IC ₄ Flow = 350 gpm P = 435 psi	Brine Level @ 23' 3"
6-4-82	"	Brine Flow = 215 gpm IC ₄ Flow = 340 gpm P = 435 psi	
6-7-82	"	"	
6-8-82	"	"	

3.7 APPENDIX C - 500 KW DIRECT CONTACT HEAT EXCHANGER TESTING LOG
FOR 1982

Date	Configuration	Conditions	Comments
6-9-82	Brine Distributor @ 21' 5"	Brine Flow = 216 gpm IC ₄ Flow = 340 gpm P = 400 psi	Brine Level @ 27' 1"
6-10-82	"	"	
6-11-82	"	Acidizing of the sand-trap loop	Brine Level @ 25' 11"
6-14-82	"	"	
6-15-82	"	"	
6-16-82	"	Brine Flow = 175 gpm IC ₄ Flow = 275 gpm P = 435 psi	
6-17-82	"	Brine Flow = 210 gpm IC ₄ Flow = 317 gpm P = 435 psi	
6-18-82	"	"	
6-23-82	"	"	
6-24-82	"	"	
6-25-82	"	"	
6-28-82	"	Brine Flow = 216 gpm IC ₄ Flow = 342 gpm P = 400 psi	
6-29-82	"	"	
6-20-82	"	Startups and Shutdowns	
7-1-82	"	"	Brine Level @ 23' 3"
7-6-82	"	Brine Flow = 216 gpm IC ₄ Flow = 340 gpm P = 400 psi	
7-7-82	"	"	
7-8-82	"	"	
7-9-82	"	"	

3.7 APPENDIX C - 500 KW DIRECT CONTACT HEAT EXCHANGER TESTING LOG
FOR 1982

Date	Configuration	Conditions	Comments
7-12-82	"	Brine Flow = 210 gpm IC ₄ Flow = 335 gpm P = 430 psi	
7-13-82	"	"	
7-14-82	"	Brine Flow = 206 gpm IC ₄ Flow = 330 gpm P = 430 psi	
7-15-82	"	Brine Flow = 216 gpm IC ₄ Flow = 340 gpm P = 435 psi	
7-22-82	"	Brine Flow = 200 gpm IC ₄ Flow = 320 gpm P = 435 psi	
8-2-82	Brine Distributor @ 19' 3"	Brine Flow = 200 gpm IC ₄ Flow = 320 gpm P = 435 psi	Brine Level @ 20' 11"
8-13-82	"	Brine Flow = 212 gpm IC ₄ Flow = 311 gpm P = 435 psi	
8-16-82	"	Brine Flow = 215 gpm IC ₄ Flow = 330 gpm P = 435 psi	
8-17-82	"	Brine Flow = 215 gpm IC ₄ Flow = 320 gpm P = 435 psi	
8-18-82	"	"	
8-30-82	"	Startup and Shutdown	
9-1-82	"	Brine Flow = 175 gpm IC ₄ Flow = 265 gpm P = 435 psi	
9-7-82	"	Maximum brine flow rate is gradually decreasing	REDA pump problems started
9-14-82	"	"	"
9-15-82	"	"	

3.7 APPENDIX C - 500 KW DIRECT CONTACT HEAT EXCHANGER TESTING LOG
FOR 1982

Date	Configuration	Conditions	Comments
9-22-82	Brine Distributor @ 16' 3"		Brine level @ 20' 11"
10-11-82	"	Brine Flow = 210 gpm IC ₄ Flow = 320 gpm P = 430 psi	New REDA pump installed
10-12-82	"	"	
10-13-82	"	Brine Flow = 212 gpm IC ₄ Flow = 330 gpm P = 435 psi	
10-14-82	"	Brine Flow = 175 gpm IC ₄ Flow = 270 gpm P = 435 psi	
10-15-82	"	"	
10-19-82	Brine Distributor @ 19' 3"	Brine Flow = 212 gpm IC ₄ Flow = 325 gpm P = 400 psi	
10-20-82	"	Brine Flow = 240 gpm IC ₄ Flow = 390 gpm P = 400 psi	

4.0 WORKING FLUID RECOVERY EVALUATION

4.1 INTRODUCTION

The use of a direct contact heat exchanger in recovering energy from geothermal brines is advantageous for binary power plants because it eliminates heat exchanger fouling at the heat input side of the power loop. A typical working fluid used in the direct contact approach has low miscibility in brine; however, significant amounts can still be lost from the plant unless working fluid recovery equipment is provided. For a 5 MW geothermal power plant using isobutane as the working fluid, isobutane losses amount to approximately \$.02/kW-hr if equilibrium solubility limits are reached in low salinity effluent brine. A cost effective recovery system reduces the isobutane losses to a few mils per kilowatt hour, thus reducing operating costs and improving power plant economics.

There are many approaches to recovering working fluids from the brine before it leaves the plant. Both physical and chemical separation techniques are available. All the available recovery methods are complicated by the presence of entrained noncondensable gases, such as carbon dioxide and nitrogen. The presence of these gases require that the recovery method be a two-step process. The first step liberates or strips the working fluid from the brine. The second step separates the working fluid from any noncondensable gases.

The 500 kW DCHX pilot plant at East Mesa has demonstrated cost effective recovery of working fluid using a vacuum flash extraction and compressor condensation recovery system. This approach was chosen for its simplicity.

The objective of this study was to identify the most cost effective recovery system approach for a direct contact system. The scope of this study was constrained to the conditions existing with the 500 kW demonstration pilot plant for selecting the optimum system. This constraint is not considered restrictive because the conditions of the brine used in the pilot plant are similar to most

brines in existence, and the isobutane working fluid is a logical selection for brine temperatures greater than 300°F. Working fluids and noncondensable gas conditions other than those used with the 500 kW pilot plant were considered. The costs cited in this study were based on a 5 MW plant using the same process as the 500 kW pilot plant at East Mesa. The economic evaluation did not include operational labor costs, detailed maintenance and downtime expenses for any of the systems compared.

4.2 EVALUATION SUMMARY

Isobutane recovery is complicated by noncondensable gases entrained in the brine which makes recovery of the working fluid a two-step process. To reflect this, the study has been broken into two segments, one dealing with liberating the working fluid from the brine and the other dealing with separating the working fluid from noncondensable gases that are also liberated. Liberating the isobutane from the brine is referred to as the stripping operation. Isolating the working fluid from the noncondensable gases is referred to as the separation operation.

Four basic approaches for the stripping operation and four approaches for the separation operation were evaluated. Various combinations of the stripping and separation operations are inter-related but separate evaluation allows comparison of various combinations.

The stripping processes evaluated were vacuum flash extraction, gas stripping, salting and absorption with a lean oil. The vacuum flashing and gas stripping processes both require the second step of separating the working fluid from the noncondensable gases liberated. For both the salting and the lean oil absorption processes, the noncondensable liberation problem is probably avoided and separation is not required.

The most practical approach for stripping working fluid from the brine is a vacuum flash technique. The other three stripping processes are ruled out by both practicality and economics. The

gas stripping technique requires large amounts of recirculated non-condensable gas to achieve its performance. Circulating this large quantity of gas requires large operating power levels and significantly dilutes the recovered working fluid making later separation difficult and expensive. The salting approach is impractical due to the large quantities of salt required to reduce the working fluid solubility in the brine to the point where working fluid losses are acceptable. Stripping with lean oil eliminates the cost of lost working fluid but creates a new cost associated with replenishing lost lean oil due to its solubility in the brine.

Where practical, several vacuum flash configurations were evaluated to establish the economic tradeoffs of the total combined stripping/separation approach. Vacuum stripping in a multi-stage configuration reduces the working fluid lost in the discharged brine to less than 10 ppm or a predicted equivalent flow rate less than 10 lbs/hr for a 5 MW plant.

Four techniques were evaluated for separating the working fluid from noncondensibles and returning the recovered working fluid to the power system. Separating the two gases is desirable because returning noncondensable gases to the power loop increases their level in the power condenser which increases condenser pressure. Increased condensing pressure reduces the power that can be extracted from the working fluid by the power turbine, creating an economic penalty to the plant. The four approaches evaluated include permeable membranes, compression and condensation, selective absorption and distillation. Three of these approaches are practical and had operating costs that were competitive. These systems had a payback of less than one year and operating costs ranging from 1.6 mils per kilowatt hour to 3.3 mils per kilowatt hour. The costs did not include equipment maintenance or operational man-hours required. The distillation system was noncompetitive. Within the scope of this study, it was not possible to optimize all three cost competitive techniques. The compression and condensation approach was optimized because it appears to have the lowest potential operating cost due to its simple hardware configuration.

The evaluation and optimization of the compression condensation technique showed that minimum operating costs occurred when the liberated working fluid and noncondensable gases were returned directly to the power loop. Even after accounting for the power loss penalty due to the increased levels of noncondensable gases in the system, the operating cost was 1.4 mils per kilowatt hour. This approach will work for any working fluid and any noncondensable gases normally present in the brine.

The evaluations of the different approaches to working fluid recovery and the ground rules used in the evaluations are summarized in the following sections.

4.3 RECOVERY SYSTEM EVALUATION GROUND RULES

For the recovery evaluation study, a 5 MW plant was selected. A power plant this size without a recovery system would experience an operating cost penalty of approximately \$.02/kW-hr due to the isobutane lost in the brine which is discharged from the plant. A 5 MW plant, being relatively small, might experience higher relative recovery system costs than a larger plant due to the size scaling effect. The 5 MW size does not affect the comparison between the different types of systems. The 500 kW pilot plant at East Mesa is considered a reasonable model of a commercial plant in terms of operating conditions, and brine resource temperature and composition. The state points of the 500 kW plant were used as a basis for evaluating and comparing the various approaches to working fluid recovery. Applications of the direct contact principle to other resources could influence the choice of recovery systems due to a different working fluid or variations in the noncondensable gases in the resource brine. The brine resource at the East Mesa site contains high levels of CO₂ and, like any resource with significant amounts of noncondensable gases, a preflash of the incoming brine is required to reduce the noncondensable gases to a level acceptable for good power production. The brine resource reaches the 500 kW pilot plant at East Mesa at a temperature of 340°F. Resources with higher temperatures will have increased power production per pound of brine and therefore lower costs associated with fluid recovery.

To provide a direct cost comparison between the various systems and to establish a reasonable estimate of the operating cost penalty associated with power production, the following items were included in evaluating each approach:

1. The costs of all unrecovered or lost working fluid.
2. The power costs required to operate pumps and compressors.
3. Power cost to boost the brine pressure back to a 25 psia level for discharge from the power plant and power lost due to increase of the discharge pressure for the hydraulic turbine above 25 psia.
4. For the optimized systems, the penalties associated with lost power production due to CO₂ returned to the power system.
5. The cost of brine that was used to provide heat energy.
6. Equipment costs were established for each system and then converted to an equivalent operating cost based on a 40,000-hour depreciation neglecting any interest, taxes or investment cost penalties. The assumption of 40,000 hours to depreciate the hardware represents a 5-year life.

Recovery system equipment costs were estimated after the equipment was sized based on a costing guide titled "Current Costs of Process Equipment", published in Chemical Engineering, April 5, 1982. No attempt was made to evaluate the potential operating costs associated with man-power required to run the given system or with maintenance costs or the potential effect of system downtime on plant capacity factors.

The following efficiencies, cost factors and plant conditions were selected as a basis for evaluating all of the recovery approaches:

1.	Plant output power level	5 MW (net)
2.	Plant brine flow rate	10^6 lbs/hr
3.	Initial CO ₂ content of brine (after pre-flash)	300 lbs/hr
4.	IC ₄ content of brine leaving DCHX	200 lbs/hr
5.	Compressor efficiency	55%
6.	Motor efficiency	85%
7.	Pump efficiency	55%
8.	Electrical power value	\$.07/kW-hr
9.	IC ₄ cost	\$.50/lb
10.	Hydraulic turbine efficiency	85%
11.	Value of brine	\$.034/lb

4.4 TECHNIQUES FOR STRIPPING ISOBUTANE FROM THE BRINE

The four methods evaluated for separating the IC₄ from the brine leaving the DCHX were absorption of the working fluid directly from the brine, gas stripping or sparging, solubility reduction and vacuum flash stripping. The evaluation showed that the vacuum flash is superior to other methods. This approach is compatible with all of the common noncondensibles and operates at a low cost. The vacuum flash approach to working fluid stripping was evaluated in depth to establish the most effective approach for various configurations and to predict their performance potential.

4.4.1 Lean Oil Absorption of IC₄

Lean oil absorption of IC₄ is a type of liquid-to-liquid extraction used directly with the brine. Liquid-to-liquid extraction is normally used when a mixture of two completely miscible liquids is separated into two components. It is based on the principle that a solvent not completely miscible with the mixture can selectively dissolve one of the constituents of the mixture. For this application liquid-to-liquid extraction was applied to a mixture of two partially miscible fluids.

For a concentration of IC₄ in the exit brine of 200 ppm, the solubilities of the solvent in the brine must be less than 10-15

ppm or a significant percentage of the IC_4 along with the solvent will be lost. Since solubilities of the IC_4 solvents are considerably higher than the solubility of the isobutane in water (for example; kerosene - 250 ppm, paraffin oil - 220 ppm, chloroform - 750 ppm) the problem of the IC_4 recovery will only be converted to a greater problem of solvent recovery. Hence this approach would not appear to be practical and was not evaluated any further.

4.4.2 Gas Stripping/Sparging

Gas stripping of the isobutane from the brine requires bubbling a stream of noncondensable gas, such as CO_2 or N_2 , through the brine. The noncondensable gas bubbles rise through the brine absorbing isobutane. The gas mixture is then directed to a separation system where the isobutane is removed from the gas mixture and recovered. The stripping gas may then be either vented or recycled to the separating equipment. Experimental work by Ike Sheinbaum in this area, summarized in his report entitled, "Stripping of Isobutane from Geothermal Brine", prepared for Lawrence Berkeley Laboratories, dated October, 1980, showed that approximately 100% of the isobutane could be stripped using either N_2 or CO_2 . It is felt that nitrogen might be superior because of its lower solubility in the brine and, hence, less stripping gas would be carried away with the brine. Referring to Sheinbaum's data, the gas flow rates used for stripping gave an average of 0.456 mole percent of isobutane in the stripping gas. For a 5 MW plant having a brine flow of 1 million lbs/hr and approximately 200 ppm of dissolved isobutane, approximately 21,000 lbs/hr of gas would have to be processed for this operation. The equipment costs to recycle this amount of nitrogen were not estimated but were felt to present an unreasonable penalty. Based on operating experience with the 500 kW plant and a review of the process industry literature, the major problem is not removing the isobutane from the spent brine but separating and recovering the isobutane from this gas mixture. The high dilution of the isobutane that would occur with a gas stripping system would increase the difficulty of recovering the isobutane. Therefore, gas stripping of the spent brine to recover isobutane is not recommended because it offers significant cost and operating difficulties.

4.4.3 Reduced Solubility with Increased Salinity

Figure 4.1 shows solubility of isobutane in brine with various levels of salinity. The figure shows that at high levels of salinity the solubility of isobutane is reduced to levels that result in loss rates costing less than 2 mils per kilowatt hour. This characteristic indicates that the recovery problem is reduced with higher salinity brines.

The process of adding salt to increase salinity to a level that would reduce isobutane losses to an acceptable level for East Mesa brines requires 1,000 lbs of sodium chloride per pound of isobutane recovered. It is apparent that the use of salt to prevent IC_4 loss is not practical because of the large amounts of salt required or the inclusion of a salt recovery system. This approach was not studied in further detail.

4.4.4 Vacuum Flash

The vacuum flash stripping operation reduces the pressure of the brine and provides a space for the working fluid to flash off as a gas. The solubility of the working fluid is proportional to its partial pressure and decreasing the total pressure of the brine reduces the amount of IC_4 that can remain in solution in the brine. By spraying the brine into a gas-filled chamber, a large surface area can be created allowing rapid evolution of the working fluid into the gas space. Vacuum flash stripping of the working fluid from the brine works well regardless of working fluid or type of noncondensable gases involved.

A configuration with multiple stages of flashing generally provides better performance than a single stage of flash. Flash configurations of up to three stages were evaluated. There was a considerable improvement in isobutane stripping with the two-stage approach over a single-stage but a minimal amount of improvement with the three-stage over the two-stage. The economic aspects of the flash configurations were evaluated with the systems used to separate and recover the isobutane from the noncondensibles. All

SOLUBILITY OF ISOBUTANE IN WATER AS A FUNCTION OF SALINITY

FIG. 4.1

of the flash systems were evaluated over a range of system conditions to establish the optimum operating parameters with either the single or multiple stages. In terms of just stripping, the two-stage flash with the first stage occurring at 35 psia and the second stage occurring at 10 psia was superior in terms of isobutane recovery and operating power requirements.

4.4.4.1 Theoretical Model

The following model was used to analyze the flashing process and to establish the composition and flow rates. The temperature and pressure within the flash tank were selected, then the equilibrium phase concentrations of the liquid vapor stream were determined.

In Figure 4.2, F (moles/hr) of an n -component stream are introduced as feed into the flash vaporization tank.

The resulting vapor and liquid streams are withdrawn at the rates of V and L (moles/hr) respectively. The mole fractions of the components in the feed, vapor and liquid streams are designated as Z_i , Y_i and X_i respectively.

Assuming vapor-liquid equilibrium and steady-state operation, we have the following series of algebraic relationships:

$$Z_i F = X_i L + Y_i V$$

$$Y_i = H_i K_i$$

$$\sum Y_i = 1$$

$$\sum X_i = 1$$

Where K_i is the equilibrium constant for the i -th component, at the temperature and pressure in the tank. K_i values were found by using Raoult's and Henry's laws.

$$K_i = H_i / P_{\text{tot}} \quad (H = \text{Henry's constant for a very dilute solution of I-component})$$

$$K_i = P_{\text{sat}_i} / P_t \quad (P_{\text{sat}_i} = \text{vapor pressure of I-component at tank temperature})$$

These equations were used to develop the stream properties for all the flash stripping configurations which then could be evaluated in terms of performance and cost. These stream properties also became the basis for the separation studies. For the stripping studies, the brine entering the flash systems at 1,000,000 lbs/hr contains 300 ppm of free CO_2 and 200 ppm of IC_4 .

4.4.4.2 Single-Stage Flash

The single-stage flash is shown schematically in Figure 4.3.

Brine from the DCHX is run through a Pelton-type hydraulic turbine for pressure recovery. The Pelton wheel operates in a gas environment which is contained by an enclosing vessel called a tailstock. The tailstock provides the vapor space for stripping the IC_4 vapor. For some systems (i.e., pentane instead of isobutane), a recovery turbine may not be necessary and the turbine is replaced by simple spray nozzles and separation tank. The vapors are directed to a compressor that pumps them to the separation system for final recovery. Figure 4.4 shows the predicted effect of tailstock and, hence, compressor suction pressure on the flash process. As shown in the figure, decreasing the tailstock pressure increased the amount of IC_4 in the flash gas while reducing the amount of IC_4 lost in the brine. However, reducing the tailstock pressure also increased the amount of water vapor and CO_2 contained in the flash gas. These additional components can have a significant effect on the downstream separation system.

4.4.4.3 Two-Stage Flash

In general, a multi-stage flash is an improvement over a single-stage flash. A two-stage configuration is shown schematically in Figure 4.5.

COMPRESSOR INLET COMPOSITION AND
RATE OF IC₄ LOSS IN THE BRINE AS
A FUNCTION OF THE COMPRESSOR
INLET PRESSURE FOR A SINGLE-STAGE
FLASH

FIG 4.4

FIG 4.5

The first flash was again assumed to be across a Pelton wheel hydraulic turbine while the second stage flash was across spray nozzles into a vessel called the recovery tank. As shown in Figure 4.5, a single compressor is used to remove the flash gas from both locations. The compressor suction pressure was assumed to be equal to the pressure of the second stage flash. A throttle valve was used to reduce the pressure of the first stage gas (tailstock) to that of the compressor inlet. This configuration was utilized in the original 500 kW pilot plant at East Mesa.

The predicted performance of the two-stage flash is shown in Figure 4.6. Comparing Figure 4.6 for the two-stage flash with Figure 4.4 for the single-stage flash shows the improvement in IC_4 removal from the spent brine. For example, with a compressor inlet pressure of 10 psia, the two-stage flash leaves 3.2 lbs/hr of IC_4 in the brine, while the same compressor inlet pressure (tailstock pressure) with a single-stage flash leaves 5 lbs/hr of IC_4 in the spent brine. An additional advantage of the two-stage flash

COMPRESSOR INLET COMPOSITION AND
RATE OF IC₄ LOSS IN THE BRINE AS A
FUNCTION OF THE COMPRESSOR INLET
PRESSURE FOR A TWO-STAGE FLASH

FIG 4.6

COMPRESSOR INLET COMPOSITION AND
RATE OF IC₄ LOSS IN THE BRINE AS A
FUNCTION OF THE COMPRESSOR INLET
PRESSURE FOR A THREE-STAGE FLASH

FIG. 4.7

is the carbon dioxide and water vapor flow rates handled by the compressor are lower.

4.4.4.4 Three-Stage Flash

The performance of a two-stage flash might be improved by adding an additional flash stage. Again, as with the two-stage flash, in the final stage flash the vapor flow from the previous stages is throttled to the compressor suction pressure. Figure 4.7 summarizes the predicted performance of this configuration. Comparing Figure 4.7 with Figure 4.6 for a two-stage flash shows the additional flash stage resulted in only a very small improvement in isobutane recovery. For example, with a 10 psia compressor inlet pressure, the two-stage flash resulted in approximately 3.2 lbs/hr isobutane flow in the spent brine after the second flash, while for the three-stage flash, the isobutane flow was approximately 3.1 lbs/hr. The three-stage flash did not reduce the CO₂ to be handled by the separation system.

The pressures of the first and second stage flashes for Figure 4.7 were not optimized, so the three-stage system might be capable of greater performance. However, any performance increase should be slight.

4.4.4.5 Two-Stage Flash with First Flash to Power Condenser

A modification of the two-stage flash system was to have the first stage flash pressure high enough that the flash gas could be directed to the main power condenser (Figure 4.8).

The advantage of this configuration is that significant amounts of IC_4 may be liberated from the spent brine and recovered without requiring additional compressor work, thereby improving plant economics. The carbon dioxide or other noncondensable gas liberated during the flash also returns to the condenser, increasing the partial pressure of noncondensable gases in the condenser and thereby decreasing plant output.

The predicted performance of a two-stage flash systems with the first flash directed to the power condenser is summarized in Figure 4.9. Comparing this configuration with the simple two-stage flash shows this system has greater IC_4 loss because of the less than optimum isobutane recovery of the first stage flash. However, the reduced mass flow through the compressor lowered the compressor work and input power. The value of this relationship is dependent on the rest of the separation/recovery system.

4.4.4.6 Three-Stage Flash with First Flash to Power Condenser

A three-stage flash with the first stage vented to the power condenser was evaluated to provide another baseline for the separation techniques.

The predicted performance of this configuration is summarized in Figure 4.10. Comparing the results of this analysis with that for the three-stage flash shown in Figure 4.7 shows the performance to be similar. It is concluded that this method offered the same recovery as the three-stage flash, discussed in Section 4.4.4.4.

4.5 RECOVERY OF ISOBUTANE

After the working fluid has been stripped from the brine, it must be separated and returned to the Rankine cycle power loop. The ideal system would separate all the CO_2 from the working fluid, eliminate it and return the working fluid to the power loop. Four techniques for separating and recovering the working fluid were investigated: permeable membranes, absorption, compression and

COMPRESSOR INLET COMPOSITION AND
RATE OF IC₄ LOSS IN THE BRINE AS A
FUNCTION OF THE COMPRESSOR INLET
PRESSURE FOR A TWO-STAGE FLASH

FIG. 4.9

COMPRESSOR INLET COMPOSITION AND
RATE OF IC₄ LOSS IN THE BRINE AS A
FUNCTION OF THE COMPRESSOR INLET PRESSURE
FOR A THREE-STAGE FLASH SYSTEM

FIG.4.10

condensation, and distillation. The evaluation of these techniques was based on a $\text{CO}_2 + \text{H}_2\text{O} + \text{IC}_4$ mixture chemistry produced by the flash stripping process discussed previously.

The economic evaluation of each system was based on the ground rules set forth in Section 4.3. Each configuration evaluation includes equipment costs and operating costs of the appropriate flash stripping configuration required to feed it. Costs associated with unrecovered pressure energy in the brine due to high first stage flash pressure, increased power system condenser pressure, pumping to raise brine pressure back to 25 psia for discharge and required recovery system equipment are all accounted for.

4.5.1 Membrane Permeation

When a pressure gradient is established across a specifically prepared membrane, carbon dioxide will permeate more rapidly than hydrocarbon gases. Monsanto Co. has recently introduced a new membrane system referred to as a prism separator which makes gas separation by permeable membranes economically feasible.

The Monsanto prism separator physically resembles a shell-and-tube heat exchanger with a bundle of hollow fibers oriented vertically inside a tubular pressure vessel. Feed gas is injected on the shell side of the fibers and the carbon dioxide permeates the polysulfane polymer substrate and flows through the bore of the fibers. The pressure difference between the shell and the fiber bore provides the driving force.

The Monsanto Corporation proposed the isobutane recovery system depicted in Figure 4.11. The process consists of two stages of membrane separation. Gases extracted from the brine in a two-stage flash make up the feedstream. Flash vapors from each stage combine for a total feed gas composition at 145°F and 10.5 psia of:

CO_2	68.9 lbs/hr
IC_4	196.8 lbs/hr
H_2O	15.4 lbs/hr

MEMBRANE SEPARATION SYSTEM

FIGURE 4.11

The feed gas is compressed to 260 psia and then it is mixed with effluent gas from the second stage separator. This high pressure mixture is fed to the first separation stage. Effluent gas from the first stage separation is returned to the power condenser containing approximately 3.5 lbs/hr of CO_2 . The permeate gas from the first stage separator is compressed to 295 psia before entering the second stage separator. Effluent gas from the second stage separator is returned to the mixer while permeate gas containing 5.9 lbs/hr of IC_4 is vented. In the #2 compressor intercooler, 37.2 lbs/hr of IC_4 and 10 lbs/hr of CO_2 are condensed and returned to the power system.

The proposed system recovers 190.0 lbs/hr of isobutane. Operating input power costs are estimated at \$12,000 annually. An additional \$5,000/yr is required for a 10% membrane replacement in three years. Monstanto's estimated capital cost, which includes membrane, skid, controls and a two-stage compressor, is \$150,000. Barber-Nichols included an additional \$12,000 for the feed stream compressor and \$1.43/hr for compressor input power cost. Based on the economic ground rules, this translates into \$14,000/hr operating cost including the vacuum flash stripping.

This system offers a payback of less than one year. The membrane is not suitable for a feed gas containing nitrogen. The rate at which a gas permeates a membrane depends on its ability to diffuse through the substrate and its affinity for the polymer. Nitrogen is relatively inert and is considered a "slow" gas as is isobutane making membrane separation impractical for nitrogen dominated brines. Membrane systems are more economical when used as a topping process or preseparation system in conjunction with another separation technique. In a topping process, the effluent from the final membrane separation stage would contain approximately 20 mole percent CO_2 . For further purification, an absorption process or compression and condensation might be used. Due to the low CO_2 mole percent in the feed stream, a topping process is not justified for this application.

The CO_2 returned with the IC_4 to the Rankine cycle increases the condenser total pressure and reduces the power available to the turbine and the output power. This effect was not included in the analysis.

4.5.2 Selective Absorption & Recovery

A typical absorption system is illustrated in Figure 4.12. A solution selective to the component to be absorbed is introduced counter current to the feed gas stream in a column called the absorber. The solution leaving the absorber which is rich in the absorbed gas is sent to a stripping column for regeneration which separates the gas from the solution. Regeneration in the stripping column is dependent upon favorable temperatures and pressures of the rich solution. Heat is typically added to the rich solution in a reboiler before entering the stripping column. Heat requirements are reduced by a waste heat exchanger. The regenerated solution requires cooling before introduction into the absorber. In the stripper, the solution is regenerated by stripping using a vapor or a gas stream, typically steam. In addition to the absorber and stripper columns, Hoffman advises in Phase and Flow Behavior in Petroleum Production that it is usually advantageous to employ a rectifying section using the column overhead vapors, cooled and partially condensed, to provide a reflux stream. The performance of an absorber-stripper system depends upon operating temperatures and pressures, feed gas composition and purity required. Economics are determined by separation requirements, recirculation rates and utility requirements to regenerate the rich solution

To size equipment for an absorption system, the input gas composition was that of a two-stage flash. The gas mixture for sizing the plant contained 63.9 lbs/hr of CO_2 , 196.8 lbs/hr of IC_4 , and 15.4 lbs/hr of H_2O . Absorbent recovery of the IC_4 can be accomplished either by absorbing the CO_2 from the gas stream with any of the three readily available solutions or by absorbing the IC_4 .

ABSORPTION SYSTEM

FIGURE 4.12

4.5.2.1 Selective Absorption of Noncondensable Gas

Typical CO₂ absorption systems use sodium carbonate, potassium carbonate and the amines as the absorbing liquid. All three mediums depend on a chemical reaction between the absorbed gas and the solution.

Industrial processes based on these solutions can be divided into two groups, those that operate the absorber at essentially ambient temperatures, 70-100°F, and those that occur at approximately the same temperature at which regeneration takes place. The latter process is primarily suitable for feed gases under pressure.

A sodium carbonate system with the absorption process taking place at ambient temperatures was considered. The solution reacts with the CO₂ as follows:

The forward reaction occurs in the absorber and the reverse reaction in the stripper. Regeneration of the solution is by vacuum distillation in the stripper column. The heat required for activation is provided by steam.

A hot potassium (Benfield) process provides an effective CO₂ removal system at pressures of approximately 300 psia. A high absorption temperature greatly increases the absorption rate and reduces the heat required to heat the rich solution to stripping temperatures. With lower absorber pressures, lean circulation rates and steam requirements increase significantly. Regeneration of the solution is by flashing to atmospheric pressure and using steam as a stripper. Operational difficulties for the high pressure system include foaming and corrosion. These problems are generally controlled by using additives.

Amines, while effective for CO₂ removal, do not offer an economic advantage to the two systems just discussed. Amines react

with the CO_2 to form the amine salt of carbonic acid. Monoethanolamine, diethanolamine and triethanolamine are commonly used in industry although use of the latter has dropped due to its low reactivity and relative instability. For relatively low CO_2 concentrations and few impurities such as COS and CS_2 which form stable corrosive compounds with the primary amines, the monoethanolamine solutions are generally preferred. Diethanolamines are much less reactive with CO_2 and CS_2 and are used when gas streams contain appreciable amounts of these contaminants.

All of the above systems are variable candidates for separating carbon dioxide and isobutane. Both alkali carbonate systems offer the advantage of not contaminating the working fluid as in the case with the amines. Based on the selected gas stream composition and using the absorption-stripping factor method and assumptions presented in Hoffman's book, Phase and Flow Behavior, the absorption systems were sized. Estimated fabrication costs of the absorption systems are approximately \$230,000 not including a compressor. Compressor costs are \$7,600 for the sodium carbonate system and \$13,000 for the potassium and amine systems. Total capital costs are equivalent to \$5.88 to \$6.08/hr. Operating costs range from \$8,700/yr for the sodium carbonate system to \$13,760/yr for the amine absorption system which is equivalent to \$1.09 to \$1.72/hr. Including the costs of the vacuum extraction process and again neglecting the effect of CO_2 returned with isobutane, net system operating costs are \$13.48/hr, \$14.15/hr and \$14.19/hr for sodium carbonate, potassium carbonate and amine systems respectively.

The estimated payback of the CO_2 absorption recovery systems is less than one year. Performance is dependent, however, on the geothermal resource characteristics.

For a different resource, such as the brines of Raft River with nitrogen instead of carbon dioxide, alkali and amine solutions are ineffective because they will not react with nitrogen.

4.5.2.2 Selective Absorption of the Isobutane

Absorption of the isobutane from the stripped gas mixture would require a lean-oil solvent similar to kerosene. An isobutane absorption system has been proposed by Otto Schkade of Vatrol Co. (which designs and builds these systems) and is shown in Figure 4.13. Isobutane is absorbed by chilled lean-oil in the scrubber absorber at an operating pressure of 25 psia. The rich oil leaving the absorber is preheated in the recovery heat exchanger by the hot lean-oil leaving the stripper. The remaining stripping heat is provided by a steam heater. The absorbed isobutane is released in the stripper by flashing in a vacuum. The lean-oil is then chilled by a refrigeration unit before returning to the absorber.

Recovery of the isobutane is estimated at 97 mass percent or 190.9 lbm/hr. Capital costs are estimated between \$241,000 and \$291,000, equivalent to \$6.23 to \$7.52/hr. Utility and steam requirements are projected to be \$3.13/hr. Including the two-stage flash costs, the net operating costs is \$16.20/hr.

The projected system has a payback of less than one year. The lean-oil system is suitable for a feed gas composed of IC_4 and CO_2 or nitrogen. There is the possibility, however, of contaminating the working fluid with the oil. Vatrol estimated contamination would reach equilibrium at 0.75 mass percent of recovered isobutane, a cost penalty of approximately \$0.21/hr.

4.5.3 Compression and Condensation Separation

The IC_4 working fluid can be condensed into liquid to separate it from the noncondensable gases at reasonable pressure levels for temperatures ranging from 35 to 100°F. With the IC_4 condensed out of the gas stream, the remaining gases can be vented. Nominally, the higher the condensing pressure the lower the mole percentage of IC_4 or other working fluid vented with the noncondensable gases. In the case of an IC_4/CO_2 mixture, the higher pressures result in high solubility of the CO_2 and the IC_4 . This interaction requires

LEAN OIL ABSORPTION SYSTEM

FIGURE 4.13

an evaluation of a range of conditions to establish the best economic operating point, including the flash conditions. Compression-condensation was evaluated for single and dual flash and the dual flash was evaluated for both 95°F and 35°F condensing temperatures to explore this interaction. The evaluations are presented as a function of compressor power as this was considered to be the major recovery system cost factor.

Evaluation of the condensation pressures showed that the lowest cost for all the configurations occurred at pressure levels where all of the CO₂ is dissolved in the condensed IC₄, resulting in zero CO₂ vent. This finding leads to the obvious approach of eliminating the recovery condenser and returning all the stripped flow to the main condenser, thus achieving minimum compressor power and eliminating the complexity of a recovery condenser. In all of the configurations, the CO₂ introduced into the loop causes the same performance penalty. The use of the main condenser to condense the returned IC₄ does not measurably affect its heat load. The direct return to the power loop of the stripped mixture is the lowest cost approach and provides the best configuration in terms of equipment simplicity.

4.5.3.1 Composition Modeling

The model and equations for determining composition and flow rates for the recovery condenser are given in Appendix B, Section 4.7. System equilibrium conditions are based on concentration factors confirmed experimentally with the pilot plant and the fact that at equilibrium the input must be equal to the output. For a direct contact power loop the CO₂ entering must be equal to the sum of the CO₂ returned to the power loop from the separation system plus that leaving the plant in the brine. A separation system that allows CO₂ return to the power loop increases the CO₂ content of the gas stream it treats. Within the power loop itself the CO₂ concentration in the IC₄ is increased by a factor of eight over the level in the brine leaving the column, producing a measurable influence on the condensing pressure. The analysis of the absorption and the membrane systems ignored these recirculation effects in the

first-cut evaluation and only the analysis of the more attractive condensation schemes included them.

4.5.3.2 Initial Evaluation of Compression/Condensation

The single-stage flash and the two-stage flash were compared with a 95°F condensing temperature. The two-stage flash was evaluated for both an optimized first stage pressure and for first stage flash to the power condenser. Schematically the three configurations are shown in Figures 4.14, 4.15 and 4.16. The evaluation of these three configurations include cost of both the IC_4 left in the brine and that lost with the vented CO_2 . Pumping power is charged for boosting brine pressure from the last flash to the selected discharge pressure of 25 psia. Power lost in the hydraulic turbine due to expansion to a pressure higher than 25 psia is accounted for. Compressor power, as a function of the pressure ratio and mass flow, is determined by flash pressures, incoming brine composition and condensation pressure. These analyses did not account for any CO_2 returned to the power loop with the condensed IC_4 . The CO_2 does affect the power loop and, to a lesser extent, the recovery system itself but it does not affect the comparison of the three configurations. The results of these evaluations are shown as the system operating cost versus compressor power in Figures 4.17, 4.18 and 4.19. These figures also show the contribution of the IC_4 loss, compressor power and equipment costs, and the power penalties from the hydraulic turbine and the brine return pump. The three cost figures show that the two-stage flash with a tailstock pressure of 35 psia is the best performer by 30%.

4.5.3.3 Optimization of Two-Stage Flash with Condensation

To develop the total picture of the two-stage flash with condensation recovery, an iterative computer program was developed so the effect of CO_2 returned to the power loop could be accounted for. Both the effect on the power loop condensation pressure and the increase in CO_2 level in the brine leaving the DCHX were included. The results of the optimization, including the effects of the recirculated CO_2 , are shown in Figure 4.20 and comparison with the

SINGLE STAGE FLASH

FIGURE 4.14

TWO STAGE FLASH

FIGURE 4.15

TWO STAGE FLASH FIRST STAGE TO POWER CONDENSER

FIGURE 4.16

OPERATING COSTS FOR A SINGLE STAGE FLASH RECOVERY SYSTEM WITH 95°F CONDENSER

TAILSTOCK PRESSURE = 5.5 TO 50 PSIA.

CO₂ = 300 PPM, BRINE FLOW = 10⁶ LB/HR

IC₄ = 200 PPM

EFFECT OF
CO₂ RETURN
TO POWER
LOOP
NEGLECTED

FIG 4.17

OPERATING COST OF A TWO-STAGE FLASH RECOVERY SYSTEM WITH A 95°F CONDENSER

TAILSTOCK	P = 35 PSIA
RECOVERY TANK	P = 6 TO 25.5 PSIA
TEMPERATURE	= 140°F
BRINE FLOW RATE	= 10 ⁶ LB/HR
CO ₂	= 300 PPM
IC ₄	= 200 PPM

FIG. 4.18

OPERATING COST OF TWO-STAGE FLASH
RECOVERY SYSTEM WITH 95° CONDENSER
AND WITH FIRST STAGE FLASH TO
POWER CONDENSER

TAILSTOCK $P = 35$ PSIA
 RECOVERY TANK $P = 6$ TO 25.5 PSIA
 TEMPERATURE $= 140^{\circ}\text{F}$
 BRINE FLOW RATE $= 10^6$ LB/HR
 CO₂ $= 300$ PPM
 IC₄ $= 200$ PPM

FIG. 4.19

OPERATING COSTS OF A TWO-STAGE FLASH RECOVERY SYSTEM WITH 95°F CONDENSER

FIG. 4.20

preliminary evaluation shown in Figure 4.19 shows an operating cost of \$7.96/hr, an increase of \$2.70/hr when the CO_2 effects are accounted for including a shift in the optimum operating point. A very interesting aspect of the condensation study that is not illustrated in Figure 4.19 is that the majority of the loci of minimum costs occur for a range of condensation pressures where all of the CO_2 is absorbed in the condensed IC_4 and none is vented. Figure 4.21 shows IC_4 loss and CO_2 returned to the power loop for different second stage flash pressures as a function of compressor power. Where the CO_2 or IC_4 total loss curves flatten off, the condensing pressure is high enough for all the CO_2 to be absorbed and returned to the power loop with the IC_4 and there is no condenser vent. The locus of points of minimum total isobutane loss is also indicated on the figure. Figure 4.22 shows that the minimum cost total of IC_4 loss and system power follows the points of total CO_2 absorption, or no vent, until flash pressures of 8 psia or less are encountered. Adding the compressor costs, a flash tank cost and brine pump costs to the curve labeled "Cost of IC_4 Loss and Power Loss" in Figure 4.22 results in the total equivalent recovery system operating cost curve of Figure 4.20.

4.5.3.4 Optimization of Two-Stage Flash with 35°F Condensation

Condensation at a lower temperature using refrigeration was explored to ascertain the performance of a lower temperature condenser. A water chiller is the most practical configuration to provide the low temperature condensation source and it has a lower limit of 35°F to prevent freezing the water vapor. The evaluation for the 35°F condensing source produced the results shown in Figures 4.23, 4.24 and 4.25. It can be seen that for the 35°F condenser the optimum operating cost is very close to that for the 95°F condenser and the minimum cost point also occurs with all the CO_2 absorbed and returned with the IC_4 to the power loop. The lower temperature condenser allows the optimum condensing pressures to drop for a given set of flash conditions and the power saved in compression more than offsets the cost of the refrigeration.

Two types of refrigeration cooling were evaluated, a conven-

RECOVERY FLOWS FOR A TWO STAGE FLASH
RECOVERY SYSTEM WITH A 95°F CONDENSER

FIG. 4.21

COST OF IC₄ LOSS AND POWER
LOSS OF A TWO STAGE FLASH
RECOVERY SYSTEM WITH A 95°F CONDENSER

- COST OF IC₄ LOSS AND POWER LOSS AT
 CONSTANT FINAL FLASH PRESSURE.
 - - - - - COST OF POWER LOSS DUE TO HIGHER NON-
 CONDENSIBLE PRESS. IN THE CONDENSER.
 ———— COST OF IC₄ LOSS

FIG 4.22

RECOVERY FLOWS FOR A TWO STAGE FLASH
RECOVERY SYSTEM WITH A 35°F CONDENSER

— IC_4 LOSS (TOTAL); - - CO_2 ADDED TO WORKING LOOP

FIG 4.23

COST OF IC₄ LOSS AND POWER
LOSS OF A TWO STAGE FLASH
RECOVERY SYSTEM WITH A 35°F CONDENSER

FIG 4.24

OPERATING COSTS OF A TWO STAGE RECOVERY SYSTEM WITH A 35°F CONDENSER

FIG 4.25

tional electrically powered R-12 system and an absorption unit using brine as a heat source. Either unit has a relatively constant heat load for all of the evaluated conditions since the mass of IC_4 condensed does not vary appreciably. A heat load capability of 5 tons handles system requirements and was used to derive operating and equipment costs. The electric powered unit required 7 kW and the heat powered absorption unit 1,000 lbs/hr of brine assuming a COP of 0.6. Based on the APRR catalog, Reference 8, equipment costs were estimated at \$3,300 for each. The operating and equipment costs thus work out to an equivalent \$0.57/hr for the electric-powered and \$0.34/hr for the brine-powered absorption refrigeration system.

4.5.3.5 Direct Return of Two-Stage Flash to Power Condenser

Optimization of the compression and condensation systems resulted in a minimum cost operational zone wherein all the CO_2 was absorbed in the condensed IC_4 and there was no noncondensable gas venting. A two-stage flash whose gas streams are piped directly to the power loop (Figure 4.26) has many attractive features. Compressor power is minimized, no additional separation equipment is required and the power system losses due to the returned CO_2 are the same as for the optimum condenser system with no vent. Obviously with less equipment and extreme simplicity, maintenance, operating and downtime costs are minimal. The results of optimizing the operating conditions for the two-stage direct return flash recovery systems are shown in Figure 4.27. System operating costs are down to \$6.90/hr versus a minimum of \$7.60/hr primarily because of reduced compressor power for a separate condenser.

4.5.4 Distillation

A distillation column has the potential for providing higher purity IC_4 for return to the power loop than the other systems. A typical system configuration is shown in Figure 4.28. The vent gas from the flash stripper is sent to a distillation column where it is processed with the liquid reflux. The reflux liquid absorbs part of the gas stream and carries it to the bottom of the column where it is separated by the reboiler. The effluent stream, labeled bot-

"DIRECT RETURN" RECOVERY SYSTEM

FIG. 4.26

OPERATING COST OF TWO STAGE FLASH WITH DIRECT RETURN TO POWER CONDENSER

FIG. 4.27

DISTILLATION SYSTEM

FIG. 4.28

tom products, from the reboiler can be adjusted by process conditions to provide an IC_4 stream containing a very low level of CO_2 . This high level of purity is the major asset of this process. In adjusting the system to provide the high purity IC_4 , the amount of CO_2 vented is increased. With equilibrium conditions, the CO_2 vented is equal to the difference in CO_2 flow of the incoming stream and that contained in the liquid leaving the system. The distillate products shown leaving the reflux accumulator contain high levels of CO_2 and for maximum performance would not be removed but would just be added to the reflux stream.

This system does provide clean IC_4 but when the CO_2 is vented, IC_4 goes with it. The higher the purity of the IC_4 the higher the loss rate of the IC_4 . Evaluating the operating limits with either all CO_2 being vented or no CO_2 vented, it is apparent that the distillation process is not applicable for a direct contact plant. Adjusting conditions for the minimum IC_4 loss in the vent stream, a $35^{\circ}F$ condensing temperature and 200 psia, the IC_4 vent loss would be 14 lbs/hr at an equivalent cost of \$7.00/hr. The other extreme is zero vent of CO_2 and no IC_4 loss with an equivalent operating cost of \$3.40/hr due to the increased CO_2 level in the power loop. Costs for intermediate operating conditions would lie between these two extremes. These costs neglect the power required by the compressor but the result is identical to the condensation method of separation in that the most cost effective condition is with all the CO_2 returning to the loop so that the maximum amount of IC_4 can be recovered.

4.5.5 Recovery Approach Comparison

The practical approaches for recovery of working fluid from the brine discharge of a direct contact plant include configurations with a vacuum flash to strip the working fluid from the brine and several alternate separation techniques required to remove the non-condensibles stripped with the IC_4 from the brine. The practical configurations are summarized in Table 4.1 with their equipment costs and equivalent operating costs. The vacuum flash stripping accounts for approximately \$19,800 in equipment cost less compressor for all

TABLE 4.1

DIRECT CONTACT IC₄ RECOVERY SYSTEMS
COST COMPARISON

	<u>Equipment Costs</u>	<u>Power Costs</u>	<u>Lost IC₄ Costs</u>	<u>Total Equivalent Operating Costs</u>
Two-Stage Flash with Permeable Membrane*	\$181,800	\$4.91/hr	\$4.55/hr	2.8 mils/kW-hr
Two-Stage Flash with CO ₂ Absorption*	\$257,400	\$2.44/hr	\$4.55/hr	2.7 mils/kW-hr
Two-Stage Flash with IC ₄ Absorption (Lean Oil)*	\$285,800	\$4.69 /hr	\$4.55/hr	3.3 mils/kW-hr
Two-Stage Flash with 95°F Condenser	\$ 29,500	\$4.37/hr	\$2.90/hr	1.6 mils/kW-hr
Two-Stage Flash with 35°F Condenser & Electric Powered Refrig.	\$ 30,800	\$4.11/hr	\$2.90/hr	1.6 mils/kW-hr
Absorption Refrig.	\$ 30,800	\$3.96/hr	\$2.90/hr	1.5 mils/kW-hr
Two-Stage Flash with Direct Return to Power Loop	\$ 27,400	\$3.89/hr	\$2.25/hr	1.4 mils/kW-hr

* Systems so marked were not penalized for CO₂ returned to power loop

of the systems and \$1.35/hr operating cost for the brine return pump.

4.5.6 Noncondensable Gas Utilization

The potential of selling the separated CO₂ was investigated by establishing its economic value. Union Chemical of Los Angeles was contacted to establish the wholesale value of industrial CO₂, which was \$32/ton in February 1982. At a production rate of 20 to 100 lbs/hr, this would not provide enough income to pay for processing equipment. It was assumed that the value of N₂ would be the same or less than that of CO₂, therefore, it was not pursued.

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4.7 APPENDICES

4.7.1 Appendix A - Theoretical Model of a Flash Stripping System

Consider a mixture with overall composition given by the mole fractions $Z_1, Z_2, \dots, Z_i$ at a temperature and pressure such that it consists of two phases in equilibrium, a liquid phase with mole fraction $X_1, X_2, \dots, X_i$ and a vapor phase with mole fractions $Y_1, Y_2, \dots, Y_i$.

Assume that F (moles/hr) of an n -component stream are introduced as feed into the flash vaporization tank.

The resulting vapor and liquid streams are withdrawn at the rates of V and L (moles/hr). The mole fractions of the components in the feed, vapor and liquid streams are designated as Z_i, Y_i and X_i respectively.

Assuming vapor-liquid equilibrium and steady-state operation, we have the following series of algebraic relationships:

$$Z_i F = X_i L + Y_i V \quad (\text{material balance for component } i)$$

$$Y_i = X_i K_i \quad (\text{equilibrium relation})$$

$$\sum Y_i = 1 \quad (\text{by definition})$$

$$\sum X_i = 1 \quad (\text{by definition})$$

$$F = V + L \quad (\text{material balance})$$

K_i is the equilibrium constant for the i -th component, at the temperature and pressure in the tank. K_i values can be found by using Raoult's and Henry's laws.

$$K_i = H_i / P_{\text{tot}} \quad (H = \text{Henry's constant for a very dilute solution of } i\text{-component})$$

$$K_i = P_{\text{sat}_i} / p_t \quad (P_{\text{sat}} = \text{vapor pressure of } i\text{-component at tank temperature})$$

Thus there are $2_i + 2$ equations and $2_i + 2$ unknowns, and the state of the system is completely determined. That this should be the case is a consequence of Duhem's Theorem, which states that for any closed system formed initially from a given mass of particular chemical species the equilibrium state is completely determined when any two independent variables are fixed.

Although an iterative technique is required to solve this system of equations in general form, algebraic solution is possible for three component mixtures.

Assume dilute solution of CO_2 and IC_4 in water.

$$\begin{aligned} Z_{\text{CO}_2} &= Y_{\text{CO}_2} \frac{V}{F} + X_{\text{CO}_2} \frac{L}{F} \\ Z_{\text{IC}_4} &= Y_{\text{IC}_4} \frac{V}{F} + X_{\text{IC}_4} \frac{L}{F} \\ Z_{\text{H}_2\text{O}} &= Y_{\text{H}_2\text{O}} \frac{V}{F} + X_{\text{H}_2\text{O}} \frac{L}{F} \end{aligned} \quad \left. \vphantom{\begin{aligned} Z_{\text{CO}_2} \\ Z_{\text{IC}_4} \\ Z_{\text{H}_2\text{O}} \end{aligned}} \right\} \text{MATERIAL BALANCES}$$

$$Y_{\text{CO}_2} = \frac{H_{\text{CO}_2-\text{H}_2\text{O}}}{P_T} X_{\text{CO}_2} \quad \left. \vphantom{Y_{\text{CO}_2}} \right\} \text{EQUILIBRIUM RELATION}$$

$$\left. \begin{aligned} Y_{IC_4} &= \frac{H_{IC_4-H_2O}}{P_T} X_{IC_4} \\ Y_{H_2O} &= \frac{P_{SAT H_2O}}{P_T} X_{H_2O} \end{aligned} \right\} \text{EQUILIBRIUM RELATIONS}$$

$$\left. \begin{aligned} Y_{CO_2} + Y_{IC_4} + Y_{H_2O} &= 1 \\ X_{CO_2} + X_{IC_4} + X_{H_2O} &= 1 \end{aligned} \right\} \text{BY DEFINITION}$$

Using algebra, the following solution for each unknown is found:

$$Z_{CO_2} = Y_{CO_2} \frac{V}{F} + X_{CO_2} \left(1 - \frac{V}{F}\right) = X_{CO_2} \left[\left(\frac{H_{CO_2-H_2O}}{P_T} - 1 \right) \frac{V}{F} + 1 \right]$$

$$Z_{IC_4} = X_{IC_4} \left[\left(\frac{H_{IC_4-H_2O}}{P_T} - 1 \right) \frac{V}{F} + 1 \right]$$

$$Z_{H_2O} = X_{H_2O} \left[\left(\frac{P_{SAT H_2O}}{P_T} - 1 \right) \frac{V}{F} + 1 \right]$$

$$X_{CO_2} \frac{H_{CO_2-H_2O}}{P_T} + X_{IC_4} \frac{H_{IC_4-H_2O}}{P_T} + X_{H_2O} \frac{P_{SAT H_2O}}{P_T} =$$

$$X_{CO_2} + X_{IC_4} + X_{H_2O}$$

$$X_{CO_2} \left(\frac{H_{CO_2-H_2O}}{P_T} - 1 \right) + X_{IC_4} \left(\frac{H_{IC_4-H_2O}}{P_T} - 1 \right) + X_{H_2O} \left(\frac{P_{SAT H_2O}}{P_T} - 1 \right) = 0$$

$$\frac{Z_{CO_2} \left(\frac{H_{CO_2-H_2O}}{P_T} - 1 \right)}{\frac{V}{F} \left(\frac{H_{CO_2-H_2O}}{P_T} - 1 \right) + 1} + \frac{Z_{IC_4} \left(\frac{H_{IC_4-H_2O}}{P_T} - 1 \right)}{\frac{V}{F} \left(\frac{H_{IC_4-H_2O}}{P_T} - 1 \right) + 1} + \frac{Z_{H_2O} \left(\frac{P_{SAT H_2O}}{P_T} - 1 \right)}{\frac{V}{F} \left(\frac{P_{SAT H_2O}}{P_T} - 1 \right) + 1} = 0$$

$$A = Z_{\text{CO}_2} \left(\frac{P_T}{H_{\text{IC}_4} - P_T} + \frac{P_T}{P_{\text{SAT H}_2\text{O}} - P_T} \right) + Z_{\text{IC}_4} \left(\frac{P_T}{H_{\text{CO}_2} - P_T} + \frac{P_T}{P_{\text{SAT H}_2\text{O}} - P_T} \right) + Z_{\text{H}_2\text{O}} \left(\frac{P_T}{H_{\text{CO}_2} - P_T} + \frac{P_T}{H_{\text{IC}_4} - P_T} \right)$$

$$B = \frac{Z_{\text{CO}_2} P_T^2}{(H_{\text{IC}_4} - P_T)(P_{\text{SAT H}_2\text{O}} - P_T)} + \frac{Z_{\text{IC}_4} P_T^2}{(H_{\text{CO}_2} - P_T)(P_{\text{SAT H}_2\text{O}} - P_T)} + \frac{Z_{\text{H}_2\text{O}} P_T^2}{(H_{\text{CO}_2} - P_T)(H_{\text{IC}_4} - P_T)}$$

$$\left(\frac{V}{F} \right)^2 + A \left(\frac{V}{F} \right) + B = 0$$

$$\frac{V}{F} = \frac{A}{2} + \sqrt{\frac{A^2}{4} - B}$$

$$X_{\text{CO}_2} = \frac{Z_{\text{CO}_2}}{\left[\left(\frac{H_{\text{CO}_2}}{P_T} - 1 \right) \frac{V}{F} + 1 \right]}$$

$$X_{\text{IC}_4} = \frac{Z_{\text{IC}_4}}{\left[\left(\frac{H_{\text{IC}_4}}{P_T} - 1 \right) \frac{V}{F} + 1 \right]}$$

$$X_{\text{H}_2\text{O}} = 1 - X_{\text{CO}_2} - X_{\text{IC}_4}$$

$$Y_{\text{CO}_2} = \left(\frac{H_{\text{CO}_2}}{P_T} \right) X_{\text{CO}_2}$$

$$Y_{IC4} = \left(F_{IC4} / F_T \right) X_{IC4}$$

$$Y_{H_2O} = 1 - Y_{CO_2} - Y_{IC4}$$

$$L/F = 1 - V/F$$

4.7.2 Appendix B - Theoretical Model of the Recovery Condenser

Two essentially immiscible liquid phases are present in the recovery condenser, H_2O and IC_4 . A vapor phase is common to both liquid phases. If composition of the vapor in the condenser feed stream and condensing temperature and pressure are known, the equilibrium phase concentrations in the liquid and vent stream can then be determined.

FIG. 2

From the material balance the following set of equations can be derived:

$$Z_{CO_2} = Y_{CO_2} \frac{V}{F} + X_{CO_2-H_2O} \frac{L_{H_2O}}{F} + X_{CO_2-IC_4} \frac{L_{IC_4}}{F}$$

$$Z_{IC_4} = Y_{IC_4} \frac{V}{F} + X_{IC_4} \frac{L_{IC_4}}{F}$$

$$Z_{H_2O} = Y_{H_2O} \frac{V}{F} + X_{H_2O} \frac{L_{H_2O}}{F}$$

The following set of equations comes from the equilibrium relations:

$$Y_{IC_4} = \left(\frac{P_{SAT\ IC_4}}{P_C} \right) X_{IC_4}$$

$$Y_{H_2O} = \left(\frac{P_{SAT\ H_2O}}{P_C} \right) X_{H_2O}$$

$$Y_{CO_2} = \left(\frac{H_{CO_2-H_2O}}{P_C} \right) X_{CO_2-H_2O}$$

$$Y_{CO_2} = \left(\frac{H_{IC_4-H_2O}}{P_C} \right) X_{CO_2-IC_4}$$

By definition the following equations are true:

$$Y_{CO_2} + Y_{IC_4} + Y_{H_2O} = 1$$

$$X_{CO_2-IC_4} + X_{IC_4} = 1$$

$$X_{CO_2-H_2O} + X_{H_2O} = 1$$

Now by algebraic manipulation, a solution for each unknown can be found:

$$Y_{CO_2} = 1 - \frac{P_{IC_4}}{P_C} X_{IC_4} - \frac{P_{SATH_2O}}{P_C} X_{H_2O}$$

$$= 1 - \frac{P_{IC_4}}{P_C} \left(1 - \frac{Y_{CO_2} P_C}{H_{CO_2-IC_4}} \right) - \frac{P_{SATH_2O}}{P_C} \left(1 - \frac{Y_{CO_2} P_C}{H_{CO_2-H_2O}} \right)$$

$$= 1 - \frac{P_{IC_4}}{P_C} - \frac{P_{SATH_2O}}{P_C} + Y_{CO_2} \left(\frac{P_{SATIC_4}}{H_{CO_2-IC_4}} + \frac{P_{SATH_2O}}{H_{CO_2-H_2O}} \right)$$

$$\therefore Y_{CO_2} = \frac{1 - \frac{P_{SATIC_4}}{P_C} - \frac{P_{SATH_2O}}{P_C}}{1 - \frac{P_{SATIC_4}}{H_{CO_2-IC_4}} - \frac{P_{SATH_2O}}{H_{CO_2-H_2O}}}$$

$$Z_{CO_2} = Y_{CO_2} \frac{V}{F} + \frac{Y_{CO_2} P_C}{H_{CO_2-H_2O}} \left(\frac{Z_{H_2O} - \frac{P_{SAT H_2O}}{P_C} \left(1 - \frac{Y_{CO_2} P_C}{H_{CO_2-H_2O}} \right) \frac{V}{F}}{1 - \frac{Y_{CO_2} P_C}{H_{CO_2-H_2O}}} \right) +$$

$$\frac{Y_{CO_2} P_C}{H_{CO_2-IC_4}} \left(\frac{Z_{IC_4} - \frac{P_{SAT IC_4}}{P_C} \left(1 - \frac{Y_{CO_2} P_C}{H_{CO_2-IC_4}} \right) \frac{V}{F}}{1 - \frac{Y_{CO_2} P_C}{H_{CO_2-IC_4}}} \right)$$

$$= Y_{CO_2} \frac{V}{F} + Y_{CO_2} \left(\frac{P_C Z_{H_2O}}{H_{CO_2-H_2O} - Y_{CO_2} P_C} \right) - Y_{CO_2} \frac{V}{F} \left(\frac{P_{SAT H_2O}}{H_{CO_2-H_2O}} \right) +$$

$$Y_{CO_2} \left(\frac{P_C Z_{IC_4}}{H_{CO_2-H_2O} - Y_{CO_2} P_C} \right) - Y_{CO_2} \frac{V}{F} \left(\frac{P_{SAT IC_4}}{H_{CO_2-IC_4}} \right)$$

$$\frac{V}{F} = \frac{\frac{Z_{CO_2}}{Y_{CO_2}} - \frac{Z_{IC_4} P_C}{H_{CO_2-IC_4} Y_{CO_2} P_C} - \frac{Z_{H_2O} P_C}{H_{CO_2-H_2O} Y_{CO_2} P_C}}{1 - \frac{P_{SAT IC_4}}{H_{CO_2-IC_4}} - \frac{P_{SAT H_2O}}{H_{CO_2-H_2O}}}$$

If recovery condenser pressure increases vent stream (or V/F ratio) decreases. When the vent stream (or V/F ratio) is equal to zero, further increases of the condenser pressure is impossible, therefore condenser pressure is equal to the maximum pressure $P_{C_{max}}$.

Assume $P_C = P_{C_{max}}$ and $V/F = 0$: Then recovery condenser pressure can be found by algebraic solution:

$$Z_{CO_2} = X_{CO_2-H_2O} \frac{L_{H_2O}}{F} + X_{CO_2-IC_4} \frac{L_{IC_4}}{F}$$

$$Z_{IC_4} = X_{IC_4} \frac{L_{IC_4}}{F}$$

$$Z_{H_2O} = X_{H_2O} \frac{L_{H_2O}}{F}$$

$$X_{CO_2-H_2O} + X_{H_2O} = 1$$

$$X_{CO_2-IC_4} + X_{IC_4} = 1$$

$$Y_{CO_2} = \left(\frac{H_{CO_2-IC_4}}{P_{C_{MAX}}} \right) X_{CO_2-IC_4}$$

$$Y_{CO_2} = \left(\frac{H_{CO_2-H_2O}}{P_{C_{MAX}}} \right) X_{CO_2-H_2O}$$

$$Y_{IC_4} = \left(\frac{P_{SAT IC_4}}{P_{C_{MAX}}} \right) X_{IC_4}$$

$$Y_{H_2O} = \left(\frac{P_{SAT H_2O}}{P_{C MAX}} \right) X_{H_2O}$$

$$Y_{CO_2} + Y_{IC_4} + Y_{H_2O} = 1$$

$$Z_{CO_2} = X_{CO_2-H_2O} \left(\frac{Z_{CO_2}}{X_{H_2O}} \right) + X_{CO_2-IC_4} \left(\frac{Z_{IC_4}}{X_{IC_4}} \right)$$

$$= \frac{Y_{CO_2} \left(\frac{P_{C MAX}}{H_{CO_2-H_2O}} \right) Z_{H_2O}}{1 - \frac{Y_{CO_2} P_{C MAX}}{H_{CO_2-H_2O}}} + \frac{Y_{CO_2} \left(\frac{P_{C MAX}}{H_{CO_2-IC_4}} \right) Z_{IC_4}}{1 - \frac{Y_{CO_2} P_{C MAX}}{H_{CO_2-IC_4}}}$$

$$= Y_{CO_2} \left(\frac{P_{C MAX} Z_{H_2O}}{H_{CO_2-H_2O} - Y_{CO_2} P_{C MAX}} \right) + Y_{CO_2} \left(\frac{P_{C MAX} Z_{IC_4}}{H_{CO_2-IC_4} - Y_{CO_2} P_{C MAX}} \right)$$

$$Y_{CO_2} P_{C MAX} = \frac{P_{C MAX} - P_{SAT IC_4} - P_{SAT H_2O}}{1 - \frac{P_{SAT IC_4}}{H_{CO_2-IC_4}} - \frac{P_{SAT H_2O}}{H_{CO_2-H_2O}}}$$

$$Z_{CO_2} H_{CO_2-H_2O} H_{CO_2-IC_4} - Z_{CO_2} (Y_{CO_2} P_{C MAX}) H_{CO_2-IC_4} -$$

$$Z_{CO_2} (Y_{CO_2} P_{C MAX}) H_{CO_2-H_2O} + Z_{CO_2} (Y_{CO_2} P_C)^2 + Z_{H_2O} (Y_{CO_2} P_{C MAX})^2 -$$

$$Z_{H_2O} (Y_{CO_2} P_{C MAX}) H_{CO_2-IC_4} - Z_{IC_4} (Y_{CO_2} P_{C MAX}) H_{CO_2-H_2O} + Z_{IC_4} (Y_{CO_2} P_C)^2 = 0$$

$$(Y_{CO_2} P_C)^2 + (Z_{CO_2} H_{CO_2-H_2O} H_{CO_2-IC_4}) -$$

$$(Z_{CO_2} H_{CO_2-IC_4} + Z_{CO_2} H_{CO_2-H_2O} + Z_{IC_4} H_{CO_2-H_2O} + Z_{H_2O} H_{CO_2-IC_4})(Y_{CO_2} P_C) = 0$$

$$A = \frac{1}{2} \left[Z_{CO_2} (H_{CO_2-IC_4} + H_{CO_2-H_2O}) + Z_{IC_4} H_{CO_2-H_2O} + Z_{H_2O} H_{CO_2-IC_4} \right]$$

$$Y_{CO_2} P_{C_{MAX}} = A - \sqrt{A^2 - Z_{CO_2} H_{CO_2-IC_4} H_{CO_2-H_2O}}$$

$$P_{C_{MAX}} = (Y_{CO_2} P_{C_{MAX}}) \left(1 - \frac{P_{SAT IC_4}}{H_{CO_2-IC_4}} - \frac{P_{SAT H_2O}}{H_{CO_2-H_2O}} \right) + P_{SAT IC_4} + P_{SAT H_2O}$$

5.0 ALTERNATE TURBINE DESIGN AND EVALUATION

5.1 INTRODUCTION

Performance evaluation of the 500 kW plant indicated that the power turbine was performing in the 72% efficiency range. Design goal for the turbine was 83%. The design was evaluated and it was established that a modified design should be able to meet the performance goal. The original wheel was machined from aluminum because of cost considerations and schedule constraints. The decision was made to modify the design and fabricate a new wheel using titanium or 15-5 ph stainless. The two configurations and their performance are covered in the following section.

5.2 DISCUSSION

The original turbine for the 500 kW direct contact heat exchanger pilot plant was designed and built by Mafi Trench with a predicted performance efficiency of 83%. Performance calculations indicated that its efficiency was in the 72% range at design flow conditions and based on a supplier-predicted gearbox loss of 21 kW. Evaluation of this original turbine design indicated that the low efficiency could be attributed to the pressure balance design used to eliminate the turbine thrust loads. The balance approach used depended on a labyrinth seal at the OD of the wheel back shroud and axial holes drilled through the back shroud to pick up exhaust pressure near the exducer hub. This approach does an excellent job of reducing the thrust load; however, it does allow gas vapor from the inlet to leak through the labyrinth seal, behind the back shroud, through the holes and out to the exhaust. This leakage flow behind the turbine wheel passes over a cooled housing with the potential of condensing the isobutane working fluid vapors. Condensation occurring in this region would be thrown centrifugally out through the OD of the wheel and would be pumped through the labyrinth. The pumping of the liquid would consume additional power, thus reducing the net turbine output. Calculations indicated that it would not take a significant amount of condensation in this area to account for the observed low efficiency.

A decision was made to design and fabricate a new turbine wheel with a different approach to pressure balance that would eliminate the potential for the condensation behind the turbine wheel disk. Aerodynamically, there was very little difference in the design of the basic turbine geometry; however, the labyrinth and the balance holes were eliminated in favor of pump-out vanes to control thrust. This approach to thrust control is only partial; however, and an increase in axial turbine thrust from 340 pounds to 1000 pounds was predicted with the change in balancing approach. A new thrust bearing was designed and incorporated into the gearbox to handle the higher turbine thrust. The new turbine design was also configured so that it could be constructed of cast titanium or cast steel to achieve greater wear resistance than the previous design that used a machined aluminum impeller.

The first wheel tested of the new design was fabricated from cast titanium and only a limited amount of data was obtained before destruction of the wheel. The failure of the cast titanium wheel was the result of a poor casting and the high fatigue susceptibility of the titanium material. A second rotor, cast in 15-5 ph steel was installed. The testing of this rotor was limited by two seal failures which were the result of differential thermal expansion between the mating ring and shaft with insufficient clearances. The lack of adequate clearance occurred when the gearbox shaft was replaced after the titanium rotor failure.

Evaluation of the new turbine wheel's performance indicated that the gross alternator output was only slightly better than the original. The temperature of the exhaust gas indicated that the turbine should be performing better since the exit temperature was low and near the value predicted for the expected turbine efficiency. Further investigation indicated that the gearbox losses were much higher than the supplier-estimated 21 kW that had been used in all the previous turbine performance calculations. The gearbox losses were calculated based on estimated lubricant flow and the temperature rise of the lubricant as measured across the gearbox. With the original turbine, the temperature rise and flowrate indicate a gearbox loss of 48 kW which means that the turbine was 75% efficient at

the design flowrates. The gearbox losses with the large thrust bearing were calculated at 62 kW which means that the second design was performing in the 80% range. These gearbox losses are confirmed by calculating them from the turbine efficiencies derived from the gas properties and flowrates measured across the turbine.

The higher efficiency of the second turbine design could be utilized with proper modifications to the gearbox. By reversing the helix angle on the high speed gear mesh, the thrust loads of the turbine could be balanced by the gear thrust loads. In the present design, the two thrust loads are additive. With the reduced thrust load, a small thrust bearing can be used with a large reduction in parasitic horsepower. Based on present industry data, it also appears that the gearbox was designed on a very conservative basis and that the losses could be reduced to the originally predicted 20 kW level.

6.0 DOWNWELL PUMP

6.1 INTRODUCTION

As a part of the 500 kW direct contact pilot plant program, there was an ancillary effort with REDA Pump of Bartlesville, Oklahoma to develop and obtain operating experience with a submersible, electric pump in a geothermal well. In addition to supporting the development of a pump for geothermal use, the 500 kW pilot plant could operate at design conditions using the wells at East Mesa if these wells were pumped. The following discussion summarizes the results of the REDA downwell pump effort.

6.2 DISCUSSION

The wells at East Mesa could not provide the 200 minimum gallons per minute required by the direct contact pilot plant in a free flowing basis. The plant also required that the geothermal brine be delivered at sufficient pressure to prevent any release of the CO₂ contained in the brine or any flashing in the well bore. This dictated a pump that would provide over 200 gpm with a minimum

pressure, at the 500 kW pad, of 200 psig or better.

During the early part of the program, REDA Pump of Bartlesville had indicated a strong interest in developing a line of geothermal pumps and offered to provide the engineering and modifications required to build a Model G220-540 series pumps that would perform in the geothermal environment. This pump used an 80 hp motor and 27 stages that would provide the required flow and head. An agreement was made with REDA that the program would purchase one of these modified pumps and install it in the wells at East Mesa providing that REDA would demonstrate that the pump could operate without degradation for a 30-day period in a simulated thermal environment. REDA did demonstrate that their modified pump could meet this 30-day endurance requirement with thermal cycling and sustained periods of operation at temperatures of 370°F.

A modified pump was procured in 1979 and installed in well 8-1 in East Mesa. The first pump failed in 315 hours with a short in the pothead assembly, that brings power into the motor protector assembly, and a burned out motor. This pump was refurbished with a new motor and reinstalled in November 1979. It proceeded to run successfully for 4000 hours. After the 4000 hours, the pump could not be restarted after a power outage caused shutdown. The pump was pulled and inspection revealed a half gallon of water in the motor and that the motor dielectric was breaking down due to the water. The pump had some scale buildup but it did not appear to be sufficient to have caused the high motor overload that prevented restarting the pump in the well. Five more pumps were installed between May 1980 and October 1981, all of which were plagued by poor performance.

A summary of pump operating experience is given in Table 6.1. Some of it is attributed to poor installation procedures as well as component failures in the electrical system. Evaluation of the problems encountered and some of the potential weaknesses of the hardware resulted in the conclusion that an active oil pressurization system could protect the pump motor and protector assembly. Of the numerous failures in which water had penetrated the motor and protector

Build Number	Installation (Start) Date	Failure (Removal) Date	Well Number	Set Depth Ft.	Brine Temp. °F	Operation Time Hours	Run and Failure Notes
1	7/21/79 (8/14/79)	9/25/79 (10/24/79)	8-1	620	333	315	Overload. Short phase to ground at wellhead. RTV sealing around production tube decomposed. Approximately 2 pints of H ₂ O in motor. Cable OK. Short in pothead. Motor burnout in stator. Winding shot. Suspected insulation weakness.
2	11/16/79 (11/17/79)	5/05/80 (5/19/80)	8-1	620	333	4052	Overload. Short phase to ground. Much carbonate scale buildup on pump inlet screen. Pump seized. One-half gallon H ₂ O in motor. Pothead OK. Motor case dielectric wet.
3	6/25/80 (6/26/80)	8/26/80 (9/22/80)	8-1	620	333	1481	Overload. Short in packoff. No H ₂ O circulation barrier in packoff. Pump and cable still OK but were deteriorating. No water in protector. Some water in motor.
4	8/21/80 (8/22/80)	8/26/80 12/18/80 (9/18/80) (1/15/81)	6-2 6-2	762 731	350 350	100 2184	Overload. Short in packoff. No H ₂ O circulation barrier. Lead sheath in packoff had cut through the insulation. Packoff was repaired with the pump still in the well. Shutdown. Cable at 150 K Ω at shutdown and 500 K Ω at removal. Bubble tube was dropped during removal, possibly damaging cable. Bubble tube failed in a Sch. 40 welded pipe nipple. Less than 1 pint H ₂ O in motor. Motor windings OK.
5	1/16/81 (1/17/81) (2/06/81)	1/31/81 2/06/81 2/11/81 (2/10/81) (2/17/81)	6-2 6-2 6-2	762 731 700	350 350 350	336 1/2 4.5	Underload. Cable meggered at 70 K Ω . Brine leakage at coupling to wellhead weld. Water at top and bottom of packoff; packoff reinstalled at wellhead. All repairs completed with pump in well. Shutdown because of brine leak at wellhead coupling. Weld procedure was improper. Cable damaged during repairs. All repairs completed with pump in well. After repairs, pump meggered at 4.5 M Ω . Flow rate increased and pump failed. Cable meggered at L.T. 3K Ω . Packoff OK. Cable appeared damaged. Cracked solder at pothead. One gallon of H ₂ O in motor. Pump thrust bearing bad. Motor head and stator rusty.
6	3/17/81 (3/18/82)	6/27/81 (7/24/81)	6-2	729	350	2352	Power outage. Meggered at 50 K Ω . One pint of H ₂ O in motor. Much foreign matter in motor oil. Motor thrust washers overheated. Heat-caused 'blueing' seen in motor shaft and rotor ends.
7	9/04/81 (9/08/81)	9/14/81 (9/22/81)	6-2	739	350	136	Shutdown. Motor pressure system out of oil. Short phase to ground. Packoff OK. Oil tube broken below wellhead. O-ring at pothead shaved. Three quarts H ₂ O in motor. Electrical failure in cable. Pump had seized for unknown reasons.
8	(10/23/81) (10/23/81)	9/22/82 (10/04/82)	6-2	615	350	8000	Pump performance degraded during run. Performance decreased to point where pump could not supply plant needs. B/N oil pressurization system worked. Field inspection showed 2000 M Ω in old motor. Pump 'worn out' because of sand.

TABLE 6.1

Build Number	Installation (Start) Date	Failure (Removal) Date	Well Number	Set Depth Ft.	Brine Temp. °F	Operation Time Hours	Run and Failure Notes
9	10/06/82 (10/07/82)	1/04/83	6-2	615	350	1176	Shut down 11/24/81, end of 500 kW testing.

assemblies were several occurrences in which the source of the leakage path for the water into the motor could not be established.

The present power cable for these downwell pumps is a composite structure consisting of insulation wrapped with EPDM rubber, a lead sheath over the EDPM and a spiral steel shield over the lead. With time, water penetrates to the insulation either through mechanical damage in the outer layers or through manufacturing flaws. The water provides an electrical leakage path in the insulation of the power cables to ground resulting in insulation resistance loss to ground from the power leads. In evaluating the cable problem, various vendors and REDA were consulted. The most attractive approach to solving the cable problem is one in which the cable is contained within a welded tube. Several cable manufacturers through they had processes with which they could fabricate cable in the size required.

Because of the time constraint in the program, it was decided that Barber-Nichols would design and fabricate an oil pressurization system for the motor and protector and relegate to a later time the engineering and procurement for a new type of cable. The eighth pump installed at the East Mesa facility was installed with an oil pressurization system. The system, which is surface-mounted, provided controlled oil pressure to the motor and protector assemblies of the downwell pump through a feed pump. The system was designed to sense the downwell pressure at the pump and control the oil pressure that was fed to the pump to a 15 psi delta above the pump ambient. This system would maintain this positive differential during operation and during shutdown and would follow any changes in the ambient pressure resulting from changes in flowrate or ground reservoir conditions.

This pump ran for approximately 8000 hours before it had to be removed because of falling production. Figure 6.1 shows loss of production capability of the pump starting from its installation and at the end prior to being pulled. During this operational period, the insulation resistance of the cable motor protector assembly was monitored. Readings of the insulation resistance was checked every time the pump was shut down. All of the early shutdowns were due to

REDA PUMP #8 PERFORMANCE HISTORY

power interruption. The measured insulation resistance history is shown in Figure 6.2 and shows the steady decrease of the insulation resistance with operating time in the well. In the late months of operation, the insulation resistance showed an interesting phenomena. A resistance measurement made at the time of shutdown would be lower than the succeeding measurements of the insulation resistance with time while the pump was down. Typically, in the last few months of operation, insulation resistance would be in a range of 60,000 to 90,000 ohms at the point of shutdown and within a few hours would be up to 100,000 ohms and within a day would be as high as 300,000 ohms. During the period of time between final shutdown of the pump and its actual removal from the well, the insulation resistance had increased to almost 2 megaohms.

Inspection of the removed submersible pump revealed the motor protector assembly was in excellent shape with meg readings of the pump and the pothead exceeding 2000 megaohms to ground. The pumping section itself had experienced considerable wear with some of the stages being worn down to the point that their original configurations were not even recognizable. Pictures of the damaged hardware are shown in Figures 6.3 and 6.4. The extreme wear of the impellers results from the design approach to balance the thrust loads on the stages. The balancing approach uses labyrinth seals on the shroud. The impeller on the left in the pictures still has its shroud. The one on the right has worn away its shroud and most of the vanes. The fine sand that was being pumped in the brine abraded the sealing labyrinth allowing thrust loads to increase which produced the extreme wear with time as shown in the photograph.

This aspect of the pilot plant program was partially successful in that it demonstrated that an electric submersible pump does have the potential to be used successfully in moderate temperature geothermal brines. Review of the experience gained shows that a pump properly designed for this service, manufactured and installed with the required quality and its control, is the criteria for a useful service life.

FIGURE 6.2

REDA PUMP NO. 8 - EAST MESA WELL 6-2
MOTOR AND CABLE MEGGAR READINGS

BARBER-NICHOLS
ENGINEERING CO.

FIGURE 6.3

FIGURE 6.4