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APPLICATION OF DIRECT CONTACT HEAT
EXCHANGERS TO GEOTHERMAL POWER
PRODUCTION CYCLES

Project Review, December 1, 1974—May 31, 1977

By
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R. F. Boehm
A. C. Hansen

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ABSTRACT

Work performed at the University on the development of direct contact heat exchanger power cycles for geothermal applications is reviewed. The period covered in the report is from the inception of the project in 1974 through May 31, 1977. Results from a large experimental program on heat exchanger development as well as from many analyses of components and cycle performance and economics are given. A number of working fluids and operating conditions have been considered, and no major obstacles for the implementation of the concept have been discovered.

TABLE OF CONTENTS

ABSTRACT	
SUMMARY AND CONCLUSIONS	
CHAPTER I. INTRODUCTION	1
A. Background	1
B. Comparisons to Other Programs	5
C. Report Philosophy	7
CHAPTER II. SYSTEM ANALYSIS	8
A. Introduction	8
B. Direct Contact Cycle Design Consideration	17
C. Direct Contact Geothermal Cycle Analysis	20
1. System Components	27
a. Direct Contact Boiler	27
b. Binary Vapor Turbine	27
c. Condenser	28
d. Preheaters	28
2. Thermodynamic Properties	28
a. Subroutine Water	29
b. Subroutine Carbon	29
c. Subroutine Freon	30
d. Potential Fluids	31
e. Transport Properties	32
3. Cycle Performance	32
CHAPTER III. COMPONENT DESIGN, TESTING AND PERFORMANCE	38
A. Boilers	38
1. Bench Tests	38
a. Introduction	38
b. Volume-Type Boiler	39
c. Surface-Type Boilers	45
d. Further Work Plans	50

2.	Large Scale Boiler Tests	52
a.	Boiler Design	52
b.	Results and Discussion	63
i.	Heat Transfer Coefficients	64
ii.	Mass Flow Ratios	65
iii.	Approach Temperature in the Boiler	71
iv.	Heat Transfer Correlation	74
v.	Comparison with Other Investigators	76
B.	Turbines	77
1.	Performance Considerations	77
2.	Relative Cost Estimates	88
C.	Preheaters	97
1.	Overall Considerations	97
2.	Parallel-Flow Concepts	98
a.	Stratified Types	98
b.	Mixed Types	98
3.	Counterflow Configurations	102
a.	Spray Tower	104
b.	Baffled Towers	105
c.	Perforated Plate Towers	107
d.	Packed Towers	107
e.	Wetted Wall Towers	109
4.	Comparisons of Various Configurations	110
D.	Direct Contact Condensers	115
1.	Introduction	115
2.	Direct Contact Condensing Devices - Heat Transfer Considerations	116
a.	Saturated Vapor	117
i.	Film Type Direct Contact Condensers	118
ii.	Drop Type Direct Contact Condensers	118
iii.	Bubble Type Direct Contact Condensers	120
b.	Superheated Vapor	124
3.	Recommendations on Condenser Related Work	125
CHAPTER IV.	WORKING FLUID LOSS CONSIDERATIONS	127
A.	Working Fluid Solubility in Brine	127
B.	Effects of Noncondensibles	135

C.	Recovery of Working Fluid From Spent Brine	137
CHAPTER V.	ECONOMIC ANALYSIS	141
A.	Vapor Turbines	142
B.	Pumps	143
C.	Conventional Heat Exchangers	143
D.	Direct Contact Heat Exchangers and Pertinent Vessels	144
E.	Wells	147
F.	Piping Costs and Auxiliary Equipment	150
G.	Operating and Maintenance	152
H.	Conceptual Design and Evaluation of a 50 MWe Plant .	153
I.	Comparison of DCHE and STHE Plants	158
J.	Comparison of Isobutane and Pentane DCHE Cycles . .	158
REFERENCES		161
APPENDIX:	SUMMARY OF THE LARGE-SCALE DIRECT CONTACT BOILER TEST RESULTS	166

SUMMARY

Work performed on the University of Utah Direct Contact Power Generation concept from its inception through May 31, 1977 is summarized. It is shown that the program is very broad in its scope such that virtually all of the tools are developed that are necessary to accomplish rational design and costing of direct contact heat exchanger plants. This program is the only one of the various direct contact projects that is not limited in scope to a particular fluid so that the various comparisons of fluids and operating conditions can be made.

Details given in this report support the following conclusions:

1. Approaches for the economical utilization of hypersaline geothermal brines for production of power have been lacking. The direct contact heat exchanger cycle offers a solution to this problem.
2. While simple hydrocarbon fluids (particularly pentane and isobutane) are generally preferred for use in direct contact power plants, halogenated hydrocarbons may definitely be preferred for application where safety is a primary consideration. Small power plants are examples of this latter category.
3. From cost and performance standpoints, pentane is the working fluid to use for wells below (about) 300F, while isobutane appears to be a better choice for higher temperature wells.
4. An analytical technique has been developed to predict turbine performance resulting from the flow of the two fluid vapors.
5. For net power outputs up to 10 Mw, a single stage turbine unit with rotor less than 6 feet in diameter will be required for virtually all candidate fluids considered for direct contact heat exchanger cycles.
6. Turbine costs were estimated for a wide variety of turbine inlet temperatures and working fluids. Cost rankings (from lowest to highest) were as follows: propane, isobutane, butane, Freon 114, pentane and Freon 113. The cost

6. (cont'd) variations were predicted to range over two orders of magnitude between the highest and lowest. However, turbine cost will probably not be a dominating factor on plant cost.
7. Boiler tests have been performed with both a spray column configuration ("volume boiler") and a "surface boiler" configuration. No major performance differences were noted.
8. Both volume boiler and surface boiler schemes, when tested with R-113, performed well for very shallow continuous phase (water) depths.
9. A remote controlled, highly instrumented, mobile test loop has been constructed and tested using both R-113 and pentane as working fluids. Operation is very stable and requires a minimum of operator attention.
10. Essentially full scale boiler tests have been successfully performed with both the volume and surface schemes on a Raft River geothermal well using R-113 and pentane.
11. The shallow tray approach to the boiler design is preferred to the more conventional spray column because of problems with the latter in extrapolating performance from small to large diameters, as well as the flexibility of the former in handling fluids whose liquids are either heavier or lighter than water.
12. Extensive non-dimensional heat transfer predictions have been developed which correlate performance of our boiler as well as the limited data from other laboratories.
13. Comparisons of minimum mass flow ratios demonstrated in the experimental program compare very favorably with those set from conservation of energy considerations.
14. Overall heat transfer coefficients for boiling as high as $UA=4200$ Btu/hrF have been demonstrated for large scale operation. If larger capacity pumps had been used to yield 20 gpm, the equivalent UA would be 12,500 Btu/hrF.
15. A wide variety of direct contact devices appropriate for use as liquid-liquid heaters have been evaluated. As expected counterflow devices promise more efficient operation compared to parallel flow devices. Of the counterflow devices, the mechanically agitated direct contactors show great promise.
16. Potential benefits exist for the use of direct contact condensers in geothermal power cycles. Experimental work must be performed to evaluate the more promising concepts for this application.

17. Determination of the equilibrium solubility for candidate working fluids in typical brines at various temperatures enables the prediction of an upper limit on the working fluid loss expected from a direct contact cycle.
18. Equilibrium solubility measurements for Freon-113, Freon-114, pentane and isobutane have been made in this laboratory for NaCl solutions of varying concentrations and temperatures up to (approximately) 100C. The results were found to correlate quite closely with Henry's Law.
19. Pentane appears to offer the lowest potential fluid loss, based upon equilibrium solubility considerations and cycle performance factors.
20. Removal of non-condensable gases from the cycle can cause severe working fluid loss unless special techniques are used to minimize this.
21. Sparging of the spent brine with a gas may be an effective means of stripping the dissolved working fluid from the brine.
22. A comprehensive cycle analysis computer program has been developed that allows the evaluation of various geothermal power plant configurations. Currently, this has been used to determine performance of direct contact heat exchanger cycles.
23. Thermodynamically optimal direct contact cycles have been chosen for a variety of working fluids and wellhead conditions.
24. Direct contact cycles demonstrate lower capital costs and, at most, roughly equivalent operating costs compared to conventional shell and tube binary cycles. If working fluid loss is less than the equilibrium solubility predictions overall power costs would be less for the direct contact cycle.
25. When pentane and isobutane plant costs are compared for a 300F resource the pentane appears to offer more economical power.
26. Based on our work, the direct contact cycle offers no technical problems to implementation. Definition of costs need additional study.

CHAPTER I

INTRODUCTION

I.A. Background

It has been estimated that the geothermal resources in the Western United States could have the capability of producing up to 385,000 MWe by the year 2000 if adequate funds were expended on resource development. However, this estimate is based on development of not only vapor and liquid dominated resources but dry geothermal resources as well. The technology for the vapor dominated resources is well understood and has been demonstrated successfully at the Geysers. Liquid dominated resources are reasonably well understood; however, the corrosive and fouling nature of the brines (up to 300,000 ppm dissolved solids) is such as to make normal heat exchangers inapplicable in many instances. Thus, new heat exchanger technology needs to be developed for dealing with much of the liquid dominated resource. Hot dry rock requires development of new drilling technology as well as energy extractive technology.

Of the known geothermal resources in the United States, most are liquid dominated. It has been estimated that this resource is at least twenty times more prevalent than vapor dominated systems. Thus, a significant part of the U.S.E.R.D.A.'s Geothermal program has been aimed at developing liquid dominated systems.

The utilization of these liquid dominated resources is being investigated using several approaches. A good recent summary of many aspects of this technology is given by Milora and Tester (1976). Typically, however, three schemes or combinations are considered for this application. Perhaps the most considered concept is the flash system. Here the hot geothermal water undergoes a series of throttle processes and portions of the liquid

vaporize and travel through a turbine. It can be shown that the efficiency of the conversion process can be affected by increasing the number of flashes, with the attendant increase in cycle complexity and equipment cost.

A second scheme called the "Total Flow Concept" (TFC) has been proposed by workers at the Lawrence Livermore Laboratory. It is claimed by the developers that the TFC theoretically demonstrates the highest performance of any scheme. Much work remains on the development of equipment, particularly a turbine capable of expanding a high pressure brine.

The third major type of utilization scheme is the binary cycle. In this concept a secondary working fluid exchanges heat with the geothermal liquid and the former vaporizes. The secondary fluid then flows through a typical Rankine cycle where useful work is extracted. Clearly if the geothermal source contains high concentrations of dissolved solids, conventional heat exchangers will be rendered ineffective because of fouling problems.

Although for geothermal sources above 200°C there is some question as to the best utilization techniques, for geothermal resources below 200°C binary cycles become the most practical means of extracting the energy. The choice of the exact working fluid would change depending upon the characteristics of the resource. However, it would normally be one of the light hydrocarbons or a halogenated refrigerant. Most of these fluids have desirable thermodynamic properties but poor thermal transport properties. Thus, the use of conventional heat exchangers would require large surface areas, the cost of which has been estimated by Milora and Tester (1976) to be second only to well drilling as a capital expenditure. Further difficulties are encountered due to the chemical nature of the brine. Although no specific statement can be made as to the dissolved solids' loading or its chemistry, the general trend is the higher the source temperature, the greater its loading.

The dissolved solids are normally chlorine salts and silicon oxides with a preponderance being NaCl. The dissolved gases are normally H_2S and CO_2 . Wells drilled a short distance apart can, as in the case of the Imperial Valley, vary from a few thousand parts up to more than 200 thousand parts per million.

The corrosive and scaling characteristics of geothermal brines led workers of this laboratory (Boehm, Jacobs and Coates, 1974) to suggest direct contact heat exchange as a means of reducing capital costs and maintenance for binary geothermal cycles, particularly for the higher salinity brines. The concept involved the introduction of the working fluid in direct contact with the geothermal brine. While direct contact processes were not new, their application to power generation cycles appeared to be quite unique. Areas where pertinent work had already been accomplished and reported in the literature were catalogued. The overall concept was then analyzed to pinpoint areas where additional work was required.

Proposals for project support were submitted to NSF-RANN and AEC (through the Idaho National Engineering Laboratory). The NSF proposal was oriented toward developing the basic science and engineering necessary to evaluate virtually all aspects of the concept for comparisons to other concepts. On the other hand the AEC work was oriented in its initial stages to boiler hardware development. The rationale for this was that the bulk of the heat duty in a given cycle is carried out in the boiler, and relatively less information was available in the literature on direct contact boilers compared to liquid/liquid devices. An outline of the tasks is given in Table I.1, where the two programs as formulated in the early stages are delineated. Also shown are extensions to the (originally) AEC contract for

TABLE I.1
SUMMARY OF TASKS PERFORMED

I. AEC-Raft River Work (Now ERDA E(10-1)1523)

A. Systems

1. Approximate Cycle Analysis--Fluid Investigation
2. Design, Construction, Installation and Testing of Initial Test Loop
3. Design, Construction, Installation and Testing of High Pressure Test Loop
4. Preliminary Optimization Analysis of Cycles

B. Heat Transfer

1. Laboratory Tests; Design, Construction and Testing
 - a. Volume and Surface Boiler Schemes
 - b. R-113, pentane
2. Raft River Tests; Design, Construction and Testing
 - a. Volume and Surface Boiler Schemes
 - b. Study of Various Parameters
 - c. R-113
3. High Pressure Test; Design, Construction and Testing
 - a. Volume and Surface Boiler Schemes
 - b. Study of Various Parameters
 - c. Raft River and Imperial Valley*
 - d. R-113 and Pentane*
4. Direct Contact Condensers
 - a. Outline of Existing Problems
 - b. Design, Construction and Testing of Appropriate Schemes*
 - i. Spray Columns
 - ii. Packed Beds

C. Solubility

1. Review of Literature
2. Evaluation of Stripping Techniques*

II. NSF-RANN--Basic Development (Now ERDA E(10-1)1549)

A. Systems

1. Cycle Computer Code Development
2. Fluid Evaluations
3. Detailed Analysis of Candidates
4. Fluid Loss Evaluations
5. Costing Estimates
6. Operational System Design*

B. Heat Transfer

1. Boilers
 - a. Model Analysis
 - b. Testing
2. Preheaters
 - a. Parallel Flow Concepts
 - b. Counter Flow Concepts

C. Solubility Determination

1. Hydrocarbons and Fluorocarbons in Water
2. Hydrocarbons and Fluorocarbons in Salt Water

*Denotes task not yet completed.

additional field testing and for other aspects related to plant evaluation.

The overall philosophy of the program has been to study the effects on plant design of the many parameters for a broad range of well and operating conditions. In this regard, as many options as possible were left open. For example, although R-113 was used extensively in the early field tests, all analyses and hardware were developed to handle a variety of fluids. It was anticipated, and will be demonstrated in this report, that certain fluids may be preferred for certain reasons and conditions, while other fluids may be preferred for others.

It was also clear that a broad range of tasks (see above) had to be accomplished. Many tools had to be developed.

The work of Milora and Tester (1976) which appeared during the performance of the present work, is the only report currently available that touches as broad a spectrum as the present work. Since the direct contact cycle is in reality a combined binary-flash system, the binary work of Milora and Tester can be compared to some aspects of the present work, but only for the case where the working fluid saturation pressure is very high compared to that of water, and then only approximately. This will be demonstrated later.

I.B. Comparisons to Other Programs

Shortly after the present work was initiated, the AEC issued an RFP to begin direct contact programs at other laboratories. Three similar projects were then initiated at the Ben Holt Company, DSS Engineers and Occidental Research.

The Ben Holt work has been completed and a final report has been issued (Sims, 1976). This work involved studies of the heat transfer between hexane

and water in a small spray column at low pressure. Some system studies and solubility measurements were also performed.

DSS Engineers has been performing studies on a six-inch diameter spray tower using subcritical isobutane with water and artificial brine. These early studies were performed at Wrightsville Beach, North Carolina. A preliminary plant costing study was performed. Results are to be found in a major report which has been issued (Suratt and Hart, 1977). Recently their loop has been moved to the East Mesa facility in the Imperial Valley of California where they have been testing with actual geothermal brine.

Occidental Research has been pursuing supercritical testing of a counterflow tower with isobutane and geothermal brine. Results have been recently described in an unprinted paper presented at the 1977 Geothermal Resources Council Meeting in San Diego.

It should be noted that all of these programs have a very specific set of objectives. Typically a configuration using somewhat conventional design (see Section III.C.3, below) is being or has been evaluated.

Two other projects should also be noted because they relate to some aspects of the present work. One of these is the development of the program GEOTHM at Lawrence Berkeley Laboratory. This program is a versatile optimizing routine for analyzing power cycles (Green, et al., 1977). While it is extremely powerful, it has not yet been adapted to direct contact cycles.

The work of Ken Starling's group at the University of Oklahoma also is of pertinence to the present work. (See, for example, Starling, et al., 1976). Their work has dealt with several computer studies. This includes the development of thermodynamic models, the study of fluid mixtures, and the study of cost factors of power cycles. Preliminary work related to direct contact cycles has been performed recently.

I.C. Report Philosophy

The report that follows is a summary of the bulk of the work that has transpired since the beginning of both projects. There are a variety of situations described ranging from essentially technology reviews to final summaries of extensive projects.

Several other ERDA reports have been issued from this project, and many technical papers have resulted from this work. For the most part these papers and reports are treated implicitly as unattached appendices to the present summary. Where pertinent, these reports and papers are referenced within the present text, and the interested reader is directed to them for further information.

In the cases where no prior release of the information has been made in a separate report* or paper, essentially complete detail is included here.

*It should be noted that virtually all aspects have been mentioned to some degree in the regularly issued quarterly reports from the Raft River Geothermal Project.

CHAPTER II

SYSTEM ANALYSIS

II.A. Introduction

A major concern of the University of Utah geothermal program has been to develop the tools necessary to analyze possible direct contact binary cycles for optimization purposes and for comparison with alternative schemes such as the ordinary binary cycle and the flash processes in generating electrical energy from the liquid dominated geothermal resource. In order to do this it was necessary to acquire the component tools such as thermodynamic properties and solubility data for the potential working fluids as well as the brine. It was also necessary to obtain design information on system components and develop a rationale for analysis. These were primary goals of the ERDA (NSF-RANN) grant, AER-75-01034. The application of the system analysis to the design of a direct contact power plant using moderate temperature brines has been a goal of the continuation of contract AT(10-1) 1523 from ERDA, Idaho Falls.

In this section the rationale developed for the analysis of direct contact geothermal cycles will be discussed together with some of the results acquired. Details of the work on system components are covered in Chapter III of this report.

In order to discuss the rationale needed in designing a direct contact cycle let us first review a basic vapor cycle. Such a cycle is shown in Figure II.1 with the corresponding temperature-entropy diagram in Figure II.2. Due to the nature of the fluid and operating constraints for the turbine a certain degree of superheat may be necessary. Thus the work produced (the area 1-2-3-4) is considerably less than for a Carnot cycle operating between the same temperature limits. The choice of the working

Figure II.1. Simple Rankine cycle

Figure II.2. Temperature-entropy diagram for an ideal Rankine cycle

fluid can greatly influence the efficiency of the cycle as can be seen from Figure II.3. If the saturated vapor curve has a negative slope as in curve B (typical of water), the limitations on moisture content in the turbine require superheating the vapor. For the same maximum temperature such a fluid rejects more heat and thus lowers the efficiency as well as increasing the heat load of the condenser. For a fluid with a positive slope to its vapor curve as with curve C (typical of isobutane) the efficiency is again reduced since the vapor exits the turbine superheated. Superheat results in a slightly greater heat load for the condenser than if it exited at saturation. This greatly increases the condenser size. This increase in size is due to the fact that the heat transfer coefficient for a gas or superheated vapor is considerably less than when that vapor is saturated and condensing. Thus, ideally the selection of a fluid such as "A" whose saturated vapor curve is a line of constant entropy would minimize equipment size and maximize the efficiency, all other things being equal. All of the above discussions, however, are independent of the source of heat and problems associated with the transfer of heat from it.

Let us now consider the geothermal brine. Figure II.4 represents the brine being available from the well at temperature T_1 with a heat sink at T_2 . If the brine could all be expanded through some sort of isentropic expander to state 2 the potential work is described by the area 1-2-3-4. Such a process is referred to as the total flow concept. The problem with this is that no such device exists, although basic research is being conducted at the Lawrence Livermore Laboratory on a potential design. If the brine is used as a heat source, as in a binary cycle, some fraction of its energy is available. In terms of brine

Figure II.3. Comparison of the effect of the slope of the saturated vapor curve on Rankine cycle efficiency

Figure II.4. Brine T-S diagram

efficiency the available work may be represented by the area 1-2-3-4. This remaining available energy may be all or partially used depending upon how it is utilized both in terms of the system and the final working fluid. One such system would entail the isoenthalpic expansion of the brine to a lower pressure in order to generate steam. This steam would then be expanded through a turbine, condensed and then rejected from the system or possibly dumped directly from the turbine. Such systems and processes are referred to as flash processes. These are without a doubt the cheapest systems to construct and operate if one places no value on the geothermal brine. They are not necessarily the cheapest if the brine has a value attached to its usable energy. Of course, this must be considered, if only to recover the cost of drilling wells. The energy remaining in the liquid after flashing is lost.

A second method of utilizing the available energy is to transfer the heat to a different working fluid which could then be expanded through a turbine. Such a system is called a binary cycle. Binary cycles have been the subject of considerable interest, as applied to geothermal resource utilization. (See references by Green and Pines (1975), Walter and Wilson (1976), Lawson (1976), Miller (1974), Starling, et al. (1975), Beaulaurier (1974), Cortez, et al. (1973) and Milora and Tester (1976)) They have been used in Italy with ethyl chloride as a working fluid and in the Soviet Union with R-12 as a working fluid (Moskvicheva and Popor (1970)). Considerable interest in the design of a power plant using isobutane as the working fluid has been expressed by a variety of firms who have analyzed such systems, i.e., Ben Holt Co., Bechtel, LBL, Magmax, Sandia, Stanford Research Institute, Battelle Northwest. Various investigators have shown that supercritical cycles appear the most attractive for the binary cycle.

This is due to the shape of the available energy from the brine as shown in Figure II.4. This becomes obvious on referring to Figure II.5. The unavailable energy is due primarily to the required temperature difference in the primary heat exchanger; however, the choice of a fluid which leaves the turbine in a superheated state can also decrease the available energy. It is readily apparent that the ideal working fluid would have to be tailored to the geothermal resource for the simple binary cycle and it should be so tailored as to operate as a supercritical cycle. It should, however, be selected so that pumping work is not excessive and that capital equipment costs and maintenance costs are small. These desirable attributes may not be available, however, and a compromise must then be made.

From a strictly thermodynamic analysis viewpoint binary supercritical cycles are quite attractive as seen above. Milora and Tester (1976) have shown that if no problems are introduced by the brine then fluids such as R-115 are extremely attractive when applied to moderate temperature brines. However, when availability of the working fluid and the total economics are considered hydrocarbons become favorable. The primary difficulties introduced by refrigerants and hydrocarbons are the poor heat transfer characteristics of the fluids. When conventional heat exchangers are used Milora and Tester have shown that the heat exchangers are the major cost items of the plant and that fully one third of the cost of producing electrical power can be attributed to them. This is true without even considering the extremely detrimental effects of the brine chemistry.

In investigating possible means of reducing the cost of binary power plants we first introduced the concept of a Direct Contact Binary Cycle in 1974 (Boehm, et al., 1974). Further support for this concept followed

VAPOR DOME FOR NEGATIVE
SLOPE SATURATED VAPOR LINE

VAPOR DOME FOR POSITIVE
SLOPE SATURATED VAPOR LINE

Figure II.5. Supercritical geothermal cycles

from I. Sheinbaum, (1975) then of the Ben Holt Co. The utilization and optimization of the direct contact cycle, however, offers complications not encountered in the ordinary binary cycle although it is definitely a means of reducing the cost of the heat exchangers and a means of avoiding the problems associated with brine chemistry, i.e., fouling and corrosion of the heat exchanger surfaces. The two major problems introduced by direct contact are associated with the facts that all fluids are partially soluble and that two fluids brought in contact must be at the same pressure.

II.B. Direct Contact Cycle Design Consideration

Consider the simple direct contact cycle shown in Figure II.6. The brine from the well condition A, may be at a pressure only slightly above its saturation pressure. For a given working fluid this could be considerably below the ideal pressure for the turbine entrance conditions, 3. In that case a brine pressure boost pump is necessary to provide the brine at condition B where $P_B = P_3$. The net work is reduced by this required pump work below that possible for a standard binary cycle where the brine pressure can be considerably below that of the working fluid. It is possible, however, that the smaller approach temperatures, $T_C - T_2$, that could be achieved by a direct contact heat exchanger might compensate for this increased pump work.

Depending upon the intimacy of the mixing between the working fluid and the brine, a loss of working fluid will occur due to its solubility in the brine. Conservatively, it will be at equilibrium conditions at the temperature and pressure at which the brine leaves the system, T_C and P_C . In the simple cycle of Figure II.6, $P_C = P_3$. The degree of salinity of the brine affects the solubility. The higher the salinity the lower the solubility. In this fact it is clear that the poorer the

Figure II.6. Schematic of a basic direct contact binary power cycle

conditions are for the use of conventional heat exchangers the better they are for direct contact heat exchangers! However, it must be remembered that any loss of working fluid increases the cost of producing the electricity.

The above discussion points out that in comparing direct contact binary cycles to standard binary cycles, the operating pressure takes on increased significance in the selecting of the working fluid. The solubility of the working fluid is introduced as a new parameter which can completely eliminate from contention fluid that looked attractive for a conventional binary cycle. As an example, ammonia has desirable thermodynamic properties but is eliminated due to its high solubility in water. (Its toxicity also poses a hazard.)

Another factor unique to the direct contact cycle is that the fluid entering the turbine is always a mixture. That is, if the cycle is supercritical there will be water in the working fluid according to its solubility in the working fluid. If the cycle is subcritical the amount of moisture in the vapor will be dependent upon its partial pressure, which may be calculated in the same way that one calculates the saturated water content of the atmosphere. For example, given the temperature of the vapor leaving the primary heat exchanger, T_3 , the equation of state for water yields the pressure $P_{H_2O(3)}$ for which the specific volume is also known. If the pressure at 3 is known, $P_{WF3} = P_3 - P_{H_2O(3)}$. From the working fluid equation of state one can find V_{WF3} . The mass ratio of water to working fluid in the vapor is thus

$$\left. \frac{M_{H_2O}}{M_{WF}} \right|_3 = \frac{V_{WF3}}{V_{H_2O(3)}}$$

As an example, with pentane as the working fluid, $T_3 = 130^\circ\text{C}$ (266°F) with $P_3 = 13.5$ atmospheres, $M_{\text{H}_2\text{O}(3)}/M_{\text{Pent } 3} = .0487$. Thus, a mixture of nearly five percent water vapor would be required to expand through the turbine. As the water vapor has a much greater enthalpy than pentane, the energy density is increased (available enthalpy per unit mass of vapor mixture). It has been shown at the University of Utah that this results in a smaller turbine size and lower mass flow rates than would be possible with pentane alone (Bliss, et al., 1976). Thus, a definite advantage is shown for the direct contact binary cycle that was not previously recognized as the decrease in turbine size indicates a decrease in cost. (See section III.B for detailed discussions and comparisons for different fluids.)

In attempting to establish a criteria for fluid selection, J. Whitbeck at EG&G, Idaho, attempted, in 1975, to calculate the relative size of turbines. Such an analysis indicated that the size could roughly be proportioned to the inverse of the molecular weight raised to a power. Since most potential working fluids have higher molecular weights than water, the presence of water vapor reduces turbine size by reducing the effective molecular weight. This is discussed in detail in Section III.B and in the report by Bliss, et al. (1976) in which a modification of the theory of Balje (1962) is given. These facts have not been considered by other investigators and point to another reason why one cannot simply assume that the best choice of a working fluid with direct contact follows directly from ordinary binary geothermal cycle analysis.

II.C. Direct Contact Geothermal Cycle Analysis

Due to the many particularities that make direct contact geothermal cycles unique, and the lack of availability of a means of analysis the

development of the computer program DIRGEO was initiated (Riemer, et al., 1976). The program was designed to use available information and new data or techniques as they became available. Thus, DIRGEO carries out most of its analysis utilizing subroutines. Subroutines exist for calculating thermodynamic properties, carrying out turbine analyses, and heat exchanger design etc. (Riemer, et al., 1976a, b, c). Figure II.7 shows the main program hierarchy as it now exists. Current modifications include extension of the subroutine PREHET to assess a variety of liquid-liquid direct contact preheaters including spray columns, baffle columns, turbulent pipe heat exchangers and mechanically agitated columns. Another subroutine is to be added to design the condenser system including a variety of direct contact condensers. A third addition will allow for design of systems for working fluid recovery and separation of water from working fluid in the condenser. The latter is partially included within the present computer code by assessing the potential recovery by flashing the brine to a specified pressure prior to rejection. The hardware requirements necessary for this type of system have not been included. A total costing package also needs to be included.

Potential direct contact geothermal cycles analyzed initially assumed that the geothermal brine is available at the turbine inlet pressure. The simple calculation of the required pump work for the brine if this is not the case has now been added. Calculations now assess this penalty to the cycle design imposed by the direct contact heat exchanger concept. Thus, an additional input of brine pressure is required.

The main cycle program DIRGEO differs from binary cycle computer programs in that it must analyze a turbine with a binary mixture as a working fluid, that it must assess fluid losses and a wide range of

Figure II.7. DIRGE0 main program hierarchy

possible types of heat exchangers. Due to the latter it became necessary to develop a unique system for numbering the thermodynamic states. The numbering sequence starts with the boiler inlet brine condition as being one. The boiler exit to the turbine was numbered two. The exit state of the last turbine stage is then two plus the number of turbine stages. The condenser exit state number is the number of turbine stages plus three. The numbering sequence must next account for the number of preheater stages if any. The preheater states are numbered for the secondary fluid consecutively from the condenser to the boiler. The brine states are numbered starting from the boiler exit as the number of turbine stages plus two times the number of preheater stages plus 6 through to the exit from the final preheat stage. Secondary processes on the brine after exiting the final preheater would be numbered from there. Figure II.8 shows a typical cycle with two parallel flow direct contact preheaters.

If a counter-current type preheater were used there would normally be only one preheater with the requirement probably for a single pump. Thus, the boiler feed pump would require no work as the pump in front of the first preheater would in fact be the boiler feed pump. Some of the so called counter-current direct contact heat exchangers do not really give counter-current flow. This may be particularly true of a large diameter-to-height ratio spray column (See Section III.C). Consider the cycle shown in Figure II.9 and its temperature duty chart, Figure II.10. It is obvious that four theoretical stages are required, but a large diameter-to-height spray column may only be able to achieve one stage without internal vertical baffling. If no baffling is used two preheaters and an interstage pump might be required.

Figure II.8. Schematic diagram of a direct contact cycle indicating the state numbering system

Figure II.9. Direct contact binary geothermal cycle - moderate temperatures

Figure II.10. Temperature vs duty for the contactors of Figure II.9

The computer program is presently equipped to handle the following fluids as potential working fluids in a direct contact cycle: R-113, R-114, R-11, R-12, R-13, R-14, R-21, R-22, R-23, butane, ethane, ethylene, heptane, hexane, isobutane, isopentane, methane, octane, pentane, propane, and propylene. It is planned to add the following five cyclo-hydrocarbons: cyclo-pentadiene, cyclo-pentane, cyclo-pentene, methyl-cyclo-butane and methyl-cyclo-pentane. The latter five compounds were selected at the end of a thorough search of the literature as possible direct contact binary cycle working fluids.

1. System Components

1.a. Direct Contact Boiler

The boiler analyzed is a direct contact binary fluid boiler of arbitrary configuration. Program DIRGEO requires as input data the brine inlet and outlet temperatures for the boiler, the degree of superheat of the working fluid vapor and the vapor exit temperature. The boiler approach temperature is specified implicitly by the three inlet and outlet temperatures. For most of the cases run the boiler approach temperature was fixed at 2.8°C (5°F).

The working fluid enters the boiler very near its saturation temperature. The ratio of brine/working fluid mass flow rates is then determined from an energy balance in the boiler. The boiler pressure is taken as the sum of the saturation pressure of water at the vapor exit temperature and the saturation pressure of the working fluid at its saturation temperature.

At this time no physical size or cost estimate for the boiler is calculated in program DIRGEO.

1.b. Binary Vapor Turbine

The turbine analysis is described in detail in Section III.B and in

the report by Bliss, et al. (1976). Program DIRGEO is unique in that it calculates the expansion of a mixture of water and working fluid vapor in the turbine. The required inputs to the program are the turbine adiabatic efficiency and the gross turbine shaft power output. From this power output, the inlet conditions (conditions at the boiler exit) and the condenser pressure, the total fluid mass flow rate is calculated.

1.c. Condenser

The condenser is analyzed as a conventional tube and shell condenser. The condensing temperature is specified as program input data. It is anticipated that the program will be modified to calculate operating conditions for a direct contact condenser.

1.d. Preheaters

Either counter- or co-current flow can be analyzed by the preheater subroutines of program DIRGEO. The co-current direct contact preheater is analyzed using the theory of Johnson (1975) to calculate the size exchangers required for up to ten preheater stages.

The counter current heat exchanger subroutine performs an energy balance on the two streams. The minimum approach temperature is always equal to or slightly greater than the boiler approach temperature.

2. Thermodynamic Properties

In order to determine the "best" working fluid for use in a geothermal binary cycle it is necessary to have reliable thermodynamic properties for the potential fluids as well as the brine. In June 1975 under grant number AER-75-01039 a major effort was directed toward the development of thermodynamic codes for use in the direct contact cycle analyses program, DIRGEO. In order to minimize duplication of work it was necessary to develop three separate subroutines. These subroutines are called WATER,

CARBON and FREON. These subroutines are described in detail in three University of Utah reports (Riemer, et al., 1976a,b,c). These routines use the basic equations of state to compute the fluid properties thus allowing greater versatility within a fixed computer core storage than would be allowed with tabular look-up schemes at near the same computation time.

2.a. Subroutine Water

This subroutine computes the thermodynamic properties of water from the formulas of Keenan, Keyes, Hill and Moore as published in the 1969 steam tables. Computed values are thus accurate to the same degree as the tables. The basic equations of state were written as functions of the free energy given density and temperature. Using Maxwell's relations the basic mathematical operations were performed allowing for the rapid calculation of all thermodynamic processes given any of the following pairs:

- 1) Temperature - Pressure
- 2) Temperature - Quality
- 3) Pressure - Entropy
- 4) Pressure - Enthalpy
- 5) Pressure - Quality

This allows for the analysis of all processes in a typical vapor cycle to be analyzed rapidly.

2.b. Subroutine Carbon

This subroutine was developed from the Benedict-Webb-Rubin equation of state as modified by Starling (1970). This program was developed to handle pure fluids and as such is much faster than Starling's mixture computer program. It does use all of Starling's constants for the various

compounds. Mathematical operations were carried out prior to programming to allow for rapid calculation of all of the processes for a vapor cycle mentioned above. Currently the computer code includes the following fluids: n-butane, ethane, ethylene, heptane, hexane, isobutane, isopentane, methane, octane, n-pentane, propane, and propylene. Other fluids may be added as the constants for the BWRS equation of state are determined.

2.c. Subroutine Freon

Currently the constants are not available for the BWRS equation of state for the fluorocarbon refrigerants. The existing tables as published by DuPont were calculated by Downing (1971) using the Martin-Hou equation of state. Using this equation of state we have developed a computer program which will rapidly calculate the thermodynamic properties of R-113, R-114, R-11, R-12, R-13, R-14, R-21, R-22 and R-23 given any of the following pairs:

- 1) Temperature - Pressure
- 2) Temperature - Quality
- 3) Temperature - Specific Volume
- 4) Pressure - Quality
- 5) Pressure - Entropy
- 6) Pressure - Enthalpy

As the M-H equation of state only predicts properties in the vapor region it was necessary to use the Clapeyron equation to compute saturated liquid enthalpy and entropy. Saturated liquid specific volume is computed as a function of temperature. Compressed liquid states are approximated by saturated liquid states at the same temperature.

If constants become available for the BWRS equation of state for the fluorocarbon refrigerants, subroutine CARBON will be modified to include

them and subroutine FREON will be discarded. However, at this time, subroutine FREON is to our knowledge the most accurate means available.

2.d. Potential Fluids

The above subroutines have been well checked out using pentane, isobutane, propane, hexane, R-113, R-114 and R-11 as possible working fluids. Preliminary studies indicated that the fluids with lower saturation temperatures at atmospheric pressures (pentane, hexane, R-113) held more promise for direct contact cycles due to decreased pump work as well as high thermodynamic efficiency. During the summer of 1976 it was thus desired to carry out a library search for potential fluids not currently in the list of CARBON or FREON. In carrying out the literature search restrictions were placed that the compounds must boil at temperatures between 25°C and 70°C at one atmosphere pressure, that they be classified as insoluble in water, that the molecular weight be under 100, that they be chemically stable at temperature up to 250°C and that they be of low toxicity. The compounds were also restricted to having no hydroxyl groups attached and being otherwise chemically non-reactive with the brine.

Five compounds were found which generally fit the requirement. They are, as stated on page 27, cyclo-pentane, cyclo-methyl-butane, cyclo-methyl-pentane, cyclo-pentadiene and cyclo-pentene. These compounds have the following boiling points at one atmospheric pressure, 49°C, 36.3°C, 71.8°C, 40°C and 44.24°C respectively. They have molecular weights of 70, 70, 84, 66, and 68. Their specific gravities are .746, .684, .749, .802 and .772. Most of these compounds are classified as specialty products. (Cyclo-pentane became available commercially for the first time in 1968.) Limited thermodynamic data is available at this time for these fluids. Solubility in water is generally not reported, but rather they are stated as being insoluble. It is expected that solubilities of under 100 ppm are

probable. Thus, these fluids deserve further consideration. This is particularly true of cyclo-pentane, cyclo-methyl-butane and cyclo-pentene which are found naturally occurring in U.S.A. petroleum. A market for the products seems to be a major factor in cost at this time.

2.e. Transport Properties

At this time the transport properties are only used in sizing heat exchangers. Rapid means of determining viscosity and thermal conductivity is the major requirement necessary in terms of properties yet to be included in DIRGEO. At this time simple correlations versus temperature are input only after thermodynamic analyses indicate attractiveness for a cycle. Solubility data is also handled singularly as discussed in Section IV.A.

3. Cycle Performance

Comparisons of the performance of three working fluids in a direct contact power cycle have been made for a variety of cycle conditions. Pentane, isobutane and R-113 are the working fluid candidates that have been examined most closely, though the program (DIRGEO) is capable of examining many other fluids. The Thermodynamic Figure of Merit (TFM) has been taken as the primary measure of the cycles' thermodynamic efficiency and used as the basis for thermodynamic optimization. (The optimum TFM also implies the optimum value of brine flow/kw-hr of net power.) Though thermodynamic optimization will not necessarily imply cost optimization it is still one essential feature required for comparing relative merits of different working fluids.

In calculating the results to be presented in this section the following assumptions were made:

Boiler approach temperature = 2.8°C (5°F)

Counter flow preheater

Turbine adiabatic efficiency = 86%

Pump efficiency = 80%

Single turbine stage

Net power output 50MWe

Brine specific heat $C_p = 1.0 \text{ Cal/gm-C}$ (1 BTU/lbm-F)

0° superheat in boiler

Figure II.11 shows how the TFM varies with boiler pressure for the three working fluids. These calculations were performed for a 138°C (280°F) geothermal source temperature, a 27°C (80°F) condenser temperature and a 100 psia well pressure. The TFM reaches a maximum for the isobutane cycle at a pressure of 225 psia. At higher pressures there is insufficient temperature drop in the brine and at lower pressures it is necessary to pump more isobutane and there is a smaller pressure drop across the turbine. The TFM for the pentane and R-113 cycles increases steadily with decreasing boiler pressure until the pressure reaches the saturation pressure of the incoming brine.

It can be seen that the same TFM can be attained using either pentane or isobutane in a binary cycle using 138°C brine. Thus selection of the best working fluid for this low temperature cycle must be made on the basis of other considerations. The pentane cycle operates at a lower pressure which results in a lower capital investment in the heat exchanger vessels. But a pentane condenser will be at sub-atmospheric pressures when the condenser temperature is below approximately 38°C . This poses the potential hazard of oxygen leaking into the system. Thus the additional cost of high pressure isobutane vessels must be weighed against the cost of safeguards

Figure II.11. Thermodynamic figure of merit for three working fluids and a geothermal source temperature of 280 F

such as an oxygen detector in the pentane condenser. Of course, explosive danger is maximized when the fuel to oxidizer ratio is low; thus, it may well be safer to operate a low pressure condenser system with pentane rather than a high pressure butane system with its potential danger by leaks into the atmosphere. The R-113 cycle offers good thermodynamic performance at low source temperatures and the advantage of operating with a non-flammable fluid. As will be shown in a later section of this report, the cost of R-113 lost in solution in the spent brine and the large turbine diameter required may be excessive.

At higher geothermal source temperatures the isobutane shows better performance than either pentane or R-113. Figure II.12 shows the best TFM attainable in a sub-critical binary cycle as a function of the source temperature. As mentioned earlier, at low temperatures the three fluids have the same TFM. But at higher temperatures the isobutane cycle is superior. This is primarily because of the lower vapor pressures of the pentane and R-113. The boiler vapor exit temperature must remain higher than otherwise desirable to prevent flashing of the brine as it enters the boiler. This reduces the utilization of the brine. It is clear from this figure that fluid selection can be very dependent on the conditions of each individual well. At source temperatures above approximately 150°C isobutane may be the best fluid for a subcritical cycle. But at temperatures below 150°C the choice must be made with many considerations other than TFM in mind.

The effect of condenser temperature on cycle efficiency is shown in Figure II.13. As expected the TFM is strongly dependent upon the condensing temperature.

Figure II.12. The optimum thermodynamic figure of merit attainable with three different working fluids

Figure II.13. The effect of condenser temperature on the optimum thermodynamic figure of merit

CHAPTER III

COMPONENT DESIGN, TESTING AND PERFORMANCE

III.A. Boilers

1. Bench Tests

1.a. Introduction

At the beginning of the present program, it was obvious that a study and optimization of the performance of a boiler was a crucial ingredient to the overall success of the concept. The boiler was singled out for particular emphasis for several reasons. First of all, a search through the literature indicated that of all studies of the direct contact devices reported, relatively little was known about direct contact boiling and relatively more was known about preheaters. A second reason is that usually more heat transfer is required from the boiling mode of a power cycle than from the preheating mode. It was also considered that some preheating might also be accomplished in an optimized boiler, thus satisfying the bulk of the heat addition process.

Initial definition of program directions came from discussions with personnel associated with the Raft River Project. They desired to have a working model in operation on a well at a relatively early date.

From our standpoint it appeared that two concepts had merit. One was the spray-column configuration previously given some study in Israel (See, for example, Sideman and Gat, 1966). This technique involves a counterflow device which is extremely desirable when only sensible heat is transferred. For application where change of phase heat transfer occurs there is no advantage to the counterflow arrangement. A second configuration had been suggested in some water desalting work in Turkey (Somer et al., 1973) where one fluid had been sprayed upon a layer of

another fluid. This design appeared to have desirable characteristics for the introduction of the brine (no complex injection manifold) as well as minimal problems associated with increasing heat exchanger size from smaller designs.

Based upon the aforementioned considerations, work started on boiler development in the spring of 1975. The basic plan was to initiate design, construction and testing of two bench-scale boiler models based upon the two schemes noted above.

A key consideration in the testing program was the choice of a working fluid. Where all factors were evaluated concerning the choice of a fluid, only two general categories were considered to be known in sufficient detail to be definitive in choosing between a large number of fluids. These were the vapor-pressure characteristics and the safety considerations. When these two were taken together, the selection of R-113 was made. In addition to being non-flammable and non-toxic, it was also considered to boil at an appropriate (neither too high nor too low) pressure at the Raft River geothermal source temperature.

1.b. Volume-Type Boiler

Initially it was hoped that boiling experiments on the volume-type boiler (spray column device) might reasonably be performed in a small, high pressure vessel. Even though R-113 was chosen as the initial fluid for testing, it was planned that other, perhaps higher pressure, fluids might also be tested. Hence, the vessel as shown in Figure III.1 was constructed. While the vessel design had flexibility incorporated, it was found after initial testing that sufficient degrees of control and monitoring did not exist. Because of this an alternative design was used.

Figure III.1. Assembly drawing of the original volume boiler vessel

The new design was patterned after the scheme used by Sideman and Gat and is shown in Figure III.2. While the new scheme was restricted to operation near atmospheric pressure, it did allow excellent visual observation as well as many control features when incorporated in an appropriate test loop. A schematic of the test loop is shown in Figure III.3. For details of the experimental apparatus and procedure, the reader is directed to the complete report of this work (Blair, et al., 1976). Only a summary of key results is presented here.

Essentially all of the controllable variables were examined including:

1. Mass flow ratio
2. Magnitude of each flow rate
3. R-113 injection velocity
4. Column height (water level)
5. Injection matrix (jet spacing)
6. R-113 subcooling
7. Water inlet temperature

Not all performance characteristics will be analyzed here. Only those pertaining to later testing considerations will be discussed.

A typical set of experimental data is shown in Figure III.4. Results are shown in terms of a UA product, which is equivalent to the total heat transfer divided by the log mean temperature difference. Note that as the mass flow ratio decreases, the UA product increases. While it is desirable to operate at a very low mass flow to maximize the heat transfer, in fact a practical limitation exists in that there exists a point where not enough energy is present in the brine to completely vaporize the secondary fluid. This may be either a localized phenomena (primarily to incorrect design or operation) or a result of simply exhausting all the possible temperature

Figure III.2. Assembly drawing of the second volume boiler test vessel

Figure III.3. Schematic of the volume boiler apparatus

Figure III.4. UA vs. mass flow ratio at a water inlet temperature of 60.0°C

driving force. In column operation with simple hydrocarbons, which in liquid form are less dense than fluorocarbons, a typical concern is to operate at high through-puts without inserting so much secondary fluid that the flow of water is choked off ("flooding"). This phenomenon was not witnessed in the present experiments with R-113. Hence, a major concern, along with general performance considerations, became the definition of the point where incomplete boiling became apparent (defined here as "carry-under").

Perhaps the most important variable studied was the column height. It was this factor that had the largest effect on later designs. When column height was increased, it was found that the primary result was that the amount of superheat increased, but that the overall heat transfer performance varied little. It was noted that the most efficient design from a container standpoint was a very shallow level of water. Hence, further operation on both small scale and large scale units was generally with water levels of 1 1/2" or less.

Blair made an early attempt to define a criteria for mapping points where carryunder might occur. The results given for this apparatus are shown in Figure III.5.

1.c. Surface-Type Boilers

Deeds (See Deeds, Jacobs and Boehm, 1976) pursued the surface boiling configuration where working fluid is sprayed on a layer of water. His apparatus used for R-113 tests is shown in Figure III.6. Deeds used a tray configuration to control residence time in addition to all of the other parameters of the experiment.

Heat transfer results very similar to those found by Blair were demonstrated. A very large increase in the UA product was seen as the

MASS FLOW RATIO R-II3 TO WATER

Figure III.5. Desirable operating conditions

Figure III.6. Schematic of small-scale surface boiler system

R-113 flow rate was increased at constant water flow.

Deeds pursued several aspects not explored by Blair. He investigated the effect of dissolved solids on the minimum temperature difference required for initiating evaporation. He found negligible difference in this minimum temperature (4 to 6°C) required with tap water when compared to water containing 6% NaCl. This was considerably below the tap water result found by Blair.

A result of the surface-boiler work was the development of a dimensionless correlation for presenting the heat transfer results. This is presented graphically in Figure III.7. The resulting relationship is

$$St = 1.91 H^{0.625} (\dot{m}_{H_2O}/\dot{m}_{R-113})^{0.142} \quad (III.1)$$

where

$$St = \frac{UA}{\dot{m}_{R-113} C_{pR-113}}$$

and

$$H = Pr_{V_{R-113}} Ja_{R-113} = \frac{Pr_v h_{fg}}{C_{pV_{R-113}} (T_{H_2O_{mean}} - T_{sat_{R-113}})}$$

This correlation resulted from theoretical considerations of a single droplet of R-113 boiling in bulk water. In the single droplet situation the important parameters were the drop diameter, velocity, relative viscosities of the two fluids, the difference between the continuous phase temperature and the saturation temperature of the dispersed phase, and the thermodynamic properties of the dispersed phase. Based on the simple model of a drop evaporating in a continuous immiscible fluid, it was shown that the non-dimensional numbers of prime importance in determining the Nusselt number were the Reynolds number based on the liquid properties

Figure III.7. Nondimensional heat transfer for a surface type boiler

of the dispersed phase and the Prandtl and Jakob numbers based upon the vapor properties of the dispersed phase. For the multiple drop problem in the spray situation there is no obvious dimension to characterize the flow, however the ratio of initial drop diameter to drop spacing is obviously important since it affects the ability of the drops to acquire heat from the continuous phase.

Although this approach yielded good correlation in the surface configuration, it did not explain thoroughly the volume boiling phenomena. Additional work on correlations continued with the large-scale tests (see the next sections).

Deeds also investigated the conditions that defined initiation of carry-under operation. He presented a relatively straight forward result and this is shown in Figure III.8. Again, somewhat different results between the volume and surface boilers were demonstrated.

1.d. Further Work Plans

When the two configurations were compared, it was clear that both had been successfully demonstrated. However, there were distinct differences in several aspects in their performance. It was then decided to proceed with a set of large-scale tests such that the two configurations could be tested under nearly identical situations. This series of tests is described in the next section.

Figure III.8. Acceptable operating range for R-113 water system

2. Large Scale Boiler Tests

2.a. Boiler Design

Two large scale boiler test facilities have been constructed. The first, for use with R-113, used a boiler pressure vessel which was uncoded though rated to operate at high pressure. The non-flammable nature of R-113 and its low vapor pressure assured safe operation. When the system was altered to use pentane a new boiler vessel was constructed to ASME Code specifications and provisions were added for remote operation and sensing of the boiler system. In the interest of brevity only the pentane boiler facility will be described in this report. The R-113 boiler loop is identical in all significant features excepting the use of manually operated control valves and simplified fluid level controls in the various vessels. It should also be noted that fluids that in liquid forms are either heavier or lighter than water can be used with minor modification of the loop.

A flow diagram of the pentane system is shown in Figure III.9 and the piping schematic is shown in Figure III.10. It will be noticed that the basic process is inherently simple. However due to the experimental and intermittent nature of the work, and the potential danger of using a flammable liquid, it was found necessary to include instruments and controls and more stringent safety precautions than would otherwise have been considered adequate. The equipment comprising the pentane loop is housed in two separate trailers spaced fifty feet apart due to the potentially explosive nature of the pentane. The control system, instruments and brine booster pump are located in trailer A, and the rest of the hardware in trailer B. All electrical equipment on or around this trailer is explosion proof. The main components of the loop are as follows: a

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SCALE: 1/8" = 1'	APPROVED BY:	DRAWN BY: J. L. G.
DATE: 12-21-70		REVISED:
DIRECT CONTACT HEAT EXCHANGER PENTANE LOOP		
FLOW DIAGRAM		DRAWING NUMBER C1

Figure III.9. Direct contact heat exchanger test loop schematic

Figure III.10. Piping and instrument diagram for the large-scale test loop

boiler, a brine booster pump, a pentane booster pump, an air cooled condenser, a gravity type separator, a liquid pentane transfer pump, a transfer vessel, a preheater and a pentane vapor detection system.

The major element of the hardware is the boiler (Figure III.11) in which liquid pentane is sprayed directly onto the hot geothermal brine. The brine is contained in a circular tray (Figure III.12)* and enters by means of a central orifice fixed to the bottom of the tray. The brine flows radially outward from the center of the tray and exits over four equally spaced weirs. The pentane is sprayed onto the brine through a centrally suspended hollow cone nozzle. Both the brine and the pentane are brought to the tray by means of flexible hoses connected to fittings welded through the vessel wall and attached to external feed lines. The boiler has four main lines connecting it to the rest of the loop, these are as follows: The brine feed line, the pentane feed line, the spent brine dump line and the pentane vapor/steam exit line. Brine from the well passes through the brine booster pump whose output pressure is regulated by means of a variable pressure relief valve R1. The flow into the brine feed line is regulated by means of a manual control valve B5. Both valves are situated in trailer A. Pentane from the transfer vessel passes through the pentane booster pump P3 whose output pressure is regulated by means of a pneumatically operated control valve N14 remotely actuated from trailer A. The spent brine collects in the bottom of the boiler vessel after it passes over the weirs and is then dumped through a solenoid valve situated on the bottom of the vessel. From there it passes through the spent brine dump line out into the

*Alternate tray configurations are shown in (Figures III.13 and III.14)

Figure III.11. The high pressure large scale direct contact boiler

Figure III.12. The circular "source flow" surface boiler tray

Figure III.13. Rectangular tray used in the residence time studies

NOTES:

1. INLETS ARE NOT SHOWN IN PLAN FOR CLARITY.
2. ALL MATERIAL $\frac{1}{8}$ " PLATE UNLESS NOTED.

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SCALE: NONE	APPROVED BY:	DRAWN BY: J. L. L.
DATE: 12-29-76		REVISED
DIRECT CONTACT HEAT EXCHANGER PENTANE LOOP		
TRAY 'B'		DRAWING NUMBER C6

Figure III.14. Alternative circular tray configuration

holding pond. The solenoid valve is operated by means of a sonic sensor S1 situated on an external sight gage which detects the level of brine in the bottom of the boiler and trips a relay when that level passes the sensor. Upon activation, the solenoid valve is held open for a selected length of time (about 5 seconds) by means of a variable timer. The pentane vapor and steam pass through a mist eliminator inside the boiler and exit through a fitting on the top of the boiler, passing through a pneumatically operated control valve N1 remotely actuated from trailer A, before being condensed in the air cooled condenser. Boiler pressure may be read either from a gage on the boiler or from a remote gage pneumatically linked to trailer A. The liquid pentane and steam condensate is gravity fed into the separator where, due to its greater density, the water collects in the bottom of the vessel. The water is then removed by an arrangement similar to the one used to control the level of brine in the boiler. A sonic sensor S2 situated inside the separator activates a relay whenever the water/pentane interface reaches the sensor. This in turn operates a solenoid valve which is kept open for a selected length of time (about 2 seconds) by means of a variable timer. The condensate is then collected in a drum for measurement. The liquid pentane is pumped intermittently from the separator into the transfer vessel. This vessel serves two purposes: First it provides the necessary suction head for the pentane booster pump and second it provides a means of calibrating the flowmeter F3. Three sonic sensors S3, S4, S5, on this vessel operate as follows: with the transfer pump on, the level of the pentane in the vessel rises until it reaches the top sensor, S3. The sensor detects the level and switches off the pump. The level then falls as pentane is constantly being removed by the booster pump and when it

reaches the middle sensor S4 an electric stop clock is switched on. When the level falls to the bottom sensor S5 the clock is switched off and the transfer pump switched on. The volume of the vessel between the middle and bottom sensors being known by previous calibration, it is then possible to calculate the rate of flow of pentane entering the boiler and a direct check can be made on the accuracy of the flowmeter F3. It is necessary to control the function of the clock start up and transfer pump shutdown with two different sensors. This is in order to allow any turbulence in the vessel due to transfer pumping to die down, as this could affect the repeatability of the sonic sensor and thereby the accuracy of the calibration run. The preheater* is a two pass tube and shell heat exchanger and may be used to heat the liquid pentane to a required temperature before it enters the boiler. It uses brine bled off before the brine booster pump, the temperature of the pentane being regulated by varying the flow of brine to the heat exchanger using manual control valve B2 situated in trailer A. The pentane vapor detection system is a General Monitor's Model 180 combustible gas monitor, calibrated to detect pentane. It has two main parts, the control and warning device located in trailer A and the sensor located in trailer B. This system monitors the concentration of pentane vapor around the equipment on trailer B and gives warning before an explosive concentration is reached. It also serves as a way of detecting what has been termed "carryover" from the boiler tray. Carryover is the name given to the state in which liquid pentane exits over the weir along with the spent brine inside the boiler. A sampling tube situated in the weir introduces brine to the gas monitor's sensor approximately every

*A direct contact preheater will also be used in later configurations.

20 seconds for analysis and triggers an alarm if pentane is present.

This system allows the operator to know the conditions at which the tray is no longer being utilized to boil off pentane and prevent liquid pentane from being allowed to exit from the boiler with the spent brine or from being boiled off in the bottom of the vessel.

Thermocouples and pressure gages are located as required throughout the system, the temperatures being printed out intermittently on a chart recorder. The boiler system is protected against overpressurization by means of two pressure relief valves, set below the maximum working pressure of the boiler. The relief valve R2 bypasses the control valve N1 in the event of its failing and a second relief valve R4 protects the separator and transfer vessel in the event the condensor fan fails. The loop can also be used to test various other secondary fluids by minor alterations to the piping.

2.b. Results and Discussion

Large scale tests using R-113 were begun in 1975 at the Raft River Site in Idaho. Since no work had been done previously by any researchers on a boiler of the size and type installed by the University of Utah, time was spent developing operating and measurement procedures to insure reliable test data.

One problem, since encountered by others investigating direct contact heat exchangers (Suratt and Hart, 1977), was determination of the working fluid saturation temperature under any given set of boiler conditions. The problem was resolved by determining that the water in the boiler trays is a saturated liquid and therefore the partial pressure of the working fluid vapor is simply the total pressure less the saturation pressure of the water. This method of determining the partial pressure was verified by measuring the amount of water in the condensate. However, the water measurement is not a reliable short term means of determining the partial pressure. This was also the experience of Suratt and Hart (1977). Therefore, the water measurement method was used only in long duration runs to verify the calculation technique that has been used on all the testing to be reported in the remainder of this section.

Testing proceeded at Raft River for eight weeks and then was resumed at the University of Utah. The campus site was more convenient to shop facilities and permitted frequent changes of configuration. A physical plant boiler was used as a source of fresh hot water at temperatures up to 150°C (300°F) and pressures up to 90 psig. During the time the R-113 tests were being conducted at the campus site the new pentane test loop was constructed.

The new pentane test loop is currently operating at the Raft River Test Site. Only basic system checks and debugging have occurred to date but these preliminary tests have demonstrated the smooth operation of the remote control system and the feasibility of boiling a light hydrocarbon on a water surface.

A tabulation of all the results of the R-113 tests in the various tray configurations is given in the Appendix. Values of heat transferred to the R-113 (Q_D) and heat transfer coefficient ($Q_D/\Delta T_{LM}$) are tabulated as well as temperatures and pressures of prime interest. The fluid mass flow ratio and a carryunder indicator are also listed. Carryunder is the condition when boiling of the working fluid is incomplete and liquid is lost from the boiler with the brine. An indicator of 0 means there is no carryunder. The indicator 1 denotes detection of R-113 (by smell) in the spent brine and a value of 2 indicates the measured vapor exit temperature was slightly less (about 1°F) than the calculated saturation temperature. The latter is weak indication of carryunder or a near-carryunder condition.

i. Heat Transfer Coefficients

Overall boiling heat transfer coefficients as high as $UA = 4200$ BTU/hr-°F were obtained in the tray boiler. Heat transfer rates as high as 130,000 BTU/hr were also obtained. Since both of these figures are directly dependent upon the Freon (and brine) flow rate, higher values could have been obtained by simply adding more pump capacity and increasing the flow rates (at a constant mass flow ratio). It is anticipated that the tray in use could handle 20 gpm of brine if there were sufficient pump capacity to supply the Freon and brine. With 20 gpm of brine a boiling heat transfer coefficient, UA , of 12,500 BTU/hr-°F could be obtained.

Taking a volume equal to that of a cylinder in which the entire tray and nozzle arrangement could fit, volumetric boiling heat transfer coefficients of $12,000 \text{ Btu/hr-}^\circ\text{F-ft}^3$ could be obtained. Once again, the reason this high coefficient was not attained in the boiler was merely insufficient pump capacity. The highest volumetric coefficient actually attained was $4700 \text{ BTU/hr-}^\circ\text{F-ft}^3$ at a brine flow rate of 8.0 gpm. If the flow rate were raised to 20 gpm brine and complete Freon boiling at the same mass ratio were achieved the volumetric boiling coefficient would be $12,000 \text{ BTU/hr-}^\circ\text{F-ft}^3$.

In this regard it should be noted that the economic analysis presented in Chapter V assumes a volumetric heat transfer coefficient of only $2500 \text{ BTU/hr-}^\circ\text{F-ft}^3$ in the boiler. This is a very conservative estimate because, as was just noted, coefficients 90 percent higher than this have already been achieved in a large scale boiler.

ii. Mass flow ratios

Complete boiling of the working fluid requires two conditions: there must be sufficient energy available in the brine to vaporize the fluid and the two fluids must be in contact long enough for the heat transfer to occur. The first requirement can be calculated from a simple energy balance. For a given brine inlet temperature to the boiler and a given boiler pressure (or saturation temperature) there is a limit to the amount of R-113 that can be boiled by a given amount of brine. This limiting ratio, or minimum mass flow ratio (brine/working fluid) is shown in Figure III.15 for R-113, pentane, and isobutane with brine at 290°F . Clearly, different fluids can attain different mass ratios depending upon their vapor pressures, heats of vaporization and other thermodynamic properties.

Figure III.15. Minimum attainable mass ratio calculated from an energy balance (5°F approach temperature)

Achievement of the required contact time between the fluids, or residence time, is more difficult to predict. But if the minimum mass ratios predicted from the energy balance can be achieved in a boiler then it is obvious that the residence time criterion has been met. Figure III.16 shows mass ratios that have been achieved in the University of Utah direct contact R-113 boiler. It can be seen that complete boiling is achieved at the limiting mass ratio. Therefore the tray type boiler does have sufficient residence time for the flow rates used to this point. Figure III.17 shows similar results using a nozzle with a lower pressure drop and hence a lower spray velocity. This nozzle also permits attainment of the minimum mass ratio. Figure III.18 shows results of measurements in a spray tower (Suratt and Hart, 1977) which also achieves the minimum brine flow requirements in a small scale apparatus.

A series of experiments was performed to systematically vary the height of a solid cone nozzle above the circular "source flow" tray. Variation in nozzle height allowed variation in the effective residence time for the tray. It was found that boiling was complete, or residence time sufficient, even at the high nozzle positions where the spray covered the entire tray. This result was encouraging in that it implies boiling would be complete even at high total flow rates with a lower nozzle height because in the present experiments the residence time was more than sufficient. This supports the belief that the brine flow in the present trays could be increased to 20 gpm to yield the desired higher heat transfer coefficients.

Further investigation of the residence time requirement was carried out at Raft River using the rectangular tray and a flat spray nozzle (Figure III.13). Use of the rectangular tray with a variable weir height

Figure III.16. Comparison of actual mass ratios and the predicted minimum for a high-pressure-drop nozzle

Figure III.17. Comparison of actual mass ratios and the predicted minimum for a low-pressure-drop nozzle

Figure III.18. Comparison of actual mass ratios and the predicted minimum for a volume-type boiler (Suratt and Hart, 1977). No carryunder data points were reported.

permitted independent variation of the brine flow rate and depth. This made it possible to determine the residence time required for boiling of the R-113 sprayed onto the brine. Only one data point was obtained because of the large amount of time required to determine the residence time limit. It was found that 7.5 seconds were required to boil R-113 with 260°F brine at a depth of 1.5 inches. This information was used to design a circular tray to pass 10 gpm of brine and boil R-113 at the minimum mass ratio attainable. The tray was constructed and did achieve the desired performance. This cursory look at the complex issue of boiling time demonstrates methods that might be used if further understanding is sought.

The minimum mass ratio which gives acceptable performance was also determined as a function of brine inlet and outlet temperatures and the R-113 saturation temperature. Figure III.19 show mass ratio data from both large- and small-scale tests. The minimum acceptable mass ratio can be calculated from the equation for the curve shown in the figure:

$$\frac{\dot{m}_{H_2O}}{\dot{m}_{R-113}} = 1.69 \left[\frac{c_{p_{H_2O_l}} (T_{H_2O_{ave}} - T_{SAT})}{h_{fg_{R-113}}} \right]^{-1.09}$$

Any ratio below this value will result in incomplete boiling and carry-under of working fluid.

iii. Approach temperature in the boiler

The approach temperature that can be achieved in a heat exchanger is very important to the overall power cycle performance. Therefore it is important to have a means of determining the approach temperature a boiler configuration can produce.

Approach temperatures achieved in the University of Utah boiler with two different nozzles are shown in non-dimensional form in Figure III.20.

Figure III.19. Acceptable mass flow ratios for boiler operation - Brine-R-113

Figure III.20. Normalized approach temperature in the surface-type direct contact boiler

Non-dimensional approach temperature equal to 0.10 are not at all uncommon and correspond to actual approach temperatures of 3-5°C (5-10°F). A slight difference can be seen between the approach temperature produced by the two nozzles. The high pressure drop nozzle (60 psi at 1.2 gpm) gave lower approach temperatures than the low pressure drop nozzle (10 psi at 1.2 gpm). This is as would be expected from the high speed, small droplet spray from the high pressure drop nozzle.

iv. Heat transfer correlation

Heat transfer coefficients for the large scale boiler tests as well as the bench test volume boiler can be correlated by the equation

$$St = 2.0 H^{0.6} \left(\frac{\dot{m}_{cp}}{\dot{m}_{dp}} \right)^{0.15} \quad (III.2)$$

where

$$St = \frac{UA}{\dot{m}_{dp} C_{p_{\ell dp}}} ,$$

$$H = \frac{Pr_{dp_v} h_{fg_{dp}}}{C_{p_{v_{dp}}} (T_{H_2O_{mean}} - T_{SAT})}$$

and

$$Pr_{dp_v} = \text{Prandtl number of the vapor dispersed phase.}$$

This correlation is shown in Figure III.21. All of the results for both volume and surface boiler obtained from the large scale experiments agreed reasonably well with Equation III.2 indicating that at large scale there was no great difference in how the dispersed phase was injected. In carrying out the surface boiler experiments several different inexpensive commercially available spray nozzles were used. For the volume boiler

Figure III.21. Non-dimensional heat transfer versus JAKOBxPRANDTL numbers

several different orifice sizes and separations were used with no detectable trend. This indicated that for a large scale system the induced turbulence in the continuous phase was sufficient to break up the drops or jets of dispersed phase fluid such as to make the heat transfer processes independent of initial drop size. This then made many of the single drop studies and small scale studies inapplicable to extend to full scale studies.

v. Comparison with other investigators

Comparison with the work of other contractors is limited due to the sparsity of data available from their reports. However Figure III.21 does show the results of comparison of the work of DSS Engineers and Ben Holt Company with our data and correlation. DSS Engineers designed a simple spray column boiler of six inch diameter to operate with isobutane as a working fluid. Since isobutane was chosen they required a highly pressurized system and were unable to obtain visual characteristics of their experiments. The work of Ben Holt company was with hexane and could be conducted in a glass column. However data was limited. Neither investigation team has yet published results with real geothermal brine. However, our results indicate that no great changes should occur. The experiments of DSS Engineers appear to suffer somewhat from incomplete instrumentation, but our best attempts at data interpretation indicate reasonable agreement with our own data and within our scatter. A similar conclusion can be made for the one plotted point from Ben Holt Co.

What do the above results mean as to boiler design? We interpret them to mean that all of the boilers investigated give comparable results when presented in terms of non-dimensional variables, and at this time, the correlation equation of Equation III.2 can be used to design the boilers whether they be surface or volume type and in trays or columns.

III.B. Turbines*

1. Performance Considerations.

The vapor flowing through the turbine in a direct contact binary plant is a combination of water vapor and working fluid vapor. In this regard it is a different situation than either the flash system or the "conventional" binary system.

A variety of studies have been performed to evaluate turbines for various organic working fluids. One recent example is the excellent treatise by Milora and Tester (1976). For their analysis they used the theory developed by O.E. Balje (1962).

Candidate fluid selection criteria is more stringent for the direct contact cycle, in that a working fluid that is minimally soluble in water must be considered. When this aspect is included with the vapor pressure characteristics and cost, six fluids were singled out for analysis in the present situation: isobutane, Freon 113, Freon 114, N butane, pentane and propane.

The theory of Balje (1962) can be readily extended to the case of a two-component flow. This is accomplished through the concept of energy density (defined for the present work as the total enthalpy change across the turbine divided by the exiting specific volume). With the definition of the entering and exiting thermodynamic states, the definition of the thermodynamic path and the thermodynamic properties, the energy density is easily calculated. Details of this are given in DGE/1549-7. The pertinent thermodynamic properties were evaluated by computer subroutines developed here (Riemer, et al., 1976 a,b,c).

*Most of the information given here is summarized from the report, "Analysis of Binary Vapor Turbines," by R. Bliss, R. Boehm and H. Jacobs, DGE/1549-7, December 1976.

A variety of possible entrance and exit states exist. In general these were treated parametrically. However, to reduce the number of variables somewhat it was assumed that the vapor stream entering the turbine consisted of saturated water vapor and saturated working fluid vapor. A few cases were also evaluated to assess the effects of superheating.

For the assumption of both vapors in the mixture being saturated, the ratio of the mass flow of water to working fluid is shown in Figure III.22. Since propane is supercritical over a wide range of this figure, an alternative approach had to be used for it. The band of values given for it were found by maximizing the energy density.

Since the primary factor in determining turbine size is energy density, this important variable is compared for the various fluids in Figure III.23. Note that this increases with increasing pressure fluids, primarily through the specific volume divisor.

The actual variation in turbine diameter and rotative speed for the six fluids and single stage operation is shown in Figures III.24 through III.29. Two items should be noted in passing. The term "Optimum Diameter" as shown on the left hand scale refers to the fact that these results were all evaluated at the optimum efficiency point (approximately 90%) as defined by Balje. A second item noted there is that a "maximum allowable diameter" is shown on some graphs. This value corresponds to a rotor about six feet in diameter and is simply used to bring attention to the fact that some fluids may require overall diameters (rotors plus case plus other items) that may be too large to be easily transported.

A comparison of the various turbine parameters for three different turbine output powers is shown in Figure III.30. As might be expected the

Figure III.22. Ratio of mass flows in a turbine inlet

Figure III.23. Comparison of the energy density of the six fluids

Figure III.24. Freon 113 optimum diameter and rotation speed. The net turbine power is ten megawatts.

Figure III.25. Freon 114 optimum diameter and rotation rate. The net turbine power is ten megawatts.

Figure III.26. Isobutane optimum diameter and rotation rate. The net turbine power is ten megawatts.

Figure III.27. Optimum diameter and rotation rate of butane. The net turbine power is ten megawatts.

Figure III.28. Optimum diameter and rotation rate of pentane.
The net turbine power is ten megawatts.

Figure III.29. Optimum diameter and rotation rate of propane. The net turbine power is ten megawatts.

Figure III.30. Optimum diameter and rotation rate of pentane for different turbine powers

variations of diameters and speeds with output power are not linear.

The thermodynamic process lines were analyzed for two cases (R-114/water and butane/water) to give more insight to the expansion process. Results of this study are not given here, but are shown in the original report.

Variation of the exit quality with operating conditions is of great interest. Some concern might reasonably exist regarding the amount of moisture in the turbine passages. However, as shown in Figure III.31, the total exit quality (defined as the mass of vapor divided by the total mass) is at most very low for conditions comparable to those shown in the figure.

Several other aspects were also studied in the original report and are not given here. Included are comparisons between the direct contact turbine and the simple binary turbine (i.e., the same power output without water vapor) and a first cut attempt to multistaging. This latter analysis was not deemed too successful, but it was also considered that multistaging is not required for plant output up to the 10 megawatt case.

2. Relative Cost Estimates

Costs were estimated with the empirical relationship developed by the Barber-Nichols Engineering Company (Nichols, 1975). The cost equation incorporates the following geometric features of the turbine:

- last stage pitch diameter
- blade height to diameter
- number of stages
- number of exhaust ends
- inlet pressure
- tip speed of the last stage.

Figure III.31. Total turbine exit quality versus inlet temperature

The cost equation is considered applicable for one to one hundred megawatts per exhaust end. It is recommended that the pressure ratios per stage should be about .7, although this is not a requirement. The equation applies to axial turbines which have efficiencies of .85 or better. The equation is as formulated for inlet temperatures of 100 to 300°C and condensing temperatures of 25 to 50°C for fluids with molecular weights between 17 and 200. The cost equation is:

$$\text{Dollars} = (1.04S - .04S^2)f_p(30135nf_uD^{2.1} + 16775D^3 + 20720D^2)$$

$$f_p = 4.334 \times 10^{-6}P + .9707 \text{ for pressures } \geq 34.5 \text{ bars}$$

$$f_u = -2.469 + .00274U - 7.424 \times 10^{-7}U^2 + 6.9263 \times 10^{-11}U^3 \geq 1.0$$

The multiplier $(1.04S - .04S^2)$ is for multiple exhaust ends on a common shaft and the multiplier takes into account the reduction in price as a result of the elimination of some of the main shaft seals and bearings. The multiplier is valid for one to four exhaust ends.

The term $30135nf_uD^{2.1}$ reflects the cost of the turbine stages. For the fluids and temperatures under consideration in this study the f_u factor is almost always one.

The $16775D^3$ term takes into account the casing and inlet manifold and exhaust collector costs of the turbine. The remaining term, $20720D^2$, adds into the cost equation the expenses of the main shaft seals, radial and thrust bearings and the turbine shaft.

The cost equation is for turbines which have last stage blade to pitch diameter ratios of .11 which is nearly the same as used in this study. For blade to diameter ratios of about .33, the cost should be increased by 25%. The blade to diameter ratios are typically .33 or larger in the last stage of low pressure steam turbines because the exit specific volume of the steam is very large.

Using the Barber-Nichols' cost equation and assuming only one exhaust end, one stage and a pressure coefficient of one, the cost and energy density are compared as a function of the enthalpy drop across the turbine and the exit specific volume in Figure III.32.

It must be realized that the cost model changes with time, but it is seen that the cost varies approximately as the inverse of the energy density. In the neighborhood of one hundred thousand dollars the line of constant dollar cost is nearly parallel to the line of constant energy density.

The range of this study is contained well within the range of enthalpy drops and exit specific volumes given in Figure III.32. Holding the enthalpy drop constant the cost of a turbine increases as the exit specific volume increases because the required diameter is larger. On the other hand, if the specific volume is held constant, it is seen that the cost increases with decreasing enthalpy. To maintain a constant power output as the enthalpy drop is lowered, the volume flow rate must be increased, which, in turn, increases the physical dimensions of the turbine and, hence, the cost.

Figure III.33 shows a sample case of the variation of cost with condenser temperature and inlet temperature. Also, to show how the cost varies with the turbine power, Figure III.34 is included.

Costs for various turbines are compared in Figure III.35. Freon 113 stands out as a particularly poor fluid under turbine cost considerations. The remaining fluids, including pentane, appear to give reasonably affordable turbines and the decision to choose among them should depend on the cycle analysis which estimates efficiency and the cost of lost fluid. As can be seen, Freon 114 and pentane come together at 140°C. If Freon 114 followed the optimal energy corridor instead of the saturation line, the

Figure III.32. Turbine cost and energy density comparison. The turbine is assumed to produce ten megawatts.

Figure III.33. Turbine cost as a function of inlet and condenser temperature for R-114 and water vapor. Turbine net power is ten megawatts.

Figure III.34. Turbine cost for different power outputs

Figure III.35. Comparison of turbine costs using the six fluids. Turbine net power is ten megawatts.

minimum in the Freon 114 curve would not appear and the curve would continue to decrease monotonically.

The butane, isobutane and propane curves are grouped together to the point where the differences may be negligible. Other design considerations, such as the better seals that would be required for propane or fluids with high energy densities because of their high pressures, might reduce their apparent cost advantage. Only one stage is considered in these graphs. Perhaps the high energy density fluids might require staging to reduce tip speed or exhaust energy if efficiency begins to decrease significantly. This is an area which might warrant further investigation.

In general it was found that turbine costs for DCHE plants are lower than for "conventional" binary plants. This is due to the percentage of water in the working vapor.

A final note of caution is given. There is reason to believe that the absolute magnitude of the costs given by the Barber-Nichols' cost equation may not be very accurate (Eskesen, 1977). While the results shown here are not all underestimated the same amount, they may all be underestimated. However, we feel that the relative values are approximately correct. More accurate costs will be available from the manufacturers when they begin selling turbines for these applications.

Obviously, the turbine cost is only one component that contributes to the overall evaluation of DCHE plant costs. Several other factors must be considered in evaluating comparisons with other cycles.

III.C. Preheaters

1. Overall Considerations

During the reporting period primary emphasis has been directed toward boilers rather than preheaters. However, a large portion of the heat transfer in a direct contact cycle will be between liquids. The primary reason that the emphasis was placed on boiler development is that relatively more is known about liquid-liquid heat transfer (Jacobs, 1977). Several specific questions still remain in the latter category though, and our program is directed toward giving comprehensive answers to these questions so that rational plant design and evaluation can be made.

Many fundamental differences exist between the functioning of boilers and the functioning of preheaters. Perhaps most importantly, in preheaters the duty is sensible rather than latent so the heat transfer performance distinction between counter and parallel flow becomes real. Another factor that must be kept in mind is that the temperature of the brine is at its lowest point of the system in the preheater. This has at least two possible ramifications. Since the equilibrium solubility of working fluid in brine goes up with decreasing temperature, this stage could be the determining factor on working fluid losses from the cycle. Second, the brine will be closest to the saturation point with regard to the total dissolved solids (i.e. as the temperature of the brine goes down it will not be able to sustain as much dissolved solids in solution). This factor results in the possibility of precipitation of solids from heavily-laden brines, with the associated fouling phenomena.

The liquid-liquid heat exchanger, which carries approximately thirty per cent of the heat duty (this obviously varies with the particular working fluid and operating conditions) will require greater volume than the phase

change devices. This greater volume normally implies greater cost. The liquid-liquid exchanger is also more susceptible to the perfdities of the brine as scaling and precipitation will occur more readily at lower temperatures.

2. Parallel-Flow Concepts

2.a. Stratified Types

The type of direct contact heat exchanger which would allow for non-equilibrium solubility and thus minimum loss of working fluid would have to take advantage of preferred diffusion of heat to mass. This could be accomplished by having the two fluids flowing in layers. Normally this layer flow could best be produced by laminar co-current flow. Since the flow is co-current, the exchanger must be segmented so that countercurrent operation can be approximated.

A model of a laminar flow contactor was analyzed in this laboratory, and a simple schematic of the system is shown in Figure III.36. Details of the theoretical analysis are given by Johnson and co-workers (Johnson, et al., 1975 and Jacobs, et al., 1976). Typical of the results found is the plot of the dimensionless length shown in Figure III.37. An experimental model has also been constructed and is shown in Figure III.38. With this work, the performance has been quantified. It is concluded that the required equipment size is probably too large compared to other approaches and alternative configurations will be used in all cases except those where fluid loss minimization is the critical requirement.

2.b. Mixed Types

Co-current heat exchangers of all forms offer another advantage over countercurrent devices in that they are pressure driven. This factor allows higher fluid through-puts. Pipeline contactors are a good example of a high

Figure III.36. Schematic of the general problem. The direction of flow is the z direction and the origin is at the interface.

Figure III.37. Non-dimensional length ($\zeta = z/aPe_A$) required for a specific fraction of total heat transfer as a function of $(Q_A \rho_A c_{pA} a K_B)/(Q_B \rho_B c_{pB} b K_A)$

Figure III.38. Liquid-liquid direct contact heat
exchanger test apparatus

flow device. (It must be reiterated, however, that the mixing cup temperature is the maximum limit expected in any co-current device.) Porter and co-workers (1968) obtained volumetric heat transfer coefficients as high as 200,000 Btu/hrft³F in a three inch diameter pipe with water or seawater and two different petroleum oils. These devices require efficient settlers/separators to allow the working fluid to be removed from the water. In an application to desalination, Wilke, et al. (1963) found settlers and separators to be small cost items; however for geothermal power applications the design of the settler/separator remains the major deterrent when applied to highly fouling brines. For this reason perhaps a combination of a turbulent pipe and stratified flow contactor as used by Grover and Knudsen (1955) is more advantageous. Although this device demonstrates only one theoretical stage, its lowest heat transfer rate (6,000 Btu/hrft³F) is greater than that obtained for spray columns which themselves obtain little better than one or two theoretical stages with volumetric heat transfer coefficients of only about 2,000-4,000 Btu/hrft³F.

3. Counterflow Configurations

Counter current devices that could be suggested as liquid-liquid heat exchangers are shown in Figure III.39. They include spray towers, baffle towers, perforated plate towers, packed towers and wetted wall towers. All of the first four are characterized by acting to disperse one phase as drops within the other. The fifth keeps the phases separated except at a cylindrical interface. The advantages of the former are a higher heat transfer area within the contacting volume, while the advantage of the latter is in maintaining separation and perhaps eliminating or reducing the need for the separation system.

Figure III.39. Schematic of various types of direct contact counter flow devices

3.a. Spray Tower

The device which has received the most attention as a direct contact liquid-liquid heat exchanger is the spray tower. As can be seen in Figure III.39 it consists of a vertical column, an injection nozzle for each fluid, working fluid and brine, and exit ports. A similar configuration was used as a boiler concept in this laboratory (See section III.A). Normal operation would theoretically have the heat transfer taking place between the two inlets with coalescence occurring a small distance above. Coalescence of the dispersed phase is induced by a screen made of an oleophillic or hydrophillic material depending on whether the working fluid or brine is dispersed. Design of a spray tower can be carried out following the outline described by R. Letan (1976). The design method requires only that the Reynolds number for the drops be less than 2300. For Reynolds numbers greater than 2300 the flow is turbulent and scaling of the size is not possible. The advantage of the system is its simplicity of design. However, it has several drawbacks. A spray tower suffers from strong back mixing. That is, the hydrodynamics for high through-put cause considerable back circulation, especially at higher drop Reynolds numbers. At Reynolds numbers below 2300 the heat transfer is strongly affected by the attached wake (Letan and Kehat, 1968, for example) thus there is poor heat transfer in the central portion of the tower. Typically the net effect is that a spray tower only operates effectively near the dispersion nozzle and the coalescence screen. Thus, the number of transfer units, NTU, is limited to between 1 and 3 (Letan, 1976) depending on holdup and back mixing.* Hence

*Treybal (1963) points out that backmixing is a function of height-diameter ratio and increases as the reciprocal. Thus Steinmeyer and Woodward (1969) for a 3 ft diameter tower were only able to obtain 1.3 to 2 theoretical trays with $H/D = 2$ while for a 6 inch diameter tower ($H/D = 10$) Markowitz and Bergles (1968) recorded 2 to 3.

a typical direct contact cycle would require five or more contacts hence requiring two or more spray towers in series and reducing its advantages over a co-current flow device, and significantly increasing its cost.

3.b. Baffled Towers

Baffled towers overcome the disadvantage of spray towers in that the baffles limit backmixing to the region between baffle plates. However, the region between plates is nearly isothermal. Thus, it provides any number of theoretical trays depending on the mixing efficiency between plates and the number of plates. Fair (1972a) indicates that such columns have been used with gas-liquid systems as heat exchangers but no data has been published for liquid-liquid systems. To further complicate the design no mass transfer data on baffle columns has been published; however, as a mass transfer device general observation has shown they are about 50 per cent as efficient as perforated plate towers. They do, however, allow for a greater through-put. General design would utilize the E_h for sieve towers multiplied by 50 per cent. General information on sizing baffle columns is presented by Scheiman (1965a,b). Baffle towers may be used effectively in high fouling situations such as with geothermal brines of high solids loading.

The primary reason for lower efficiency with baffled towers than with sieve plate towers is the limited redispersion of the "dispersed" phase as compared to the sieve towers in gas-liquid systems. This can be partly overcome by higher continuous phase velocity. For liquid-liquid operation the same is true. In particular, enhanced turbulence between plates or coalescences will improve the efficiency. This may also be obtained perhaps more effectively by mechanical agitation. For the geothermal application Lightning CM Contactors or Rotating Disk contactors as shown in Figure III.40 might be effective. Treybal (1963) provides some information on the design

Figure III.40. Mechanically agitated direct contactors

of such towers. Strand (1962) provides adequate information for the scale up of the rotating disk contactor as a mass transfer device which can be used to size the heat transfer assuming $E_h = E_m$. Mechanically agitated towers for liquid-liquid operation have been built up to ten feet in diameter. As a heat transfer device it must be noted that they introduce a small additional parasitic power requirement and that seals are necessary around the power shaft.

3.c. Perforated Plate Towers

Perforated plate towers or sieve towers have been recommended by I. Sheinbaum (1975, 1976) for use as a liquid-liquid heat exchanger. This device has the highest efficiency for mass transfer of all counter current gravity driven devices. It can be designed as a spray tower with only the high efficiency zones of a spray tower (end effects). Thus, the heat transfer of Steinmeyer and Woodward (1969) can be applied as well as the great amount of mass transfer described by Treybal (1963). Normally the design of the tower is such that the continuous phase wets the plates; however, with the brine as a continuous phase this could lead to fouling of the surfaces; thus, it is preferable to use a hydrophobic surface which would require the types of perforations shown in Figure III.41. The major drawbacks to this device are that the through put is reduced considerably with less than 26 per cent of the flow area available for the dispersed phase in each plate and the possibility of fouling.

3.d. Packed Towers

Packed towers have received more attention as a condenser than as any other heat transfer device. As such, volumetric coefficients as high as 60,000-150,000 Btu/hr ft³ °F have been obtained (Wilke, et al., 1963). As a liquid-liquid heat exchanger they have the lowest through put, are subject

Figure III.41. Design of perforations for fouling fluid application

to fouling and suffer from backmixing although not as severe as with a spray tower (Treybal, 1963). As a liquid-liquid mass transfer device published data is limited to small diameter towers - 3, 4 and 6 inch diameters (Liebson and Beckman, 1953). When compared with spray towers for the same duty, the NTU were reduced by factors of up to six depending on the packing used. In application to geothermal power production it is recommended that their use be directed toward the direct contact condenser.

3.e. Wetted Wall Towers

Wetted wall towers have never been seriously considered for industrial application as mass transfer devices; however, they have received attention in the laboratory for several decades (Treybal, 1963). Basically they are constructed so that one fluid wets the wall of a vertical circular conduit with a central core flowing counter-currently. An alternate scheme would utilize an inclined duct with the flow stratified with the heavier fluid flowing along the lower surface and the lighter flowing upward above it. In order to obtain the former flow situation, the walls should be constructed of a material not wettable by the central flowing fluid. For the case of the geothermal application such an exchanger should be hydrophobic and oleophilic. This would allow the working fluid to flow along the walls and reduce the possibility of fouling. The flow in such a device would be maintained laminar. For the case of a hydrodynamically developed flow the double pipe counter flow heat exchanger analysis of Stein (1966) and Blanco, Gill and Nunge (1968) should be applicable utilizing the flow rates set by the fully developed flow equations for no intervening wall. As the flow will be laminar the heat transfer coefficient will normally be small and the surface area is restricted to the interfacial area. To overcome these deficiencies as compared to all of the heat exchangers previously described, heat exchangers

made of multiple tubes as shown in Figure III.42 would have to be constructed. The primary advantage of this type of heat exchanger would be the elimination of large volumes necessary for settling and separation as compared to the prior devices.

4. Comparisons of Various Configurations

Based upon the above discussions, several different designs are possible for direct contact heat exchangers for use as a preheater in a geothermal direct contact binary cycle. In construction the simplest are the spray tower, the wetted wall tower and the turbulent pipe contactor. However they have additional complications in the fact that the spray tower suffers from back mixing, the wetted wall tower may encounter design difficulties in its inlets and outlets and the turbulent pipe contactor requires separator/settlers and several stages if it is to approximate counter current flow. The baffle, sieve and RDT require internals of increasing complexity, but offer potentially better performance. An example of the relative size of the preheaters is best illustrated by considering a specific case. Thus, we will look at the preheater for the system of Figure III.43. Although it is shown as a baffle tower it could be any of the designs mentioned. The brine enters at 220°F with a flow rate of 967,000 lb/hr and leaves at 150°F . 790,000 lb/hr of isopentane enters at 80°F and leaves as a liquid at 210°F . The total heat duty is thus 62,000,000 Btu/hr. Following the design suggestions from the appropriate references discussed previously, the sizing for the heat transfer zones for six different types of heat exchangers was calculated and is shown in Table III.1. Also tabulated are the conditions for holdup, the volumetric heat transfer coefficients attainable for the designs, and an indication of the auxiliary equipment requirements. For the spray tower two different heights are indicated. Based on data obtained

Figure III.42. Typical manifold for multiple tube wetted wall direct contact heat exchanger

Figure III.43. Cycle diagram used for preheater comparisons.

TABLE III.1.

COMPARISON OF DESIGNS OF VARIOUS LIQUID-LIQUID DIRECT CONTACT HEAT EXCHANGERS

Type	Duty	Diameter	Height Heat Transfer Zone	Holdup	Volumetric Heat Transfer Coefficient	Area Heat Transfer Coefficient Internal Equip.	Auxiliary Equipment
Spray*	62,000,000 Btu/hr	9.2 ft	14.88	26% (90% flooding)	2000 Btu/hr ft ³ °F	-	Coalescence Volumes
			9.5	"	3150	-	
Baffle (Disk-Donut according to Treybal)	"	7.9 ft	16.74 ft	40%	2400** (50% Sieve Tower)	18" Spacing Baffles	Coalescence Volumes
Sieve	"	7.9 ft	8.37	40%	4800	24" Spacing Trays 1/8" Ø holes 1/2" square spacing	Coalescence Volumes
RDT	"	7.0 ft	7.0	25% (75% flooding)	8000***	Impeller } 3.5' Dia } Stator } 4.68' Inner Dia } 18" Spacing	Coalescence Volumes 21 HP motor to drive impeller
Wetted Wall Tower	"	102,100 tubes 0.7" diameter (273 ft ² active area, 347 ft ² gross area)	8.82	-	646	16.8 Btu/hr ft ² °F	Manifolds Exit and Entrance
Turbulent Pipe Contactors	"	0.5 ft 5 stages with stages pressure increase	3.96 ft 4.75 ft 6.73 ft 8.3 ft 15.83 ft	-	200,000 (Reynolds Number 6.7 x 10 ⁶)	-	5 Settler/ Separators 5 Brine Pressure Reducers

*For Geometry Reasons Lower Value U_v Probable

**See Discussion on Baffle Towers

*** $U_v = 5U_v$ Stagnant Drops (Fig 11.31, Treybal, 1963)

from small diameter towers with height-to-diameter ratios of 10 to 20, volumetric heat transfer coefficients as high as $3150 \text{ Btu/hr ft}^3 \text{ } ^\circ\text{F}$ should be possible; however, for small height-to-diameter ratios Steinmeyer and Woodward (1969) found that the heat transfer was considerably reduced. The value of $2000 \text{ Btu/hr ft}^3 \text{ } ^\circ\text{F}$ is more consistent with their work for towers of height-to-diameter ratios of two. Thus the larger height of 14.88 feet for the heat transfer zone is more realistic. This may, due to back mixing, have to be divided into two units with additional volume for separation of the fluids as compared to a single unit.

The volumetric heat transfer coefficient for the baffle tower is very conservative and could be as high as $3150 \text{ Btu/hr ft}^3 \text{ } ^\circ\text{F}$. The design of the sieve tower is consistent with that of Sheinbaum (1975). The wetted wall tower is shown to require from four to five times the volume for the heat transfer as compared with even a spray tower; however, if simple entrance-exit zones could be designed, it might be comparable to the spray tower in actual size but would be expected to be much more costly to manufacture. The RDT design is based on limited mass transfer data and heat transfer data needs to be acquired.

The turbulent pipe contactor is the only cocurrent device shown in Table III.1. It appears quite attractive depending upon the state of the art of settler/separators.

III.D. Direct Contact Condensers

1. Introduction

With the development of direct contact heat exchangers for the preheating and boiling functions in a binary cycle, the single greatest remaining equipment cost involves the condenser. Since the quantities of heat that must be handled by those devices are large (almost the same as that transferred in the heat addition portion of the cycle), and since the condensing temperature has such a profound effect on the cycle performance (see the discussion of this aspect in the section "Systems Analysis"), overall success of the binary cycle could be profoundly affected by the condenser design.

Since direct contact processes offer the potential of improving performance while using lower cost devices, we have begun an investigation of several aspects of direct contact condensers.

Direct contact condensers operating in geothermal power cycles face a somewhat different situation than do the heat addition devices. This is in addition to the obvious aspect of boiling versus condensation. Since generally clean water can be used for condensing, there is essentially no fouling problem. If the condenser is tied to a cooling tower through the cooling water loop, there is not a serious solubility loss problem. This is due to the fact that the recirculated water is essentially a closed system that will saturate with working fluid, causing no further loss of working fluid.

If the potential of a direct contact condenser is to be evaluated, three general effects must be assessed. These relate to the heat transfer performance, equipment costs and the cooling tower penalties and/or fluid loss possibilities. The heat transfer performance could be superior

or inferior to conventional shell and tube designs. This can be evaluated, and some preliminary thinking in this area is outlined in the following subsection. The capital investment may be considerably different for the direct contact condenser. If this happens to be much less than a conventional design, this might overshadow other possible drawbacks since geothermal plants are capital intensive. One other major factor which must be assessed is a possible performance penalty if closed cooling towers are used. This type of tower may be required either for conserving water in arid regions, or it may be necessary to cut down on working fluid losses that might be present if conventional towers are used. It should be noted in passing that direct contact condensers also offer a means of separation of noncondensibles (degasification of the working fluid). Although other techniques also exist (see the section on "Fluid Loss Considerations") this could be an important characteristic. Since our work in this area is just beginning, these various factors remain to be analyzed. We have just begun heat transfer work and this is summarized in the next section.

2. Direct Contact Condensing Devices - Heat Transfer Considerations*

Direct contact condensers have been known for a considerable period of time; however, little is published as to their design. Primarily they have been used for condensing steam, as open feedwater heaters or for solvent recovery in petroleum refineries. In most cases the same fluid is used for condensing as is the vapor, and as such they are known as single phase condensers. The most common form of direct contact condenser is the barometric or low level condenser used in steam power plants. This type of condenser works by having drops of a cold fluid falling through a

*This is a summary of the report "Direct Contact Condensers - A Literature Survey", by H. R. Jacobs and H. Fannar, February 1977. ERDA document number DGE 1523-3.

saturated vapor. Another method, more commonly applied to feedwater heaters is having the hot vapor, either saturated or superheated, bubble through a liquid pool or stream. A good overview of various aspects was published a few years ago (Fair, 1972).

The direct contact condenser that might be applied to the direct contact geothermal power plant would of necessity be condensing a mixture of steam and an immiscible working fluid. In general the working fluid would consist of more than 90-95% by mass of the vapor. It thus differs from the previously mentioned condensers in that two phases are present in the condensed liquids. As both steam and working fluid are condensing the heat transfer is not necessarily reduced, as is the case when a non-condensable gas is present within the vapor.

The types of equipment that can be used for direct contact condensers are primarily based on whether one chooses to have condensation on drops or by collapsing bubbles. However, condensation on thin films is also possible. The type of equipment can also be selected on the basis of whether or not desuperheating makes up part of the heat load. Thus, duty as well as the basic mechanism constitute the means of categorizing equipment.

2.a. Saturated Vapor

For condensation of saturated vapor there is no advantage of counter-current versus co-current flow. Thus, most direct contact condensers for this application are in fact really co-current devices. The actual physical arrangement of counter current flow may be advantageous, if appreciable amounts of non-condensable gases are present, to aid in deaereating the condenser, however.

i. Film Type Direct Contact Condensers

In order to obtain large surface areas and thus large volumetric heat transfer coefficients in a film type condenser, this type of condenser is designed as a packed bed and is shown schematically in Figure III.44. A variety of bed materials can be used for packing such as Raschig, Lessing and Pall rings and Berl and Intalox saddles. Expanded metal lath could also be used.

Packed beds have been used in designs of sea water evaporation plants at the University of California and some information published (Wilke, et al., 1963). Harriott and Wiegandt (1968) designed a packed bed condenser using 1/2 inch Intalox saddles for the condensation of methylene chloride using water. Other studies concerned with packed beds include a study of a gasoline condenser (Hu, 1956) and condensation of isobutane on packed ice (Wiegandt, 1960).

Detailed information related to condensation on thin films have been discussed by Tamir and Rachmiev (1974) and Murty and Sastri (1974). The first deals with the condensation of an immiscible fluid on a thin laminar film of water flowing over a sphere while the latter applies to condensation on a falling film of coolant the same as the vapor. These papers offer a semi-theoretical rationale that could be applied to local heat transfer coefficients in packed beds.

ii. Drop-Type-Direct Contact Condensers

Although referred to as "drop-type-direct contact condensers" this classification includes all types of condensers where the vapor is the continuous media and the condensation takes place on either drops or jets of the dispersed phase. Typical of such devices are the so called barometric or low level condensers sometimes used in steam power plants and shown in

Figure III.44. Packed Bed Type Direct Contact Condenser

Figure III.45. These systems are variations on a simple spray tower as shown in Figure III.46. In attempting to obtain more compact exchangers, tray type condensers sometimes referred to as "jet type condensers" are also used. The tray type devices all enhance the heat transfer by allowing the coolant and condensate to mix on the trays and thus offer a higher heat transfer by eliminating temperature gradients within the drops and controlling drop coalescence as well as increasing the vapor holdup.

Most of the information available dealing with "drop" direct contact condensers is directed toward condensation of steam or the condensation of a vapor on its own liquid. Some "drop-jet" type configurations have been used for systems of immiscible liquids. However, comparisons between the packed bed performance and single tray performance showed lower approach temperatures with the tray arrangement (Harriott and Wiegandt, 1968). It should be noted that the basic models for condensation of one fluid on an immiscible drop of another, let alone sprays, have not as yet been formulated. For additional background information in this area the reader is directed to the complete review (Jacobs and Fannar, 1977).

iii. Bubble Type Direct Contact Condensers

Bubble type direct contact condensers include all those systems where the condensing vapor is injected directly into a continuous pool or stream of cold liquid. This type of device has been of interest as an open feed-water heater and as a vapor suppression system associated with nuclear reactors. Other applications are found in use in sea water distillation. The several types of designs, as shown in Figure III.47, differ basically in the behavior of the continuous phase.

Studies have been carried out for condensing organic vapors in a pool of water (Harris, 1970). In another study, pentane bubbles were

Figure III.45. Two designs of barometric condensers

Figure III.46. Simple spray tower

Figure III.47. Some bubble type direct contact condensers

condensed in pentane and compared to condensation in water (Sideman and Moalem, 1974). This latter study shows clearly the advantage of the use of a high conductivity fluid for the continuous phase.

Direct injection of vapor into liquid flowing turbulently in a tube is another effective condensation concept (Bankoff and Mason, 1962 and Grossman and Wyss, 1962). Heat transfer rates associated with condensing steam of nearly $400,000 \text{ BTU/hr ft}^3 \text{ }^\circ\text{F}$ were demonstrated.

2.b. Superheated Vapor

When the vapor to be condensed is superheated, the coolant must remove the sensible heat within the vapor as well as the heat of vaporization. For this type of duty there is a definite advantage to a counterflow system. The basic mechanisms of liquid film (packed bed), drop and bubble systems are all applicable; however, changes in design are necessary to insure complete or nearly complete condensation of the vapor and avoidance of loss of working fluid with any non-condensable gases that may be present. Typical equipment systems might be the counter-current designs discussed under previous sections.

The heat transfer in the superheat regime is governed by liquid-gas heat transfer relations. These are discussed by Fair (1972, 1972a), Sideman (1966) and Jacobs (1977) among others for a wide variety of equipment designs for all three mechanisms. The heat transfer coefficients attainable for removal of sensible heat are notably smaller than with changes of phase and are nominally controlled by the gas side heat transfer. Typically the heat transfer coefficients obtained for removal of sensible heat may vary from 100 to $2000 \text{ BTU/hr ft}^3 \text{ }^\circ\text{F}$. These values are several orders of magnitude below those obtainable for condensation. It can thus be seen that for even 5-10% of the total heat load superheat can

lead to significant increases in the size of the condenser-cooler.

3. Recommendations on Condenser Related Work

Basic heat transfer performance must be analyzed. While some information can readily be derived from experiments, an effort must also be made in developing theoretical models to help in optimizing systems and identifying possible stumbling blocks. Work will be proceeding on analytical models relative to packed bed condenser. Theories already developed for liquid film on a sphere will be extended to account for the condensation and liquid flow of a second fluid. A bench-scale condenser with built-in experimental versatility is currently being constructed.

The design of drop type condensers requires the development of a theoretical model for the static condensation of an immiscible fluid on a non-circulating liquid drop. This type of model is an extension of the work of Ford and Lekic (1973) who worked with steam and neglected the resistance of the forming immiscible layer. Work is progressing well here, and it should be completed in June of 1977.

The condensation or collapse of bubbles within a continuous liquid phase has been treated reasonably well by Sideman and co-workers, however, further experimental work is necessary to check his work with pentane vapor and water. Indeed the effects of condensing binary vapor has not yet been treated. However, results (Rumary, et al., 1970) indicate that volumetric heat transfer coefficients of at least $27,000 \text{ BTU/hr ft}^3 \text{ }^\circ\text{F}$ should be attainable. Thus, although not necessarily the optimum design, sufficient information is available to design a rather complete direct contact binary geothermal cycle. Work will proceed now on gaining sufficient information to optimize the cost and obtain comparison with other systems including the turbulent flow pipe condenser. Additional experimental data is being

accumulated by Mr. Heimir Fannar at the University of Strathclyde to accomplish these tasks.

The value of using a direct contact condenser for the removal of non-condensibles must be assessed. Comparisons based on efficiency and costs must be made to techniques used with closed condenser systems.

The effect of the use of a closed cooling tower must be assessed. Comparisons should be made to conventional cooling tower systems using conventional condensers as well as direct contact condensers.

CHAPTER IV

WORKING FLUID LOSS CONSIDERATIONS

IV.A. Working Fluid Solubility in Brine

A potential difficulty with direct contact heat exchangers is loss of working fluid in the spent brine. To assess the severity of the problem the equilibrium solubilities of working fluids in brine at conditions encountered in the heat exchangers must be known. Equilibrium solubilities are of interest for two reasons. Calculation of the fluid loss rate, assuming equilibrium solubility is reached, places an upper limit on the loss rate (in the absence of carryunder). This immediately allows economic assessment of the worst possible case for comparison with other heat exchanger systems. Also, knowledge of the equilibrium solubility must be available to calculate the mass transfer in the heat exchanger and determine a more accurate estimate of the actual fluid loss rate.

A literature search quickly revealed the lack of data available concerning the solubilities of candidate working fluids in aqueous solutions of varying salinities at temperatures of interest in geothermal applications. Therefore it was decided that measurements of equilibrium solubilities of n-pentane, isobutane, R-113 and R-114 in solutions of varying salinities and temperatures should be made by the University of Utah. Details of the measurements and results are presented in a report by Hellstrom, et al. (1976). Highlights of this report will be discussed in the paragraphs that follow.

Several methods of determining solubilities in aqueous solutions are available but use of a gas chromatograph was selected because of its high sensitivity and accuracy even in the low ppm range. A Hewlett Packard Model 3620A Research Chromatograph with a flame ionization detector was

used. Samples were prepared by mixing distilled water, the desired proportion of salt (NaCl) and an excess of the working fluid under investigation in a small pressure cell. The sample was stirred at the desired temperature (from 25-100°C) overnight to insure equilibrium was reached. The cell was then vented to low pressure to flash off excess liquid working fluid and allowed to reach a new equilibrium. Samples were drawn through a septum in the vessel and injected into the gas chromatograph column. The response of the instrument was calibrated to permit calculation of the mass of working fluid in the solution sample of known volume (about 5 μ l).

It was found that Henry's Law for dilute solutions applies to the results measured. Henry's Law states that the concentration (in ppm by mass) of solute gas in a dilute solution is directly proportional to the partial pressure (in atm) of the solute vapor above the liquid solution. This is valid for partial pressures up to the saturation pressure. For pressures higher than the vapor saturation pressure (when there are both liquid and vapor phases of the working fluid present) the concentration is equal to the value at the saturation pressure. In explicit form:

$$\begin{aligned} \text{Sol.} &= H \cdot P \quad \text{when} \quad P < P_{\text{Sat}} \\ \text{Sol.} &= H \cdot P_{\text{Sat}} \quad \text{when} \quad P \geq P_{\text{Sat}} \end{aligned} \tag{IV.1}$$

where

Sol. = solubility in ppm by mass

H = Henry's Law Constant

P = Partial pressure of vapor above the solution (atm)

P_{Sat} = Saturation pressure of solute fluid

Henry's Law constant is a strong function of the solution temperature and salinity as well as the solute fluid type.

Results of measurements of H are shown in Figures IV.1 - IV.4 for n -pentane, isobutane, R-113 and R-114 respectively. It can be seen that as temperature increases H decreases, and as salinity increases H decreases again. The dependence of solubility upon salinity is favorable to the use of direct contact heat exchangers with highly fouling brines. Brines with high total dissolved solids content which may foul shell-and-tube exchangers will result in low fluid loss in the non-fouling direct contact exchangers.

The solubilities shown in Figures IV.1 - IV.4 can be correlated by an equation of the form:

$$\ln(H) = A + 1000 B/T + C \ln(T/1000) + W(D + 1000E/T + F \ln(T/1000)) \quad (\text{IV.2})$$

where

H = Henry's Law constant

T = Temperature ($^{\circ}\text{K}$)

W = Weight percent NaCl

and A, B, C, D, E, F are constants to be determined for each fluid. Values of these constants for each of the four working fluids are given in Table IV.1. The curves calculated for each fluid are also shown in the figures.

TABLE IV.1
CONSTANTS USED IN EQUATION IV.2

Fluid	A	B	C	D	E	F
Pentane	2.6781	14.151	38.134	-0.055112	-0.20640	-0.54299
R-113	-0.82656	5.2801	8.8971	-0.12415	-0.073427	-0.26160
R-114	0.82009	6.2443	14.063	-0.076257	-0.22683	-0.61827
Isobutane	-0.61425	7.0329	15.946	0.095826	0.055289	0.29564

The Henry's Law constants of the four fluids are compared in Figure IV.5. The values are nearly identical for pentane and isobutane. But it must be kept in mind that the vapor pressure of isobutane is higher than

Figure IV.1. Solubility of pentane in aqueous and saline solutions

Figure IV.2. Solubility of isobutane in aqueous and saline solutions

Figure IV.3. Solubility of Refrigerant-113 in aqueous and saline solutions

Figure IV.4. Solubility of Refrigerant-114 in aqueous and saline solutions

Figure IV.5. Comparison of solubilities of four fluids in a 10% saline solution

that of pentane. Therefore, at a given temperature, the solubility of isobutane will be greater than that of pentane. It may also be greater than R-113 depending upon condensor temperature. R-114 appears to always show the highest solubilities at a given temperature of the four fluids even though H is greater for R-113.

The presence of noncondensable gases such as N_2 and CO_2 in the geothermal fluid can influence working fluid loss. The presence of these gases may reduce the solubility in the brine. The noncondensable gases will in large measure come out of solution in the brine as it enters the boiler. This is because the partial pressure of the gas over the brine will drop sharply as the brine comes in contact with the working fluid. The noncondensable gases will pass from the boiler to the condenser by way of the turbine. At this point the condenser will retain the gases and its pressure will continue to rise unless the noncondensibles are vented from the condenser. In the condenser the partial pressure of the working fluid will be high compared to that of the noncondensibles. Hence, as the noncondensibles are vented large amounts of working fluid vapor may also be vented. Thus, the condenser vent gases will have to be treated to prevent this loss.

As an example consider an isobutane cycle with a $40^\circ C$ condenser temperature and 0.5% (by mass) noncondensibles in the geothermal brine. In the condenser the partial pressure of the isobutane will be 77 psia. If the partial pressure of the N_2 is allowed to increase to 10 psia before

venting, the ratio of the mass of isobutane to the mass of N_2 vented will be approximately

$$\frac{\dot{m}_{isob}}{\dot{m}_{N_2}} = \frac{P_{isob}}{P_{N_2}} \frac{(\text{Mol. wt.})_{isob}}{(\text{Mol. wt.})_{N_2}} \approx 16$$

Thus isobutane will be vented with the noncondensibles at a rate equal to approximately eight percent of the total brine flow rate for 0.5% noncondensibles in the brine. This magnitude of fluid loss is clearly unacceptable and steps must be taken to greatly reduce the effect of noncondensibles.

The noncondensibles must be removed from the brine before it contacts the working fluid if working fluid losses are to be kept to a minimum. This can be done most simply by flashing the brine to a pressure just slightly below its saturation pressure and stripping the noncondensibles with the small amount of steam. This can reduce the level of noncondensibles to a very manageable point without a great sacrifice in cycle performance (Shaffer, C. J., 1976).

The noncondensibles remaining in the brine through a flashing operation will be flashed from the brine in the boiler and collect in the condenser. By refrigerating and compressing the gases vented from the condenser most of the working fluid can be condensed and recovered. As long as the total amount of noncondensibles in the condenser is small the parasitic load of compressors and refrigeration for the vented gases will be very small. As an example, the 50 MWe plant considered in detail in the economic analysis (in Chapter V) used partial flashing of the inlet brine and a refrigerated condenser vent to reduce the isobutane loss to 12 kg/hr in the vented gases. This compares with approximately a 5×10^5

kg/hr loss if no recovery techniques are attempted. The parasitic load of the recompression and refrigeration was the power requirements of a 6 KW compressor and 0.2 MW refrigeration unit and the flash expansion of the inlet brine lowered its temperature only approximately 1°C (Harris, J. S., et al., 1977).

IV.C. Recovery of Working Fluid From Spent Brine

In section IV-A the equilibrium solubilities of four possible working fluids in saline solutions were presented. In chapter V it will be shown that if working fluid loss is equal to that calculated from equilibrium solubility considerations, then operating costs will be significantly affected. It is important, therefore, to devise techniques for recovery of working fluid from the spent brine.

Two methods for fluid recovery have been investigated to date. The first involves sparging of the working fluid from the brine by bubbling a gas such as CO_2 through the brine. The working fluid will diffuse into the CO_2 in a mass transfer column. After the CO_2 leaves the column the working fluid can be removed from the CO_2 by condensation. The second method of fluid recovery involves expanding the spent brine to a low pressure and allowing the working fluid to flash off. Since the solubility of the fluid is proportional to its partial pressure above the brine, decreasing the pressure over the brine will be very effective at reducing the fluid loss. This method, however, may require additional pumping of the spent brine for reinjection into the ground.

Gas sparging has been investigated at the University of Utah using a simple column with sparging gas supplied from a CO_2 bottle. The concentration of the working fluid in the water is measured using a gas chromatograph in the same way the equilibrium solubility measurements reported earlier

in this chapter were made. Preliminary results indicate CO_2 sparging is an effective means of removing pentane from water.

Flashing the working fluid from the brine offers a simple and effective means of fluid recovery. One method that would require little additional equipment would be to expand the spent brine in a chamber exposed to the condenser pressure. This would drop the partial pressure of the working fluid over the spent brine to a value less than the fluid partial pressure in the condenser. If the flash chamber were kept at a pressure slightly greater than the condenser pressure the recovered working fluid vapor would pass to the condenser. Figure IV.6 shows the equilibrium solubilities of four working fluids in 10% NaCl solution when the brine is flashed to the pressure of a 30°C condenser. Figure IV.7 shows actual fluid loss for a net 50 MWe plant if the brine is expanded to the pressure of a 43°C (110°F) condenser. Clearly, pentane is the most desirable fluid because of its low Henry's constant and low vapor pressure.

The effectiveness of fluid recovery by flashing is seen by examining the isobutane cycle used for the economic analysis presented in the next chapter. Fluid losses may be reduced 75% by flashing the brine to the condenser pressure (assuming equilibrium once again). The cost for this flashing would be the pumping power required to boost the brine pressure high enough for reinjection. Using results presented in the next chapter this could result in a 10% savings in the total power cost. If the brine must be boosted to 300 psia for reinjection there would be approximately a 4-5% decrease in net power output from the plant. The 5% (approximately) net savings in conjunction with the mitigation of possible environmental effects of isobutane release from the plant present a clear case for use of recovery techniques.

Figure IV.6. Relative solubility in a 10% NaCl brine flashed to pressure of a 30°C condenser

Figure IV.7. Working fluid loss rates in thermodynamic optimum 50 MWe cycles when brine is flashed to the pressure of a 43°C condenser

CHAPTER V

ECONOMIC ANALYSIS

The economic analysis of a geothermal power plant must take place in several stages. These stages are the preliminary estimate, the budget estimate and the tender estimate. The first such estimate is generally good to about $\pm 25\%$, the second estimate to $\pm 10\%$ and the tender estimate to within $\pm 5\%$. The later two estimates require detail costing of equipment based on plant layout and actual component designs. The first estimate quite often uses figures based on prior experience with other similar plants and costing of key items. Such costing is referred to as factorial estimating. This is the type of costing carried out by Milora and Tuter (35), Green and Pines (15), Klei and Maslan (67), Battele Northwest (68), Ben Holg Company (8,53), etc.

Factorial estimating generally starts with a flow or process sheet on which major items of equipment are listed, together with sizes and weights. Each item is costed using established data or by obtaining preliminary contractors' quotations for those items of significant cost. Once the cost for each item has been evaluated, this cost is expanded by factors derived from similar plants to provide for piping, insulation, instrumentation, civil and structural work, etc. Similarly, operation and maintenance costs are obtained based on some multipliers of, perhaps, product output. In the cost of electrical power plants this would be MW_e production. Taxes, etc., would be based on total plant costs. The method of factorial estimating discussed above has been and is being incorporated into the economic analyses of Geothermal Power Plants in our computer program DIRGEO. References 35, 15, 68, 8, 53, 61, 38, and 6 have been reviewed in order to obtain the best estimates for both multiplication factors and primary item costing. In order to make comparisons, generally the procedures of References 35, 15 and 68 have been adopted.

There have been a number of discussions between Lawrence Berkeley Labs, ERDA-Idaho Falls, D.G.G. Idaho Falls and the University of Utah on system analyses. Due to these discussions a consensus has been reached on thermodynamic analyses. Thus, by closely adhering to the economic procedures reasonable comparison should be possible on system and power costs. However, it is important to document these costing procedures. Thus, details as to costing utilized at the University of Utah are outlined below. The major pieces of equipment for binary and direct contact binary cycles are the pumps, turbines, condensers, heat exchangers, separators, storage tanks and fluid recovery systems. In addition other major cost items are the hot brine wells and reinjection wells. These then form the basis for cost estimating. A cost of operation not included for binary systems is working fluid makeup, yet this must be included as part of the direct contact binary cycle. Thus, it is an add on item.

V.A. Vapor Turbines

The method for establishing costs of vapor turbines is that due to Nichols (38). This method is used by References 35, 15 and 68 in their analyses of binary cycles for application to geothermal resources. It was incorporated early into the program DIRGEO by Bliss (6). The basic cost equation in 1974* dollars is

$$\phi_T (\$) = n_e [1 - 0.04 (n_e - 1) f_2] \left[30135 n_s f_1 D_p^{2.1*} + 16775 D_p^3 + 20720 D_p^2 \right] + 225000 \left(\frac{MW_e}{10} \right)^{0.7} \quad (V-1)$$

where n_e is the number of exhaust ends, $D_p N$ is the blade tip speed, n_s is

*To obtain 1976 dollars multiply by 1.12

the number of stages, f_2 is a function of maximum operating pressure and f_1 is a function of tip speed. f_2 is equal to one for pressures up to 35 atmospheres and increases linearly to 1.5 at 200 atmospheres. f_1 is unity for tip speeds below 300 m/sec (1000 fps) but increases exponentially to a value of 3 at 600 m/sec. Discussion of turbine costing and optimum sizing for various working fluids in direct contact cycles and binary cycles has been treated by Bliss (6) in a M.S. Thesis developed under Grant No. AER-75-01039.

V-B Pumps

Pump costs have been correlated by Tester and Milora (35) from manufacturers estimates for multistage, centrifugal, high pressure, boiler type feed pumps. The costs were correlated using power output as the independent parameter, as suggested by References () and (), based on the estimates of five manufacturers. The cost equations were differentiated only for pressures under 1000 psia (6895KPa) and over 2000 psia (13,790Kpa) and thus intermediate pressures would have to be determined by linear interpolation. For direct contact systems only the low pressure costs are of real interest as the cost of pumping the brine would fast become prohibitive

$$\phi_{p_{p>1000 \text{ psia}}} (\$) = 3.5 \left(\frac{\text{HP}}{1000} \right)^{.9} \times 10^4 \quad (\text{V-2})$$

$$\phi_{p_{p>2000 \text{ psia}}} (\$) = 5.0 \left(\frac{\text{HP}}{1000} \right)^{.9} \times 10^4 \quad (\text{V-3})$$

These costs are in 1976 dollars.

V-C Conventional Heat Exchangers

Conventional heat exchangers whether used as a boiler, liquid-liquid, preheater, condenser or liquid-gas heater must be expressed as functions of tube

and shell side pressures and required heat transfer area. Typical costs as estimated from ten different sources are given by Testor and Milora (35) in Figures V-1 and V-2 based on unit sizes exceeding 20,000 ft² (1858 m²). Other sources such as Reference (53) ignore the effect of pressure on the heat exchanger costs and thus unfairly penalize low pressure fluids in favor of high pressure systems. As this may be a major factor in selecting a "best" working fluid such analyses are given less credence.

V-D Direct Contact Heat Exchangers and Pertinent Vessels

Direct contact heat exchangers being in general non-standard equipment items require designs to be submitted to vendors in order to ascertain close cost estimates; however, estimates are available on many Chemical Engineering items for preliminary costing (69,70). In general, the cost estimates are obtained from Reference (70) with a factor of 2.78 applied to account for price changes since 1969. Current exchange rates of \$1.75/L are then applied. Costs quoted are based on price per ton and assume mild steel construction. This form of costing thus accounts for pressure effects by taking into account the additional wall thickness necessary for high pressure systems.

Low Duty Vessels \$1320/ton (V-4)

Process Vessels \$1610/ton (V-5*)

The above vessels would have nominally standard numbers of inlets and two manholes constructed to appropriate standards. Weight of simple internals, vessel skirts and reinforcements may be included in the total weight. Thus baffles may be included but not perforated trays or injection nozzles.

*This method of calculating cost has been checked against a 22 ft long (6.71 m) 42 inch (1.07 m) diameter vessel with one flanged end for 1500 psi (10,340Kpa) duty and several openings also flanged. It predicted 95% of cost.

Figure V-2. Cost of Induced Draft - Air Cooled Heat Exchangers.

Figure V-1. Tube and Shell Heat Exchangers.

Typical erection costs for mild steel vessels are 10% of the cost of the vessel. Figures 6, 7, 8 and 9 from Reference 68 give alternate means for estimating vessel costs. Figure 6 may be used for heat exchangers with little additional internals but require the additions for nozzles, trays, etc. Figure 7 and 8 may be used for storage tanks. Figure 9 provides means for estimating internals costs such as perforated trays and could also be used for simple nozzles if one tripled the costs. For computation in 1976 dollars the multiplication factor of 4.87 must be applied to these figures. Simple formulas in 1976 dollars assuming mild steel are

$$\text{For trays } \phi_{\text{trays}}(\$) = 1266 \frac{\text{Tray Area ft}^2}{100 \text{ ft}^2}^{.95} \quad (\text{V-6})$$

$$\text{For storage vessels } \phi_{\text{sv}@p=5.5 \text{ kg/cm}^2} = 34,100 \frac{\text{Volume M}^3}{100 \text{ M}^3}^{.93} \quad (\text{V-7})$$

however, it is recommended that equations V-4, V-5 and V-6 be used.

Compressor costs are estimated using pump costs at this time; however vendor costs are being obtained. Costs of settlers and separators as well as heat exchangers use the equations V-3, V-4 and V-5.

V-E Wells

Well costs have been treated by a number of investigators; however we cite only 35, 67 and 68. These references indicate that the cost of the wells is strongly dependent in its drilling and casing cost on depth and rock hardness. Typical costs are given in Figure V-17 of Reference 35. The lifetime of the well is another factor in costs. Reference 67 indicates

Figure 6. Approximate Cost of Carbon Steel Vessels, Ref. 68.

Figure 7. Carbon Steel Spherical Storage Tanks Approximate Installed Cost, Ref. 68.

Figure 8. Carbon Steel Storage Tanks Approximate Installed Cost (Atmospheric Pressure), Ref 68.

Figure 9. Approximate Costs of Distillation Trays, Ref.68.

that a lifetime of twenty years would be reasonable for vapor dominated resources although they have lasted from 5 to 30 years. For liquid dominated resources data to date 67 does not indicate decay in performance. Thus, a true picture of costs appears to be lacking. In the estimates of Reference 35 twenty years was thus arbitrarily chosen to predict well lifetime. As pointed out in the introduction to this section it was chosen to follow Reference 35 recommendations. It was chosen also to assume an equal number of reinjection wells as in Reference 35. Following 35 it is being assumed in all our future cost considerations that all wells are self pumping aquifer of 2500 m depth with a maximum flow of 45 kg/sec (100 lb/sec) at a drilling and casing cost of \$200/meter. This is \$500,000 per well. All wells are assumed to have a twenty year lifetime. The wells are assumed to provide brine at saturated liquid conditions.

V-F Piping Costs and Auxiliary Equipment

Piping costs have been estimated by Reference 68 as a simple formula based upon plant total power. However, the operating pressure will influence the cost of this item and also the auxiliary equipment. Thus Reference 68 model was not selected in approximating these costs. The model used by Reference 35 follows better the factorial estimating method and should account better for the effects of pressure by directing costs directly to major equipment costs. Thus the equation

$$\phi_{F.C.}(\$) = \left[\phi_E(\$) + \phi_E(\$) f_i \right] f_I + \left[f_I^* \phi_w(\$) + f_w \phi_w(\$) \right] \quad (V-8)$$

where ϕ_{FC} is the total fixed capital investment, ϕ_E is the major equipment costs ϕ_w is the well cost. The fractions f_i of ϕ_E covers the cost of piping, buildings, structures, instrumentation, equipment installation, etc., and f_I is a factor that accounts for indirect expenses such as engineering fees, contingency, escalation during construction and related costs. The fraction f_w of ϕ_w covers the cost of piping from the wellhead to the power plant and can represent 15 to 51% of the total well costs. f_I^* includes indirect expenses associated with discovery of the geothermal field, such as land acquisition, drilling exploratory wells, surface exploration and contingencies. For non-flammable systems Reference 35 gives $1 + f_i = 1.63$ and for flammable systems 1.65. For f_w in liquid dominated systems the value selected depends upon the number of wells according to Table V-1.

Table V-1

	f_w
$n_{\text{wells}} \leq 6$	.16
$6 < n_{\text{wells}} \leq 18$	.24
$18 \leq n_{\text{wells}} \leq 36$	.34
$36 < n_{\text{wells}} \leq 60$	.44
$60 < n_{\text{wells}} \leq 90$	.49
$90 < n_{\text{wells}}$	.50

These numbers are based on pressures of 140 psia (965 KPa). It is suggested that they be multiplied by $p_{\text{well}}/140$ in psia for use with liquid dominated systems.

For f_I the value proposed is 1.70 and for f_I^* the value is 1.56. Thus the equation used for fixed capital investment is

$$\phi = 2.77 \phi_E + 1.56 (1+f_w) \phi_w \quad (V-9)$$

for non-flammable fluids, and

$$\phi = 2.81 \phi_E + 1.56 (1+f_w) \phi_w \quad (V-10)$$

for flammable fluids.

V-G Operating and Maintenance

Operating costs have been established by various manufacturers from data provided by power companies. Annual plant operating cost is given by

$$\phi_{\text{operating yr.}} (\$) = 1455 (\text{Plant Size in MW})^{0.9} \quad (V-11)$$

The maintenance cost per year is added as a f_n of plant cost

$$\phi_{\text{maintenance/yr.}} (\$) = .004 (\text{Plant Capital Cost}) \quad (V-12)$$

Plant capital equipment are amortized linearly on a twenty year life as is well replacement; however it is assumed that a stand by of 20% is provided.

Property tax and insurance are calculated as a f_n of plant cost.

$$\phi_{\text{Prop Tax/yr}} \$ = 0.025 (\text{Plant Cost}) \quad (V-13)$$

$$\phi_{\text{Prop Ins/yr}} \$ = 0.0012 (\text{Plant Cost}) \quad (V-14)$$

Working fluid cost is as provided in current dollars on tank car prices. These rates are provided by DuPont for refrigerants and Phillips Petroleum Co. for hydrocarbons.

All of the above costs are consistent with the majority of references. These cost techniques will be applied to all future system studies and are being programmed.

A.H. Conceptual Design and Evaluation of a 50 MWe Plant

The engineering and construction firm of Ford, Bacon and Davis Utah, Inc. was commissioned by EG&G Idaho to perform preliminary design and cost estimates for both a direct contact heat exchanger (DCHE) binary cycle power plant and a conventional shell-and-tube heat exchanger (STHE) binary cycle plant. The University of Utah provided all details of the power cycle parameters and other technical design information. The results of the study, which are presented in detail in a final report (Harris, et al., 1977), will be summarized in this chapter.

Isobutane was chosen as the working fluid for the binary cycles. However, a cursory comparison was made between isobutane and pentane that showed pentane to be the better fluid for use with a 149°C (300°F) geothermal source.

The University of Utah computer program DIRGEO was used to select the thermodynamic optimum cycle conditions for both the isobutane and pentane cycles. Information extrapolated from large-scale tests of surface-type direct contact boilers was used to size the boiler for the 50 MWe plant. Spray column preheaters were sized using the relatively extensive information available in the chemical engineering literature. Conventional shell and tube condensers were used in both the DCHE and STHE cycles.

The flowsheet for the DCHE cycle analyzed by Ford, Bacon and Davis Utah is shown in Figure V.1. Major equipment included brine deaerators, boilers, turbine-generator, condenser, cooling water system, noncondensable venting and working fluid recovery, preheaters and brine and pentane booster pumps.

COOLING TOWER

DUTY: 1552, X 10⁶ BTU/HR
FAN MOTORS: 2100 HP

STEAM NO.	(8)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)	(20)	(21)	(22)	(23)	(24)	(25)
TEMPERATURE °F	106	137	104	106	110	93	93	35	35	35	106	74	74	100	—	—
PRESSURE PSIA	335	85	85	90	85	75	200	75	200	75	65	15	70	—	—	—
GEO-SERIAL NO. #/HR	—	108 X 10 ⁴	108 X 10 ⁴	—	—	—	—	—	—	—	—	108 X 10 ⁴	—	—	—	—
ISOBT-TIME #/HR	9.98 X 10 ⁴	9.98 X 10 ⁴	9.98 X 10 ⁴	9.98 X 10 ⁴	21.90	85	25	210.5	60	525	26.90	2	—	—	—	—
MOV. COND. CABLES #/HR	—	125	125	—	125	125	125	—	—	—	—	—	—	—	—	—
TOTAL #/HR	9.98 X 10 ⁴	10.08 X 10 ⁴	10.08 X 10 ⁴	9.98 X 10 ⁴	2.315	210	150	210.5	60	525	26.90	108 X 10 ⁴	59.7 X 10 ⁴	59.7 X 10 ⁴	59.7 X 10 ⁴	1.55 X 10 ⁴
GRM. # TEMP. & PRESSURE	37.700	645.118 X 10	37.700	57.700	645	645	645	7.05	0.2	1.9	9.6	216	119,400	119,400	119,400	3,700

[illegible]

Figure V.1. Continued

The capital cost estimate was obtained using the method presented by Guthrie (Guthrie, 1969). He presents factors that can be used to estimate direct material and labor costs for installation as well as indirect costs based upon major equipment costs. Busbar power costs were obtained assuming:

Annual fixed cost, 17% of capital investment.

Operating factor, 80%

Annual maintenance expense, 2 1/2% of capital investment

Operating supplies, 5% of maintenance expense.

Isobutane makeup, cost of isobutane \$0.10/lb.

Fuel cost (brine supply and reinjection) 13.5 mills/KwH.

Capital and power costs are shown as a function of plant generating capacity in Figure V.2. The net power cost of a net 50 MWe DCHE plant is 49.5 mills/KwH. Fixed costs, operating and maintenance cost, and "fuel" costs are 24.6, 11.4, and 13.5 mills/KwH respectively. The isobutane loss alone accounts for 6.9 mills/KwH or over half the operating and maintenance costs. This is a very conservative figure because it assumes equilibrium solubility and no attempt at fluid recovery from the spent brine. It is anticipated that simple recovery techniques could reduce the isobutane makeup cost by at least one half.

The ratio of installed cost to major equipment cost was 2.67 for the Ford, Bacon and Davis Utah study. This compares closely with the value 2.82 used by Milora and Tester (Milora and Tester, 1976) but is low compared with the value 4.95 recently recommended by Starling (Starling, 1976). The major differences appear in the allowance for installation and piping and insulation costs.

Figure V.2. Net power and capital costs vs. DCHE plant size

A breakdown of the major direct capital costs is shown in Table V.1 for both a DCHE plant and STHE plant. Notice the condensers and cooling water systems comprise over half the major equipment direct costs. Obviously this points to an area where methods for significant savings should be examined. The lower capital cost of direct contact condensing is certainly worth further investigation.

V.1. Comparison of DCHE and STHE Plants

The DCHE plant capital costs are less than those for a STHE plant of the same net output. This can be seen by examining Table V.1. But the isobutane makeup cost causes the operating cost to be higher for the DCHE plant. The net result is that the power costs for the two plants are equal. Any significant reduction in isobutane loss would make the DCHE total power cost lower than the STHE power cost.

V.1 Comparison of Isobutane and Pentane DCHE Cycles.

Ford, Bacon and Davis Utah looked briefly at a comparison of the total power costs of DCHE cycles using pentane and isobutane as working fluids. At high source temperatures isobutane has superior thermodynamic performance and is probably the best working fluid of the two. But at moderate and low source temperatures pentane cycles are as efficient as isobutane cycles from a thermodynamic point of view. And there are several reasons why a plant using pentane is less expensive. The lower operating pressure of a pentane cycle means lower capital costs, the lower fluid loss rate of a pentane cycle reduces the operating cost, and a pentane cycle requires a smaller parasitic pumping load. The result is an estimated savings of 1.3 mills/KwH in fixed costs, 0.20 mills/KwH in maintenance and supplies, and 0.50 mills/KwH in working fluid makeup. In addition, the lower vapor pressure of pentane makes it more amenable to fluid recovery techniques and

TABLE V.1.
DIRECT CAPITAL COSTS

	<u>DCHE Plant Thousands \$</u>	<u>STHE Plant Thousands \$</u>
Brine pumping system		
Fuel	1,320	880
Booster	557	-
Re-injection	-	952
Isobutane pumping system	2,072	2,745
Deaerator system	372	-
Boilers	1,595	4,931
Preheaters	1,916	3,717
Condensers	10,994	12,207
Refrigeration	104	-
Condensate receivers	613	649
Cooling water system	5,466	5,452
Turbine-Generator	<u>6,427</u>	<u>6,660</u>
Subtotal Direct Material and Labor	31,436	38,193

thus promises an even greater savings in organic fluid makeup when simple recovery techniques are employed.

In conclusion, though the isobutane DCHE and STHE plants have the same total power costs, a pentane DCHE plant will have a lower power cost for the same size plant.

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APPENDIX
SUMMARY OF THE LARGE-SCALE DIRECT
CONTACT BOILER TEST RESULTS

NOMENCLATURE FOR DATA TABULATION

TEMPERATURES

H2O in: brine temperature in (to boiler)

H2O ex: brine temperature out

F in: freon temperature in

F out: freon temperature out

F sat: freon saturation temperature

PRESSURES

Vessel: total vessel pressure, psia

Sat: freon saturation pressure

HEAT TRANSFER

QD: heat transferred to dispersed phase (freon)

UA: overall heat transfer coefficient

MASS RATIO

Mass flow rate brine

Mass flow rate working fluid

CU

Carryunder indicator

0: no carryunder, safe operation

2: $T_{R-113} - T_{R-113 SAT} < 0$

1: detected carryunder

Note: all 1's also satisfy condition 2

STANTON NUMBER

$H = PR*JA$

$H = (\text{Prandtl No.}) \times (\text{Jacob No.})$

FREON NOZZLE AEW

DATE	TEMPERATURE (F)					PRESSURES (PSIA)		HEAT TRANSFER		MASS FLOW	CU	STANTON	H=PR*JA
	H2O IN	H2O EX	F IN	F OUT	F SAT	VESSEL	SAT	QD	UA	H2O/FLUID		NUMBER	
								BTU/HR	BTU/HR F				
400	291.3	262.1	184.6	247.1	212.0	92.5	64.2	43678.	736.	4.38	0	4.56	3.50
	292.5	271.2	185.2	257.8	204.9	92.5	58.4	44936.	795.	6.57	0	4.93	3.11
	292.4	275.8	183.9	262.4	201.3	92.5	55.6	45734.	826.	8.76	0	5.13	2.90
	290.5	253.6	197.5	237.1	217.4	92.5	68.8	58552.	1069.	3.02	0	4.53	4.15
	286.6	235.7	193.6	221.5	224.0	92.5	74.8	44282.	838.	1.64	1	4.57	5.93
401	293.7	265.7	208.2	247.1	212.0	92.5	64.2	62721.	1209.	4.17	0	4.72	3.43
	293.2	270.8	188.0	254.9	207.0	92.5	60.0	71021.	1230.	5.44	0	4.73	3.17
	291.8	267.5	194.0	248.4	211.2	92.5	63.5	83109.	1454.	4.46	0	4.57	3.41
	293.7	263.2	197.8	244.9	213.3	92.5	65.2	75009.	1323.	3.61	0	4.48	3.55
	291.6	251.4	205.2	236.5	217.7	92.5	69.0	69026.	1366.	2.48	0	4.74	4.21
	291.4	243.6	203.5	231.4	220.0	92.5	71.1	84432.	1709.	2.01	0	4.80	4.73
	291.2	259.2	203.5	239.7	216.1	92.5	67.7	86652.	1617.	3.01	0	4.54	3.86
	292.7	266.3	205.0	244.5	213.5	92.5	65.4	87455.	1603.	4.01	0	4.51	3.50
	294.4	267.8	206.8	247.1	212.0	92.5	64.2	87526.	1624.	4.01	0	4.57	3.36
402	294.5	263.8	199.8	242.8	214.5	92.5	66.2	98770.	1714.	3.61	0	4.34	3.55
	294.0	256.9	204.0	238.8	216.6	92.5	68.1	92990.	1721.	2.79	0	4.49	3.97
	292.1	242.7	208.2	230.1	220.6	92.5	71.7	88364.	1883.	1.87	0	4.92	4.78
	293.0	239.3	211.1	226.2	222.2	92.5	73.1	96003.	2144.	1.68	0	5.02	5.06
	293.7	254.2	210.5	236.6	217.7	92.5	69.0	99630.	1988.	2.52	0	4.66	4.02
	293.7	264.2	201.2	242.5	214.6	92.5	66.4	105510.	1854.	3.35	0	4.36	3.57
	294.4	261.2	200.8	240.0	216.0	92.5	67.5	117610.	2050.	2.99	0	4.30	3.60
	293.0	252.6	206.6	234.6	218.6	92.5	69.9	113320.	2182.	2.23	0	4.55	4.17
	293.6	237.9	213.1	226.2	222.2	92.5	73.1	106690.	2505.	1.50	1	5.22	5.11
403	293.9	267.7	211.3	248.1	211.4	92.5	63.7	66463.	1304.	4.33	0	4.75	3.36
	294.8	268.8	197.3	251.4	209.4	92.5	61.9	70990.	1261.	4.33	0	4.62	3.25
404	290.5	257.7	196.3	243.5	214.0	92.4	65.8	43326.	804.	3.19	0	4.73	3.83
	290.0	257.5	199.7	242.3	214.7	92.4	66.4	42591.	810.	3.27	0	4.75	3.80
	289.2	252.3	187.9	240.5	215.6	92.4	67.2	51980.	926.	2.73	0	4.65	4.15
405	293.6	275.3	183.6	262.3	201.3	92.4	55.6	43233.	769.	7.72	0	5.06	2.93
	295.3	279.7	174.9	268.4	195.9	92.4	51.7	40691.	710.	10.73	0	5.23	2.72
	291.8	270.5	185.3	258.4	204.3	92.4	57.9	37948.	685.	6.71	0	5.04	3.13
	291.4	264.5	193.6	250.5	209.8	92.4	62.3	36395.	668.	5.14	0	4.83	3.44
406	294.9	276.2	186.2	262.3	201.2	92.4	55.6	53937.	954.	7.36	0	4.98	2.89
	293.1	273.1	183.6	259.2	203.6	92.4	57.4	50091.	874.	6.61	0	4.92	3.03
	292.3	268.8	194.9	254.0	207.4	92.4	60.4	47165.	871.	5.31	0	4.89	3.24
	291.4	261.8	210.1	247.9	211.4	92.4	63.6	43315.	912.	4.01	0	5.12	3.57
	294.9	272.7	189.2	259.2	203.6	92.4	57.4	65841.	1171.	6.16	0	4.90	3.01
	292.7	266.6	189.7	251.8	208.9	92.4	61.6	63500.	1114.	5.00	0	4.72	3.33
	293.1	264.5	198.4	247.9	211.4	92.4	63.6	59427.	1081.	4.08	0	4.66	3.46
407	295.7	277.5	157.9	265.3	198.8	92.4	53.7	58791.	903.	8.71	0	4.84	2.80
	295.7	277.9	185.8	265.7	198.4	92.4	53.4	53726.	970.	8.70	0	5.15	2.78
	295.3	278.8	181.4	266.2	198.0	92.4	53.2	54523.	964.	8.72	0	5.13	2.77
	294.9	277.9	172.7	266.2	198.0	92.4	53.2	55214.	937.	8.87	0	5.09	2.79
	293.6	274.5	172.1	261.8	201.7	92.4	55.9	55345.	918.	7.52	0	4.92	2.95

	295.3	274.9	162.7	262.7	201.0	92.4	55.4	57417.	892.	7.48	0	4.78	2.90
	294.2	275.3	185.0	263.1	200.6	92.4	55.1	53410.	858.	7.46	0	5.08	2.80
	294.4	271.8	189.2	250.6	205.7	92.4	59.0	51538.	899.	5.62	0	4.77	3.08
	294.9	271.4	181.4	250.6	205.7	92.4	59.0	52126.	862.	5.71	0	4.66	3.08
	294.4	270.1	168.8	250.6	205.7	92.4	59.0	54269.	842.	5.73	0	4.59	3.12
	292.3	261.0	188.4	245.7	212.8	92.4	64.8	49268.	841.	3.82	0	4.52	3.63
	292.3	261.8	198.4	245.7	212.8	92.4	64.8	47726.	875.	3.79	0	4.66	3.60
	291.0	260.5	180.5	245.3	213.0	92.4	65.0	50622.	827.	3.82	0	4.46	3.60
	290.2	274.5	205.3	254.9	206.9	92.4	60.0	77577.	1436.	5.43	0	4.71	3.03
	290.6	275.3	213.6	256.2	206.0	92.4	59.3	75071.	1491.	5.45	0	4.89	2.98
	295.7	274.5	193.1	250.6	205.7	92.4	59.0	81371.	1411.	5.46	0	4.67	3.01
408													
	294.4	270.5	177.5	255.3	206.6	92.4	59.7	76001.	1221.	5.28	0	4.58	3.14
	294.0	271.0	188.8	256.2	206.0	92.4	59.3	72020.	1260.	5.37	0	4.79	3.12
	295.3	271.0	170.5	257.1	205.4	92.4	58.8	77157.	1198.	5.35	0	4.57	3.08
409													
	240.1	227.1	132.8	211.4	165.0	47.4	32.9	45603.	827.	9.58	0	5.65	4.00
	249.2	209.7	130.6	197.5	173.9	48.4	37.5	43224.	671.	2.90	0	4.70	4.80
	251.4	237.5	155.3	223.1	184.9	62.4	44.2	42198.	835.	8.56	0	5.76	4.37
	240.6	227.9	151.0	214.0	189.4	62.4	47.1	41275.	799.	6.22	0	5.59	5.34
	244.7	228.8	177.5	217.9	212.9	81.4	64.9	37506.	921.	4.55	0	6.32	8.81
	251.0	225.3	175.8	215.7	214.9	82.4	66.6	37523.	894.	3.78	0	6.13	9.88
	248.4	217.9	172.3	211.4	216.4	82.4	67.9	38032.	924.	2.94	2	6.34	13.63
410													
	254.9	238.4	182.7	225.3	211.1	82.4	63.4	37602.	912.	6.18	0	6.24	6.57
	249.2	230.1	178.4	219.2	213.6	82.4	65.5	38013.	953.	6.12	0	6.45	8.86
	253.1	229.7	181.8	220.1	213.3	82.4	65.2	37616.	941.	4.89	0	6.35	8.22
	251.0	225.7	176.6	214.9	215.2	82.4	66.9	37363.	884.	3.78	2	6.05	9.00
411													
	289.1	248.1	72.0	231.8	208.2	82.5	61.0	63160.	597.	3.29	0	3.63	3.91
	288.2	234.8	73.9	220.3	214.4	83.5	66.2	92603.	859.	2.18	0	3.45	4.80
	288.5	226.4	76.7	215.6	215.0	82.5	66.7	114510.	1073.	1.77	0	3.42	5.41
	288.3	221.1	82.2	215.0	215.2	82.5	66.9	147170.	1434.	1.36	1	3.50	5.81
412													
	286.9	231.1	69.5	218.0	214.1	82.4	65.9	80559.	741.	2.09	0	3.45	5.13
	287.0	227.0	72.1	217.6	214.2	82.4	66.0	87692.	823.	1.88	0	3.49	5.38
	286.8	222.8	72.0	216.1	214.8	82.4	66.5	104460.	988.	1.61	0	3.50	5.74
	286.2	225.0	71.8	217.3	214.4	82.4	66.1	113450.	1075.	1.48	1	3.52	5.58
413													
	286.5	233.6	65.6	220.5	213.1	82.4	65.0	81281.	745.	2.28	0	3.49	4.93
	286.0	224.3	71.6	217.0	214.4	82.4	66.2	104790.	995.	1.72	0	3.53	5.65
	286.2	223.3	77.0	214.9	215.2	82.4	66.8	126290.	1229.	1.38	1	3.49	5.70
414													
	287.6	242.5	63.3	226.4	222.2	92.5	73.1	63229.	576.	2.90	0	3.51	5.18
	288.1	235.3	67.5	225.2	222.6	92.5	73.5	70390.	733.	2.31	0	3.55	5.68
	287.9	235.5	69.4	225.3	222.6	92.5	73.5	85196.	803.	2.11	0	3.56	5.67
	287.3	235.8	69.8	224.8	222.8	92.5	73.6	99499.	939.	1.81	1	3.56	5.72
	280.1	246.5	77.2	235.1	229.1	102.5	79.7	53436.	542.	3.00	0	3.80	5.70
	292.8	247.7	76.0	235.8	228.8	102.5	79.4	103630.	996.	2.79	0	3.62	5.20
	291.8	243.1	83.6	232.4	230.2	102.5	80.8	110060.	1086.	2.34	0	3.60	5.75
	290.3	242.0	85.8	231.4	230.6	102.5	81.2	107640.	1079.	2.05	0	3.63	6.02
	287.7	237.1	87.4	232.8	230.0	102.5	80.6	107380.	1136.	1.85	1	3.82	6.62
415													
	289.4	227.0	63.0	217.3	202.0	72.5	56.2	91116.	817.	2.00	0	3.45	4.32
	289.3	218.6	69.1	211.8	204.2	72.5	57.9	113470.	1035.	1.60	0	3.41	4.83
	289.4	216.9	74.6	204.6	206.7	72.5	59.8	141420.	1273.	1.25	1	3.27	5.12
	285.4	215.4	80.1	206.5	206.1	72.5	59.3	123030.	1176.	1.42	1	3.42	5.38

FREC SPRAY BOILER, NOZZLE SPW

DATE	TEMPERATURE (F)					PRESSURES (PSIA)		HEAT TRANSFER		MASS RATIO	CU	STANTON	H=PR*JA	
	H2O IN	H2O EX	F IN	F OUT	F SAT	VESSEL	SAT	GD BTU/HR	UA BTU/HR F	H2O/FLUID		NUMBER		
200	264.5	253.6	171.0	224.5	181.0	60.4	41.7	46131.	785.	10.56	0	4.69	3.39	
	277.5	264.9	170.5	251.4	211.7	94.4	63.9	28910.	544.	11.03	0	5.44	3.01	
	280.5	264.9	135.0	243.1	171.2	62.4	36.0	49241.	663.	11.24	0	4.46	2.70	
	278.4	264.5	165.3	251.0	203.3	91.4	61.1	29419.	527.	11.23	0	5.29	3.74	
201	281.4	267.9	171.0	251.4	197.7	83.5	52.9	40018.	701.	10.69	0	5.11	3.21	
	278.8	259.7	157.5	243.6	201.6	92.5	55.9	42143.	670.	6.67	0	4.77	3.50	
	274.0	237.5	152.3	224.0	211.7	82.5	63.9	40109.	608.	3.39	0	4.38	5.20	
	270.1	224.9	156.6	212.7	214.9	81.5	66.6	39620.	632.	2.49	2	4.43	7.04	
	267.1	214.0	169.2	214.9	215.3	82.5	66.9	37940.	784.	1.82	2	5.45	9.07	
	274.5	233.6	132.4	215.7	193.1	65.5	49.7	49391.	633.	3.09	0	3.97	4.13	
202	274.0	220.1	155.3	211.4	214.1	80.5	65.9	42781.	672.	2.76	2	4.37	6.99	
	269.7	215.7	155.3	211.4	214.1	80.5	65.9	43356.	731.	1.67	1	4.69	8.06	
203	282.5	216.4	69.6	210.7	204.6	72.5	58.2	70400.	671.	1.66	0	3.55	5.35	
	283.0	213.9	69.9	206.9	206.0	72.5	59.2	78213.	735.	1.48	0	3.46	5.62	
	283.7	214.7	70.7	206.1	205.0	71.5	58.4	82143.	765.	1.40	1	3.42	5.42	
204	283.3	225.7	74.5	212.1	204.2	72.6	57.8	57519.	541.	2.02	0	3.46	4.78	
	283.3	217.5	78.4	207.9	205.7	72.6	59.0	68665.	660.	1.68	0	3.46	5.34	
	283.6	215.2	80.5	206.0	206.3	72.6	59.5	80804.	780.	1.41	1	3.44	5.53	

FIRED NOZZLE, FLAT SPRAY

DATE	H2O IN	H2O EX	TEMPERATURE (F) F IN F OUT	F SAT	PRESSURES (PSIA) VESSEL SAT	HEAT TRANSFER QD BTU/HR	UA BTU/HR F	MASS RATIO H2O/FLUID	CU	STANTON NUMBER	H=PR+JA
100	271.3	235.2	85.3 212.6	213.8	80.5 65.6	45993.	471.	3.22	2	3.63	5.80
	279.4	256.7	87.0 229.6	209.9	83.0 62.4	52674.	540.	5.84	0	3.74	4.03
	277.4	253.1	89.2 224.9	211.4	82.5 63.6	64215.	656.	4.95	0	3.67	4.33
101	257.2	227.0	81.2 205.8	203.9	70.6 57.6	56876.	628.	3.79	0	3.93	6.20
	258.3	226.1	85.7 205.9	206.4	72.6 59.5	55288.	619.	3.81	2	3.93	6.65
	256.6	219.9	89.7 203.9	207.0	72.6 60.1	69708.	814.	2.88	2	4.05	7.60
	257.7	214.9	90.6 204.1	207.0	72.6 60.0	86464.	1029.	2.41	2	4.11	8.10
	257.6	212.6	92.4 205.5	206.5	72.6 59.6	95606.	1174.	2.12	2	4.22	8.32
	257.5	212.1	92.7 205.2	206.6	72.6 59.7	108520.	1335.	1.87	1	4.22	8.44
102	299.4	265.5	90.5 247.3	212.0	92.5 64.1	54794.	540.	4.30	0	3.69	3.30
	299.4	251.7	84.6 229.6	220.9	92.5 71.9	92977.	834.	2.52	0	3.27	4.00
	299.5	244.7	89.9 222.0	223.9	92.5 74.7	106750.	955.	2.13	2	3.17	4.57
	298.5	237.9	95.9 221.2	224.2	92.5 75.0	124570.	1171.	1.77	2	3.27	5.00
	298.5	235.8	97.1 221.2	224.2	92.5 75.0	129220.	1230.	1.65	2	3.30	5.12
	300.1	230.6	113.6 221.7	224.0	92.5 74.8	135810.	1409.	1.58	2	3.40	5.33
	301.1	225.1	104.8 221.2	224.2	92.5 75.0	163370.	1655.	1.35	1	3.42	5.66
103	288.6	248.7	93.1 229.7	218.6	90.5 60.9	67516.	678.	2.92	0	3.58	4.51
	290.1	244.6	96.3 224.5	217.4	87.5 68.8	70054.	691.	2.54	0	3.45	4.55
	291.0	231.6	95.9 216.0	217.2	84.5 68.6	105110.	1027.	1.85	2	3.37	5.15
	298.2	221.5	99.5 214.9	217.6	84.5 69.0	105730.	1106.	1.41	2	3.57	6.00
	298.3	218.6	97.7 216.4	217.1	84.5 68.5	126480.	1340.	1.24	1	3.63	6.23
104	290.2	260.8	76.7 243.1	214.4	92.6 66.2	57467.	572.	3.36	0	3.84	3.76
	289.2	251.3	70.2 229.9	221.9	93.6 72.8	71287.	653.	3.83	0	3.49	4.60
	290.2	239.7	89.2 221.1	224.2	92.6 75.0	105570.	1009.	1.70	2	3.39	5.41
	289.2	232.1	94.5 221.9	224.0	92.6 74.7	124020.	1262.	1.42	2	3.55	6.01
	289.2	224.8	68.1 219.6	224.8	92.6 75.5	157200.	1465.	1.26	1	3.52	6.81
105	283.8	230.5	76.0 231.8	209.4	83.5 62.0	53704.	571.	2.75	0	4.03	4.93
	284.6	213.7	75.6 213.5	216.9	83.5 68.4	83117.	824.	1.72	2	3.63	7.06
	283.8	210.4	76.1 212.6	216.1	82.5 67.6	105340.	1059.	1.35	1	3.68	7.36

FREED VOLUME POILER, #1

DATE	TEMPERATURE (F)					PRESSURES (PSIA)		HEAT TRANSFER		MASS RATIO	CU	STANTON	H-PR*JA
	H2O IN	H2O EX	F IN	F OUT	F SAT	VESSEL	SAT	GO	UA	H2O/FLUID		NUMBER	
								BTU/HR	BTU/HR F				
300	292.3	272.3	100.3	265.7	198.4	92.4	53.4	53061.	682.	8.53	0	4.89	2.94
	293.6	271.8	159.2	267.5	196.8	92.4	52.3	43970.	743.	8.34	0	5.34	2.80
	294.0	272.3	140.0	267.1	201.3	95.4	55.6	43961.	694.	7.73	0	5.15	2.97
301	291.4	266.6	158.8	259.2	203.7	92.4	57.5	62696.	1002.	5.46	0	4.94	3.20
	290.5	249.2	159.2	239.2	216.3	92.4	67.8	129300.	1878.	3.01	0	4.23	4.26
	290.1	256.6	157.9	247.9	211.5	92.4	63.7	93864.	1412.	3.81	0	4.52	3.76
	284.5	249.2	172.7	239.2	216.3	92.4	67.8	81794.	1375.	3.00	0	4.65	4.51
	294.9	264.0	169.7	255.3	209.1	94.4	61.7	56857.	911.	4.77	0	4.70	3.37
	292.3	263.6	172.7	254.9	213.8	102.4	70.0	50004.	830.	4.60	0	4.77	3.81
	293.1	264.0	167.9	256.6	192.1	82.4	49.0	61357.	997.	4.97	0	4.87	2.92
	288.3	240.5	185.3	233.6	219.0	92.4	70.2	52370.	818.	2.20	1	4.04	4.94
302	260.2	256.6	162.7	253.6	181.0	73.5	41.7	22912.	566.	19.16	0	7.54	3.29
	267.1	254.9	167.5	253.1	179.7	72.5	41.0	23046.	576.	15.29	0	7.50	3.27
	267.1	251.0	168.8	248.4	183.8	72.5	43.5	24311.	567.	11.12	0	6.87	3.48
	264.9	244.9	168.4	241.4	187.5	71.5	45.9	24513.	546.	8.49	0	6.44	3.82
	267.9	249.2	172.7	241.0	190.8	73.5	48.1	64218.	1352.	6.46	0	5.97	3.75
	267.5	247.5	174.0	238.8	190.7	72.5	48.0	64191.	1348.	5.89	0	5.89	3.81
	260.4	243.1	189.7	234.0	195.1	73.5	51.1	59160.	1369.	4.43	0	6.00	4.12
303	266.2	254.0	126.6	249.2	183.1	72.5	43.0	31328.	572.	15.67	0	6.25	3.40
	267.9	253.1	126.2	249.2	183.1	72.5	43.0	31202.	552.	12.37	0	6.06	3.30
	267.5	249.2	120.9	244.9	186.5	72.5	45.2	32056.	527.	9.93	0	5.66	3.60
	257.9	240.1	137.2	231.8	195.0	72.5	51.0	29560.	528.	7.51	0	5.63	4.62
	264.5	231.4	167.9	220.1	201.5	73.0	55.7	58313.	1093.	3.04	0	5.10	5.23
	263.6	244.5	125.8	236.2	175.0	61.5	38.2	73635.	1183.	7.36	0	5.39	3.42
	265.7	243.6	127.1	235.3	177.4	62.5	39.5	73065.	1140.	5.92	0	5.19	3.47
	255.3	226.6	145.5	215.3	191.8	64.5	48.8	62743.	1079.	3.60	0	5.07	5.15
304	250.0	236.6	88.5	232.3	160.7	52.4	30.8	49162.	717.	8.70	0	5.51	3.32
	254.5	237.9	90.7	232.7	160.3	52.4	30.6	48894.	745.	10.47	0	5.72	3.33
	262.7	239.7	92.9	233.6	161.6	53.4	31.2	40637.	668.	7.30	0	5.13	3.17
	261.8	229.7	119.6	221.0	168.9	52.4	34.9	77469.	1109.	4.19	0	4.76	3.60
	262.7	233.1	110.4	224.9	170.1	54.4	35.6	82065.	1138.	4.82	0	4.70	3.54
	263.1	237.9	94.2	229.2	163.1	52.4	31.9	85139.	1120.	6.08	0	4.84	3.23
	264.9	251.0	106.9	246.6	167.7	62.4	34.3	39644.	651.	12.63	0	6.01	3.00
	261.8	247.1	106.9	241.8	175.8	64.4	38.6	38566.	625.	10.72	0	5.84	3.43
	264.5	245.3	112.2	239.2	174.4	62.4	37.8	37810.	583.	8.57	0	5.44	3.37
	262.3	226.6	189.2	217.5	214.3	82.4	66.1	70038.	1709.	2.56	0	5.95	7.63
	264.0	229.7	184.5	220.1	213.3	82.4	65.2	106830.	2397.	2.68	0	5.64	6.80
	262.7	227.1	186.2	216.6	214.6	82.4	66.3	109030.	2511.	2.24	0	5.68	7.50
	262.3	221.8	187.1	212.3	217.3	83.4	68.7	10919.	260.	2.17	1	.60	9.16
305	255.7	241.8	88.0	238.4	150.7	50.5	26.2	44824.	716.	14.54	0	6.15	3.02
	254.9	238.4	89.8	234.0	154.9	50.5	28.1	44246.	680.	11.63	0	5.82	3.14
	252.3	235.3	90.2	229.2	154.7	48.5	28.0	44481.	671.	9.55	0	5.65	3.27
	253.6	231.8	90.7	225.7	153.1	46.5	27.3	44118.	632.	7.64	0	5.33	3.27
	252.3	218.4	107.3	209.2	168.2	48.5	34.5	78733.	1097.	3.95	0	4.71	4.13
	251.4	203.1	133.2	195.8	171.0	46.5	35.9	69226.	1107.	2.30	0	4.78	4.84
	254.9	226.2	101.2	217.5	167.3	50.5	34.1	78476.	1081.	5.12	0	4.84	3.80

252.3	230.1	96.8	220.5	167.3	51.5	34.1	79828.	1128.	6.16	0	5.06	3.77
255.7	234.5	97.7	227.5	164.5	52.5	32.6	76003.	1105.	8.16	0	5.27	3.49
254.0	234.6	98.0	228.4	178.6	60.5	40.3	76242.	1067.	7.93	0	5.45	4.08
252.3	215.3	135.0	206.6	192.5	62.5	40.3	68089.	1109.	3.37	0	4.89	6.11
306												
254.9	235.7	118.3	227.9	210.0	82.5	62.5	50027.	814.	6.19	0	5.36	6.64
246.6	234.5	92.0	228.8	163.5	52.5	32.2	45518.	759.	13.03	0	6.18	3.67

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