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ASPECTS OF THE KALINA TECHNOLOGY
APPLIED TO GEOTHERMAL POWER PRODUCTION

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SUMMARY

This report contains the results of studies conducted at the Idaho National Engineering Laboratory (INEL) concerning the applicability of the Kalina technology to geothermal (hydrothermal) power production. The original work was reported in Report EGG-EP-8132. This report represents a correction and addition to that report. The Heat Cycle Research Program (HCRP) has as its primary goal the cost-effective production of electric power from moderate temperature hydrothermal resources. Recent work has included the study of supercritical cycles with counterflow condensation which utilize mixtures as working fluids. These advanced concepts are projected to give a 20 to 30% improvement in power produced per unit geofluid flow rate (geofluid effectiveness, w hr/lb). The original Kalina cycle is a system which is similar to the cycles being studied in the Heat Cycle Research program and it was felt that this new cycle should be studied in the geothermal context.

Two studies were made: first, the use of the distillation-heat-rejection subsystem of the original Kalina cycle was investigated with the working fluids which had previously been studied in the HCRP; and second, a comparison was made between the new Kalina system, System 12 and the recuperative, supercritical Rankine cycles being studied in the HCRP.

Results of the first study indicate that small performance improvements were noted with addition of a distillation heat-rejection subsystem to cycles for working fluid mixtures which have high performance in standard supercritical cycles (hydrocarbon and halocarbon mixtures). These new cycles are more complex than the simple Rankine cycle with a large number of recuperative heat exchangers. The percentage improvement with this added subsystem was greatest with working fluid mixtures which had low performance

in the Rankine cycle at the geothermal temperatures. The supercritical Rankine cycles with the advanced concepts presently being studied in the Heat Cycle Research Program had efficiencies similar to those with this enhancement when there was no restriction on geofluid outlet temperature.

This conclusion is shared by Exergy, Inc., the company which is presently commercializing the Kalina Technology. In January 1989, Exergy presented the Kalina Technology system for geothermal use. This cycle used a mixture of water and ammonia, but did not have the distillation heat-rejection subsystem which was common to the other Kalina cycles. This system called System 12 is described in the report and compared in performance to a supercritical, recuperated Rankine cycle which has been investigated in the HCRP. The second part of the study indicated that for the specific applications considered in the report, the performance of the Rankine cycle was slightly superior to that of System 12. With the exception of the Heber conditions, the performance was 4 to 5% better in terms of second-law efficiency, and 2 to 4% better in terms of net geofluid effectiveness. For the Heber Binary Plant summer design conditions, the Rankine cycle was 9% better by each criterion. (This is possibly a result of not having the optimum system for the Kalina technology cycle.) The issues of cost-effectiveness and operability have not been addressed in this report and may prove that either system may be the appropriate system for a given application.

This report contains a brief discussion of the mechanics of the Kalina cycle. A modified cycle in which the heat rejection part of a supercritical cycle with mixed hydrocarbon working fluids is discussed. Then the new Kalina technology cycle, System 12 is compared with a supercritical, recuperated Rankine cycle. Finally, the conclusions of this scoping work are given along with recommendations of the direction of future work in this area.

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The author wishes to thank the following people for their help with this report. First, the people at INEL, G. L. Mines at EG&G, and K. J. Taylor at DOE's Idaho Operations Office were helpful in their review of the document. The discussions with O. J. Demuth, originally with and now a consultant to the Heat Cycle Research Program, gave insight into the problem and helped to provide a more coherent way to present the results. The review by Dr. D. Y. Goswami of the Mechanical Engineering Department of North Carolina Agricultural and Technical State University gave an outside point-of-view. The input from the DOE program, Raymond LaSala was also helpful in producing this document. Last, the information received from Exergy, Inc. and discussions with Dr. M. Tribus and H. M. Leibowitz; and the day spent with Dr. Alexander Kalina at his Hayward, California office in which I learned his philosophy of energy conversion systems and obtained results of the Kalina cycle System 12.

ASPECTS OF THE KALINA TECHNOLOGY APPLIED TO GEOTHERMAL POWER PRODUCTION

INTRODUCTION

The Heat Cycle Research Program (HCRP) has as its primary goal the cost-effective production of electric power from moderate temperature hydrothermal resources. The investigations have concentrated on liquid resources at temperatures less than 400°F. Specifically, in the past, binary power cycles have been examined because they are more efficient than flash steam cycles (achieve a higher net brine effectiveness) at these temperatures. Recent work has included the analytical and experimental study of supercritical cycles with counterflow condensation which utilize mixtures as working fluids both with and without exhaust gas recuperation.^{1,2,3,4} These advanced concepts are projected to give a 20 to 30% improvement in power produced per unit geofluid flow rate (geofluid effectiveness, w hr/lb) over the power produced by an advanced binary power plant similar to the plant in the Heber Binary Project.

Dr. Alexander Kalina invented an innovative cycle for bottoming a gas turbine which employs a distillation system.⁵ The gas turbine bottoming duty is similar to a system which utilizes a geothermal heat source similar to those studied in the Heat Cycle Research program and it was felt that this innovative idea should be studied in the geothermal context. Since that time, Dr. Kalina at Exergy, Inc. has been applying the Kalina Technology to a number of different systems.

This report deals with the studies at the Idaho National Engineering Laboratory (INEL) to implement some of the ideas in the Kalina Technology to production of power from hydrothermal resources. It should be noted that there may additionally be a fit between the Kalina technology and power production from geopressured

resources. The production of combustable gas in the geopressed situation might supply a gas turbine which the rest of the power production system bottomed. This concept is presently being considered by the U. S. Department of Energy (DOE). The technical requirements and power cycle configurations for these two geothermal applications are quite different. This report will concentrate on the type of system for power production with a hydrothermal resource and leave study of the Kalina technology for a geopressed resource for separate consideration because the technical problems are so different.

The original study at the INEL involved use of the distillation concept in power cycles with working fluids which had been shown to be effective in simple Rankine cycles. After this study was completed, Exergy, Inc., the company which is developing the Kalina technology, devised a cycle for geothermal use. This report is, then, divided into two parts: the first section reporting on the initial study and the second section showing the performance of the new application of Kalina technology compared with systems developed in the HCRP.

The original Kalina cycle (see Figure 1) has recently received attention as a possible enhancement to gas turbine power production.⁵ Since that time, Dr. Kalina and Exergy, Inc. have produced a number of modified systems for various applications and have improved the efficiency of the original system.⁶ The system under consideration for this application consists of a conventional unrecuperated gas turbine with Kalina's unique cycle as a bottoming system using the gas turbine exhaust or waste-heat as an energy source. This combined cycle gives thermal efficiencies of around 50% (compared with 35 to 40% for conventional steam power plants).⁵ The first-ever Kalina cycle plant is being constructed and tested at the DOE's Energy Technology Engineering Center (ETEC) located at Canoga Park, California. It will use 1050°F gas from a gas-fired waste heat facility as its heat source, cooling the gas to 146°F.⁷

The original work at the INEL consisted of examining the distillation concept in conjunction with the heat rejection system in a geothermal power cycle. Because of the similarities in the gas turbine exhaust and waste heat sources with a hydrothermal heat source, it was felt that the distillation-system concept might offer a performance improvement in the geothermal application. Although both heat sources are similar (having variable temperature and approximately constant specific heat), the temperature ranges are quite different. The present Kalina cycle heat sources are at ~ 1000 to 1050°F while geothermal resources of interest are at significantly lower temperatures (300 to 450°F). Work with Rankine cycles indicates that working fluids different from the ammonia/water mixture would probably be necessary for optimized performance. Accordingly, this scoping study considered three working fluid mixtures: ammonia/water (the fluid used in the high temperature Kalina applications), isobutane/heptane (one of the best hydrocarbon mixtures with conventional supercritical cycles when used with a hydrothermal source at 360°F)^{1,2} and Refrigerant 22/114 (the fluid studied in the halocarbon study that gave comparable results to those of the isobutane/heptane hydrocarbon mixture).³ A mixture with a heavier component (Benzene/Propane) was also considered because it was felt that a mixture with a wider boiling range might improve the performance.

After this study was completed, Exergy, Inc. applied the Kalina technology to geothermal power production and devised a system which they called System 12. This new system was first presented at a seminar in January 1989.⁸ This system does not use the distillation concept but is effectively a Rankine-type cycle with a complex heat exchanger system integrating the heat addition, heat rejection and recuperation activities. It employs a mixture of water and ammonia as the working fluid, as do all applications of the Kalina technology. The second part of this report compares the Kalina System 12 to the advanced binary cycles studied by the HCRP.

This report first discusses the original Kalina cycle concepts and the new application of Kalina technology to geothermal power production. The next section considers the study that applied the Kalina concept of a distillation-heat-rejection subsystem to the conventional binary geothermal systems. The comparison of the new Kalina technology produced by Exergy, Inc. is then compared with the new technology developed by the HCRP. Finally, conclusions and recommendations about application of the Kalina technology to geothermal power systems is given. Appendices contain computer-calculated results from Exergy, Inc. for the System 12 and from INEL for the recuperative, supercritical Rankine cycles.

THE KALINA CYCLE CONCEPTS AND TECHNOLOGY

Although the present state of Kalina technology has advanced well beyond the original concept of the Kalina cycle, a study of the original concept is a good starting point. This section discusses the original Kalina cycle in some detail to orient the reader to the new concepts. Then, the extensions and refinements of the concept are discussed, including the System 12 that has been proposed by Exergy, Inc. as the appropriate application of the Kalina technology to geothermal power production.

The original Kalina cycle is a modification of the Rankine cycle using a mixture for a working fluid.⁵ Dr. Kalina proposed to use a 70/30 mass percent mixture of ammonia and water. His cycle introduced two areas of improvement over the conventional boiling Rankine cycle used with a variable temperature heat source. First, the heating and boiling of the mixture was to take place at a varying temperature and, thereby, matching the cooling curve of the heat source (turbine exhaust or geofluid) and reducing the irreversibilities generated in that heat transfer process. This effect is presently being exploited in the geothermal, mixed-hydrocarbon, supercritical, binary cycles used in the Heber plant and those being explored at the Heat Cycle Research Facility (HCRF) at the Idaho National Engineering Laboratory. In these geothermal applications, the working fluid is heated at pressures above the critical pressure, giving a good match between the heating curve of the working fluid and the cooling curve of the heat source creating little irreversibility. There is a practical concern that the boiling in the Kalina cycle may not be a very effective process. The boiling process must take place in an integral, counterflow arrangement to achieve the maximum thermodynamic benefit. The working fluid will be completely boiled and superheated. Boiling heat transfer coefficients for mixtures are substantially lower than those for pure fluids. The integral boiling would entail use of a once-through boiler with complete evaporation of the working fluid. The tube walls would dry out and there would be an area of quite inefficient heat transfer.

Second, the original Kalina cycle was the unique heat rejection system. Figure 1 shows a schematic diagram of the entire cycle. The use of the distillation subsystem allowed the composition of the working fluid in Condenser 1 to be less volatile than that of the working fluid that went through the turbine. The cost of the required distillation subsystem is, however, not negligible and must be traded-off with the potential performance improvement. In addition, although the total heat duty in Condensers 1 and 2 will be less than for the cycle without the Kalina modification because of the increased efficiency, the average temperature during condensing will be lower. This fact results in lower possible cooling water temperature increases, for the same cooling water inlet temperature and, therefore, higher cooling water flow rates, increasing the parasitic power requirement to condense the working fluid. Lower mean temperature difference will result and may possibly increase the size of the condensers over those in a conventional power cycle. Again, these apparent added capital costs must be traded-off against performance improvements.

Figure 2 shows a possible configuration of the distillation subsystem. Here, the separation is accomplished by heating the liquid stream that leaves Condenser 1 above its bubble point, and producing a two-phase fluid. The liquid and vapor phases are separated, and if required, some of the liquid is mixed with the vapor if it is necessary to balance the state of the fluid. Note that all of the energy required to boil the stream that is to be separated does not necessarily come from the turbine outlet vapor flow; the separated liquid and vapor streams may supply some of the energy. The turbine outlet vapor must supply only the temperature to give adequate pinch points in the recuperative heat exchanger. If the boiling range (difference in dew point and bubble point temperatures) is not large enough, there may not be heat available at an adequate temperature in the turbine outlet vapor to perform the separation. In the geothermal application, the heat source fluid (geofluid) may be used in addition for the distillation. Use of the

Figure 1. Original Kalina Cycle for Bottoming a Gas Turbine.

9-7448

Figure 2. Simple Distillation Subsystem for Kalina Cycle.

geothermal energy would improve the power output, but may not decrease the condenser heat load because the power increase results from more heat added to the system rather than from an increase in thermal efficiency.

The distillation-heat-rejection subsystem has been improved in later versions of the Kalina cycle. Figure 3 shows the distillation subsystem that will be used on the demonstration plant, which is being constructed at the Energy Technology Engineering Center at Canoga Park California. In this version, some preheating of the working fluid is accomplished in the subsystem. Figure 4 shows a process flow diagram indicating the hardware needed in the subsystem. In the case of the High Pressure (HP) Condenser and the Low Pressure (LP) Condenser, the function of more than one heat exchanger in Figure 3 is accomplished within a single shell. These units are rather large, the one at the LP condenser is a vertical unit 118 in. in diameter and about 60 ft. high for the 3 MW demonstration plant. The HP condenser must be also a relatively tall vertical unit. The evaporation takes place in two units, a falling-film evaporator and as part of the heat exchanger at the right. In addition, the function of the first recuperator is accomplished in two units and some of the regeneration takes place by using direct-contact sprays of working fluid within the condenser shells.

Recent literature^{6,8} published on the simple Kalina cycle applied to the heat sources at 1000 to 1100°F indicate that efficiencies based on the available energy (exergy) in the heat source stream are ~70%. The available energy is referenced to an ambient condition related to cooling water temperature into and out of the cycle. These efficiencies do not include any parasitic power requirements associated with the supply of the cooling water or the heat source fluid. Private discussions with Dr. Kalina indicate that he believes that it is generally not cost-effective to push the second-law efficiency past 70 to 75%. In all of the prior studies made at the INEL on geothermal power plants, the cooling parasitic

Figure 3. Improved Distillation Subsystem

9-7269

9-7268

Figure 4. Process Flow Diagram for Improved Distillation Subsystem

was accounted for because at the geothermal temperatures it is a substantial fraction of the net power (at the higher temperatures for these applications it is insignificant); but the heat source parasitic was not accounted for because it is a site-specific quantity.

Figure 5 shows a modified cycle that can obtain second-law efficiencies on the order of 75% for the high temperature heat sources mentioned above. Again, the efficiency considers the net plant power and does not include the parasitic power required to produce cooling water or otherwise account for heat rejection. The modifications include:

1. Reheating the working fluid after partial expansion through the first turbine with a parallel stream of source fluid
2. Recooling the working fluid after further expansion through the second turbine and partially heating part of the working fluid
3. Finally expanding through a third turbine
4. A slightly more complex distillation-heat-rejection system.

Earlier in the geothermal HCRP, some similar heat addition strategies were proposed. These included multiple-boiling cycles (See Reference 14). Although not identical to the Kalina modification, multiple boiling cycles do decrease the heat addition irreversibilities by means of more complicated heat exchanger arrangements and the use of multiple turbines. Industrial reaction to this type of cycle indicated that the system was too complex for the geothermal application. This system was not recommended by Exergy, Inc. for use with relatively low temperature heat sources as are found in the geothermal application, but is presented here to give the reader a feel for possibilities with advanced Kalina technology.

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Figure 5. Advanced Kalina Cycle

Figure 6. Kalina System 12 for Geothermal Source

The new system which Exergy, Inc. has recently proposed for geothermal use is shown in Figure 6.^{8,9} This system uses a mixture of water and ammonia which is the working fluid for all Kalina cycles do. Note that this cycle does not use the concept of the distillation heat-rejection subsystem. The cycle is a Rankine-type cycle with turbine reheat and recuperative preheating of the working fluid with the turbine exhaust. This system is quite similar to the advanced Kalina cycle recommended for high temperature heat sources shown in Figure 5. The third turbine stage and distillation-heat rejection system has been replaced with the recuperator that preheats the working fluid and the precooler has been moved to preheat the working fluid instead of vaporizing it. Otherwise, the cycles are identical. Note that one effect of the distillation-heat rejection system is to provide recuperative use of the energy in the turbine exhaust. In order to obtain a better match between heating and cooling curves inside each heat exchanger (and, thereby, reduce the heat exchange irreversibility), the source fluid has been split and flows in parallel in the superheater and the reheater. On the low end, the working fluid is split and part of it is preheated by the source fluid (geofluid) and part recuperatively by the turbine exhaust.

The experience of the HCRP has been that supercritical cycles having mixed-hydrocarbon or halocarbon working fluids produce a match between the heating curve for the working fluid and the cooling curve for the source fluid is good enough that a more complicated heat exchanger network is unnecessary.¹ Similarly, where the geofluid outlet temperature is limited (by silica precipitation concerns), a supercritical Rankine cycle with these same working fluids give good performance if the working fluid is preheated recuperatively by turbine exhaust.² Comparison of the Kalina System 12 with the HCRP recuperative cycle is made in a subsequent section of the report.

DISTILLATION-HEAT-REJECTION SUBSYSTEM IN SUPERCRITICAL CYCLES

The geothermal heat source is similar to the turbine exhaust heat source for which the original Kalina cycle was designed. For that reason it was believed that use of the distillation-heat-rejection subsystem with a supercritical cycle similar to those proposed by the early work in the HCRP might show performance improvements. The gains of the Kalina cycle on the heat addition portion of the cycle are similar to the gains realized by the supercritical heating. Therefore, addition of the distillation subsystem was felt to potentially increase performance. This section discusses the methodology, results and conclusions of this study.

METHODOLOGY OF STUDY

A preliminary investigation of the use of the distillation subsystem for a geothermal power application was carried out to assess this potential. For the study, a hydrothermal resource at 360°F was chosen as the heat source to be consistent with previous analyses.^{1,2,3} No restrictions on geofluid outlet temperature were considered. Heat rejection was assumed to be with an evaporative condenser because it was felt that this type of heat rejection system would be more favorable to the Kalina cycle with its low heat rejection temperature than would a conventional condenser with a wet cooling tower. (The heat rejection at lower temperatures requires larger condensers and potentially the added capital cost will offset the gains in performance.) This type of heat rejection system will favor the Kalina cycle with its small temperature variation during condensation over the conventional cycle that has a larger change in temperature during condensation. If conventional wet cooling tower and shell-and-tube condensers were used, the amount of cooling water circulated, and therefore, the size of the heat rejection system and heat rejection parasitic power loss would have been so large for the Kalina cycle as to make it noncompetitive with the supercritical Rankine cycle. The wet bulb temperature was taken to be 60°F and the performance was as outlined in Reference 1. All of the

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When no large gain was exhibited by these working fluids, a mixture of propane and benzene was investigated using the EXCST computer program. It was thought that this mixture would have the wide boiling range exhibited by the original ammonia/water mixture of Kalina, but have a suitable critical temperature for this low temperature application.

RESULTS OF STUDY

Table 1 gives a summary of the results for the basic Kalina cycle (and the cycle with supercritical heating) using the original three mixtures. For each mixture, a standard supercritical Rankine cycle is compared with a cycle modified by the addition of a distillation-heat-rejection subsystem. In the case of both the isobutane/heptane and Refrigerant 22/114 mixtures, there was not enough energy in the turbine exhaust at a high enough temperature to drive the distillation subsystem. In each of these cases the geofluid, after leaving the heater, was used in addition to the turbine exhaust to vaporize the stream to be separated. This caused the geofluid outlet temperature in the modified cycles to be significantly lower than in the comparable conventional cycle. The pinch points in the distillation subsystem for these two working fluids were too low to be practical, 1 to 2°F. If reasonable pinch points had been assumed, the improvement in performance would have been less than is shown in Table 1. The results for these two working fluids may then be thought of as optimistic estimates of the modified cycle improvements because in a real situation, the pinch points would be considerably larger.

The improvement with the isobutane/heptane mixture is quite small. Utilizing the turbine exhaust and the geofluid leaving the heater as energy inputs to the distillation subsystem result in a possible change in composition which creates a change in turbine back pressure from 56 psia in the standard cycle to 54 psia in the modified cycle (the turbine pressure ratio increasing from 10.7 to 11.1). This results in an increase in turbine power of less than one

Table 1. RANKINE CYCLE COMPARISON WITH EVAPORATIVE CONDENSER

($T_{\text{Wet Bulb}} = 60^{\circ}\text{F}$, $T_{\text{geofluid}} = 360^{\circ}\text{F}$)

Fluid Mixture		Isobutane/Heptane Refrigerant 22/114				Ammonia/Water	
Cycle ^a		Standard	Modified	Standard	Modified	Standard	Modified
Composition (%)	Heater Condenser	94/6 94/6	94/6 92/8	50/50 50/50	50/50 40/60	70/30 70/30	70/30 40/69
Pressure (psia)	Heater Condenser	600 56	600 54	1100 130	1100 118	200 110	200 32
Temperature ($^{\circ}\text{F}$)	Geofluid Outlet	142	92	125	95	110	110
Thermal Efficiency (%)		14.2	11.7	13.5	12.2	4.7	11.8
Second Law Plant Efficiency (%)		62.9	63.7	64.9	67.3	21.3	60.2
Power (Whr/lbgeo)	Turbine	11.21	11.31	12.52	13.07	3.44	9.98
	Pump	1.18	1.18	2.34	2.34	0.04	0.38
	Cooling Tower	0.96	1.04	0.89	1.19	0.16	0.95
	Net	9.07	9.09	9.29	9.54	3.24	8.65

^a With all but the ammonia/water working fluid, the cycles were supercritical.

percent and a decrease in thermal efficiency because of the addition of heat at a low temperature. In the geothermal cycle, the thermal efficiency is not the most important parameter, however, and the results are also expressed in terms of geofluid effectiveness (W h/lb geofluid). These results indicate a < 1% increase in net geofluid effectiveness. The turbine power is increased, but the cooling parasitic is increased also because of the lower heat rejection temperature and the decrease in thermal efficiency. There are a number of reasons for the small increment in performance in the modified cycle with this working fluid. First, the condensing (boiling) range is only around 32°F. For the modified cycle to work at its optimum, a large amount of energy should be available in the turbine exhaust at a relatively high temperature to effect the distillation, thereby, requiring a large boiling range (greater than 100°F as with the water-ammonia mixture). Second, this system is already optimized in the standard cycle and had a relatively high geofluid effectiveness.

The refrigerant 22/114 mixture shows a slightly larger performance improvement than the hydrocarbon mixture. Here, if energy directly from the geofluid is used to supplement the energy in the turbine exhaust, the turbine back pressure can be reduced from 130 to 118 psia giving an increase in turbine power of 4.3%. The cooling parasitic is increased also, for the same reasons as for the hydrocarbon mixture, and a net effectiveness increase of only 2.7% results.

The largest improvement for the modified cycle is noted in the water/ammonia mixture. Here, the turbine back pressure decreases from 110 to 32 psia giving an increase in pressure ratio from 1.8 to 6.2 and a power increase of 190%. With this wide boiling range mixture (approximately 165°F), the distillation subsystem works with only the turbine exhaust for heat input and 10°F pinch points. The net geofluid effectiveness increases 167% over the corresponding conventional cycle. The effectiveness of the standard cycle chosen was quite low, however, and was probably not the maximum for this fluid. The resulting effectiveness is still lower than for

the other working fluids, but only by ~5%. Another interesting fact is that the heater pressure with this working fluid is only 200 psia compared to 600 psia for the hydrocarbon mixture and 1100 psia for the halocarbon mixture.

Thermal efficiencies and second law efficiencies are given for each system. The thermal efficiency is the standard net power output divided by the heat input. The net work includes the turbine work as well as the working fluid pumping and heat rejection parasitic. The second law efficiency is defined in the same manner as by Kalina.^{5,6,7} This is the net work output (excluding cooling parasitic) divided by the available energy (exergy) in the heat source stream referred to the average coolant temperature as the environmental temperature. The thermal efficiency is not very meaningful in this application. A high thermal efficiency is obtained if less heat is removed from a unit mass of geofluid (therefore, at a higher average temperature). This would increase thermal efficiency at the expense of geofluid utilization which is not desirable. The second law efficiency incorporates the same ideas as the geofluid effectiveness which was used in the early geothermal studies.^{1,2,3} Note that for the most part the second law efficiency of the isobutane/heptane and R-22/R-114 mixtures that were optimized without the modification are relatively high. The modification improves them slightly. The ammonia/water cycle is improved the most, but still falls short of the performance of the cycles using the working fluids studied in the HCRP.

The geofluid outlet temperature in all of the applications with the distillation subsystem investigated is quite low. If a particular resource has an outlet temperature that is limited by precipitation of silica in the geofluid, the modified cycle would not be a good choice. Other working fluid cycles may not result in the low geofluid outlet temperatures found in this brief study. The use of the benzene/propane mixture proved to be no better than the other working fluids. It was thought that with the wider boiling range, the fluid would be more favorable for the modification. No

composition analyzed produced a cycle as favorable in performance as those already considered. The problem of obtaining energy to effect the separation from the turbine exhaust continued to be the controlling problem. In addition, the wide boiling range working fluid did not show the inherent performance of the hydrocarbon and halocarbon mixtures.

CONCLUSIONS AND RECOMMENDATIONS

Although the study was short and scoping in nature, certain conclusions may be drawn:

1. For the modified cycle, the use of geofluid as a heat source for the distillation subsystem is generally not very effective relative to the better binary cycles presently being investigated. Although the turbine power can be increased, the cooling parasitic power increase to partially nullify the gain. The required condenser size may be increased because the condenser heat load will probably be increased.
2. Large increments in performance by the addition of the distillation subsystem have been realized for working fluids with large condensing ranges. Unfortunately, the working fluids studied either have small condensing ranges (hydrocarbon and fluorocarbon mixtures studied) or are not well suited for this temperature range in a simple configuration (ammonia/water mixture). Even the propane/benzene mixture that has a wide boiling range did not produce high performance for this application. The modification overcomes the disadvantages of the ammonia/water mixture in this temperature range, but at the expense of a complex heat addition system.
3. The combination of the distillation-heat-rejection subsystem with different working fluids will give an extra degree of freedom in determining the best performance.

There may be some fluorocarbon or hydrocarbon mixtures that combined with the Kalina cycle will give superior performance to all existing cycles studied, although it is suspected that it will be difficult to surpass the geofluid effectiveness of the standard supercritical systems previously analyzed. There is a certain amount of irreversibility inherent in a cycle and the cycles previously considered are close to that limit. The primary way to decrease the irreversibility is to find a better heat rejection system.

It is recommended that further work be initiated that will consider different working fluids to be used in conjunction with the modified cycle when studies of heat rejection with saving in makeup water is studied.

COMPARISON OF KALINA'S SYSTEM 12 WITH ADVANCED HEAT CYCLE SYSTEMS

In the previous section it was shown that the simple Kalina cycle, Rankine cycle with a distillation-heat-rejection subsystem using ammonia/water as working fluid, was not an appropriate choice for the low temperature heat sources in the geothermal application. Although it is not yet clear whether other working fluids might give improved performance with this modified cycle configuration in certain applications, ammonia/water mixtures working fluids give low performance at these temperatures. Exergy, Inc. has devised a system that uses the water/ammonia mixture for heat sources in this temperature range ($<400^{\circ}\text{F}$)^a. The new system was described in a previous section of this report and is represented schematically in Figure 6. The new system does not use a distillation-heat rejection subsystem. It is a Rankine cycle with turbine reheat and recuperation of the energy in the turbine exhaust. The new system uses a unique heat exchanger network splitting both the geofluid flow and the working fluid flow during its preheating phase. This new system is compared with the supercritical, regenerative Rankine cycle that has been studied in the HCRP. This section presents the methodology of the study, the results, and conclusions.

METHODOLOGY OF STUDY

A comparison was made between this new system that uses Kalina technology and the innovations investigated by the HCRP for specific cases. Two cases were considered:

1. A resource similar to the resource of the Heber Binary Plant. A resource temperature which was available at 360°F with a minimum geofluid outlet temperature of 160°F and cooling water available at 70°F (corresponding to a wet bulb temperature of 60°F , summer design).

^aThis new system is the System 12 which was first shown to the public in January 1989 at a seminar given by Exergy, Inc. in Burlingame, California.^{8,9}

9-7265

Figure 7. Recuperative, Supercritical Rankine Cycle Studied in the HCRP.

9-7266

Figure 8. Working Fluid Pressure Losses in System 12.

9-7262

Figure 9. Working Fluid Pressure Losses in Recuperative Supercritical Rankine Cycle.

1a. A second case was run to determine cycle sensitivity to changes in geofluid outlet temperature by taking Case 1 with the minimum geofluid outlet temperature of 170°F.

2. A resource that Exergy was investigating for a geothermal developer. The resource temperature was 367°F, with a minimum geofluid outlet temperature of 170°F and cooling water available at 53°F.

The HCRP recommendation for this type of resource is a recuperated, supercritical cycle.^{1,2} Figure 7 shows a schematic diagram of this cycle. Note the similarity to the original Kalina cycle with the distillation-heat-rejection subsystem replaced by the recuperator and condenser.

Specification of all conditions for the two cycles were made similar to make a valid comparison. The turbine isentropic efficiency was set at 86%; the pump isentropic efficiency at 80% and the generator efficiency at 97.5%. The following working fluid temperature constraints were used:

1. The working fluid leaving the geofluid heater was at least 15°F lower than the geofluid temperature entering the heater.

2. The minimum stream-to-stream temperature difference (pinch point temperature difference) in each geofluid heater was 10°F.

3. The minimum stream-to-stream temperature difference in the condenser was 5°F.

4. The working fluid pressure loss in heat exchangers was assigned in a similar manner for the two analyses.

Figure 8 indicates the pressure losses assigned by Exergy, Inc. in their analysis. Note that where liquid is

preheated, 10 psi is used; for the evaporator, 5 psi and 5 psi in each superheater. On the low pressure side, 1 psi is taken in one recuperator, while 0.3 psi in the other recuperator and the condenser. Figure 9 shows similar information for the recuperative, supercritical cycle. Here, 10 psi is taken for the liquid in the recuperator and 20 psi for the geofluid heater. All experiments at the HCRF indicate that these are very conservative numbers in that the entire supercritical heaters require between 5 and 10 psi in the experiment.⁴ On the low pressure side, 1 psi was taken in the recuperator and 1 psi in the condenser. Again, the experience at the HCRF indicate that this is conservative because the measured pressure drop during the entire desuperheating and condensing process has been about 1 psi.⁴ Preliminary calculations indicate that on the vapor side, the dynamic pressures are similar while on the liquid side, the System 12 has lower dynamic pressures. Therefore, the pressure loss assignments appear to be reasonable.

The results are presented in two ways. First, Kalina has generally expressed system performance in terms of a second-law efficiency using net plant power (generator output - pump work). The denominator of the efficiency term has been taken as the change in available energy (exergy) for the heat source stream. In this case, where there is a minimum constraint on the geofluid outlet temperature, that will be the change in available energy between the inlet state and the minimum outlet temperature. Kalina references this available energy to cooling water temperature as the ambient and he has shown that the ambient should be the log-mean average between the inlet and outlet temperatures of the cooling water.

Second, analysis at the INEL has generally included the cooling parasitic power loss feeling that this parasitic loss was significant at the low heat source temperatures encountered in the geothermal

application, and referenced the net power (net plant power - cooling parasitic) to the total geofluid flow rate, i.e., net geofluid effectiveness in watt of electrical power per unit lbm/h of geofluid flow. This performance parameter has been found to come close to the economic optimum design because of the extremely high cost of the geothermal field. (The economic optimum is close to obtaining the most electrical energy possible per unit mass of geofluid.) The basis for the cooling parasitic is discussed in References 1 and 2. (For a 20°F cooling water rise, assuming 10 psi drop in the condenser, 40 ft head rise at the tower, line losses and cooling tower fan power gave 0.0770 W h/lbm cooling water. (This was confirmed with actual hardware.)) When variations in cooling water temperature rise occur, small changes in pumping power result in the following equation:

$$W_{ct}/m_{cw} = 0.0618 + 0.00076 (T_{cw})$$

where,

W_{ct} is the cooling tower parasitic power in w,

m_{cw} is the cooling water flow in lb/h,

and, T_{cw} is the cooling water temperature rise in °F.

The Exergy, Inc. results are modified in this manner.

The Exergy, Inc. calculations accounted for pressure losses in the heat exchangers. These losses are neglected in the INEL program. The Heat Cycle values are corrected by the following method:

1. The pump power is increased proportional to the increase in pressure rise from pump inlet to turbine inlet. (This will be an excellent approximation of the real value because the liquid being pumped is essentially incompressible.)
2. The turbine power is recalculated using an increased back pressure resulting from the pressure losses in the recuperator and condenser. The actual thermodynamic properties were used for this calculation.

3. The heat added was known, so the heat rejected could be determined knowing the revised pump and turbine power. The cooling parasitic power loss was adjusted by the ratio of the heat rejected including the pressure losses in heat exchangers to that neglecting those pressure losses.

STUDY RESULTS

The results of the Exergy, Inc. computer program are given in Appendix A along with the estimated cooling parasites for each case. Appendix B contains the computer results for the Heat Cycle regenerative Rankine cycle along with the adjustments for pressure losses in the heat exchangers.

Table 2 shows the comparison between the systems for each resource. The first resource specifications were essentially for the Heber Plant summer design. As a comparison, a prediction of the Heber Plant performance consistent with the assumptions of the study is given. (The heater pinch point in the Heber calculation was 10°F , however.) The System 12 gives a net geofluid effectiveness, work per unit mass of geofluid, about 14% higher than Heber. The supercritical, recuperated Rankine cycle was about 24% higher than the Heber Plant. These comparisons penalize the Heber plant because of the heater pinch point. The numbers in parentheses for the supercritical cycle represent a 10°F heater pinch point. Here, the improvement over Heber is 20% as quoted in previous work.^{2,4} When the geofluid outlet temperature constraint is raised to 170°F , the second-law efficiency of the System 12 increases dramatically while that of the Rankine cycle remains practically constant. The absolute power out from the System 12 increases, when it takes less energy from the source. This indicates that the performance of the System 12 is quite sensitive to changes in constraints placed on the cycle. Possibly, with different working fluid mixture and operating pressures, the efficiency of the first system could be increased.

Table 2. COMPARISON OF "SYSTEM 12" TO SUPERCRITICAL,
RECUPERATED RANKINE CYCLE

	Geofluid Inlet Temp.	Minimum Geofluid Outlet Temp.	Cooling Water Temperature	Second Law Efficiency ^a	Net Geofluid Effectiveness ^a
	°F	°F	°F	%	Wh/lb gas
Heber Binary Plant	360	160	70	56.8	7.73
Kalina "System 12"	360	160	70	63.9	8.81
Supercritical, Recuperated Rankine Cycle	360	160	70	69.7 (67.9)	9.50 (9.33)
Kalina "System 12"	360	170	70	63.9	9.03
Supercritical, Recuperated Rankine Cycle	360	170	70	69.8 (68.1)	9.35 (9.14)
Kalina "System 12"	367	170	53	69.4	10.76
Supercritical, Recuperated Rankine Cycle	367	170	53	73.1 (69.3)	11.03 (10.76)

^aNumbers in parameters represent 10°F heater pinch points

The third comparison, the resource that Exergy had put the most effort into optimizing, gave similar results. When the heater pinch points were the same for both systems, the Rankine cycle had a performance 3 to 5% higher. The performance was essentially the same with a 10°F pinch point in the Rankine compared with a 5°F pinch in the System 12. Because heat transfer coefficients for the ammonia/water mixture are not known as well as for the hydrocarbon mixture, it is felt that similar penalties will be applicable during mixture phase change and that mixture boiling and supercritical heating will have comparable overall heat transfer coefficients.

CONCLUSIONS

The following conclusions are made with respect to the comparison between the Kalina System 12 and a supercritical, recuperated Rankine cycle:

1. The performance of the System 12 is lower than that of the Rankine cycle for similar constraints.
2. The System 12 is more complex because of the flow splits in both working fluid and geofluid that must be controlled. There are two separate turbines in the System 12, also.
3. Some practical problems may result handling two-phase mixtures being transported between exchangers. In the System 12, the fluid leaving the recuperative preheater, the recuperator and condenser (on the condensing sides) are all two phase. A similar situation is possible in the Rankine cycle between the recuperator and condenser (on the condensing side). It is possible for the functions of these units to take place within a common shell with the two-phase mixture on the shell side if the units are vertical. This was discussed in a previous section. A more serious condition will be with the high pressure streams leaving the preheater and recuperative preheater in

the System 12. It appears that if the evaporator is to be a counterflow device, maldistribution will occur with a two-phase mixture entering the exchanger. If counterflow is not maintained in the evaporator, the thermodynamic advantage of the mixture is lost.

CONCLUSIONS

The conclusions of the various parts of this report are summarized below:

1. The addition of a distillation heat-rejection subsystem (similar to those used in a simple Kalina cycle) to a supercritical Rankine cycle optimized for geothermal use appears to give little performance improvement. This study involved the working fluids which have proved optimum for the geothermal application such as mixtures of hydrocarbons and halocarbons. The use of a water/ammonia mixture with this simple configuration did not give high performance.
2. The performance of a recuperated, supercritical Rankine cycle for a geothermal resource in which the outlet temperature is restricted to control silica precipitation appears to be a little better than Exergy, Inc.'s new System 12. The performance of the Rankine cycle is higher by 2 to 5% and it is a less complex system. Economic considerations have not been taken into account in this conclusion and it is possible that the System 12 could be more cost-effective.
3. Further examination of the distillation subsystem should be considered at in conjunction with advanced heat rejection. Systems further from optimum may give greater performance enhancement than the systems considered. Integration of this subsystem into an overall heat rejection system including combined wet and dry cooling should be pursued.

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APPENDIX A
STATE POINTS FOR THE KALINA "SYSTEM 12"

A-1/A-2

!	#	P(psia)	X	T deg F	H (Btu/lb)	G	!
	1	-	BRINE	360.00	-	2.2525	
	2	-	BRINE	331.79	-	2.2525	
	3	-	BRINE	236.00	-	2.2525	
	4	-	BRINE	160.00	-	2.2525	
	5	-	BRINE	360.00	-	1.2293	
	6	-	BRINE	360.00	-	1.0232	
	7	-	BRINE	331.79	-	1.2293	
	8	-	BRINE	331.79	-	1.0232	
	9	-	BRINE	160.00	-	2.2525	
	14	121.42	0.8405	77.00	1.04	1.0000	
	21	386.89	0.8405	77.00	2.25	1.0000	
	23	-	WATER	70.00	-	17.5105	
	24	-	WATER	91.98	-	17.5105	
	25	356.89	0.8405	345.00	771.32	1.0000	
	26	268.83	0.8405	302.23	750.21	1.0000	
	29	- 121.72	0.8405	125.03	385.86	1.0000	
	30	263.83	0.8405	345.00	779.65	1.0000	
	36	123.02	0.8405	236.00	724.88	1.0000	
	38	122.02	0.8405	160.00	472.94	1.0000	
	60	376.89	0.8405	155.00	89.33	1.0000	
	61	376.89	0.8405	155.00	89.33	.4094	
	62	376.89	0.8405	155.00	89.33	.5906	
	63	366.89	0.8405	231.00	515.87	.4094	
	64	366.89	0.8405	231.00	515.87	.5906	
	66	366.89	0.8405	231.00	515.87	1.0000	
	68	361.89	0.8405	296.73	735.95	1.0000	

Dhla= 223.05
 G1= 2.0727 G1m= 2.1307
 ----- G1 loop: Ta= 245.25
 Pa= 365.89 Ks125=0.8405 Ta= 245.25 Ha= 557.11
 Dhla= 243.56
 G1= 2.1307 G1m= 2.1765
 ----- G1 loop: Ta= 238.12
 Pa= 366.39 Ks125=0.8405 Ta= 238.12 Ha= 536.46
 Dhla= 264.31
 G1= 2.1765 G1m= 2.2171
 ----- G1 loop: Ta= 231.00
 Pa= 366.89 Ks125=0.8405 Ta= 231.00 Ha= 515.67
 Dhla= 284.89
 G1= 2.2171 G1m= 2.2525

End of G1 loop: G1= 2.2525
 T3= 236.00

P38 = 122.02 Ks139 =0.8405 T38 = 160.00 H38 = 472.84 G38 = 1.2000

P29 = 121.72 Ks129 =0.8405 T29 = 125.03 H29 = 386.85 G29 = 1.0000

Dha66= 426.56 Dh166= 426.54 Difference= .01
 T9= 160.00 Dt4= 0.00 T4= 160.00

End of ksi 14 loop

P68 = 361.89 Ks156 =0.8405 T68 = 296.73 H68 = 735.95 G68 = 1.0000
 G6= 1.0232 G5= 1.2293 G64= .5906 G63= .4064

=====
 Sum of turbine enthalpy drops= 75.89 Turbine work= 73.99
 Hin= 459.51
 Dhp= 1.20
 Dup= 1.51 Wnet= 72.49

 Nte 15.77 %

Weight flow 25 195338.55

Net Output 4148.73KWe

Second law limit 24.43 %
 Second law efficiency 64.58 %
 =====

17 Jan 1989 12:43:06

=====						
!	#	P(psia)	X	I deg F	H (Btu/lb)	G
=====						
	1	-	BRINE	360.00	-	2.4876
	2	-	BRINE	330.75	-	2.4876
	3	-	BRINE	235.70	-	2.4876
	4	-	BRINE	170.11	-	2.4876
	5	-	BRINE	360.00	-	1.0300
	6	-	BRINE	360.00	-	1.4576
	7	-	BRINE	330.75	-	1.0300
	8	-	BRINE	330.75	-	1.4576
	9	-	BRINE	170.00	-	2.4876
	14	121.71	0.8420	77.00	1.46	1.0000
	21	440.02	0.8420	77.00	2.90	1.0000
	23	-	WATER	70.00	-	17.6727
	24	-	WATER	92.51	-	17.6727
	25	410.02	0.8420	345.00	765.80	1.0000
	26	270.10	0.8420	281.88	735.35	1.0000
	29	122.01	0.8420	129.30	399.35	1.0000
	30	265.10	0.8420	345.00	778.84	1.0000
	36	123.31	0.8420	235.70	723.91	1.0000
	38	122.31	0.8420	170.00	498.00	1.0000
	60	430.02	0.8420	165.00	101.56	1.0000
	61	430.02	0.8420	165.00	101.56	.4242
	62	430.02	0.8420	165.00	101.56	.5758
	63	420.02	0.8420	230.70	493.90	.4242
	64	420.02	0.8420	230.70	493.90	.5758
	66	420.02	0.8420	230.70	493.90	1.0000
	68	415.02	0.8420	304.47	735.07	1.0000

----- G1 loop: Ta= 251.17
Pa= 418.02 Ks125=0.8420 Ta= 251.17 Ha= 553.55
Dh1a= 255.73
G1= 2.3931 G1m= 2.4148
----- G1 loop: Ta= 246.06
Pa= 418.52 Ks125=0.8420 Ta= 246.06 Ha= 539.68
Dh1a= 270.60
G1= 2.4148 G1m= 2.4351
----- G1 loop: Ta= 240.94
Pa= 419.02 Ks125=0.8420 Ta= 240.94 Ha= 523.77
Dh1a= 285.52
G1= 2.4351 G1m= 2.4541
----- G1 loop: Ta= 235.82
Pa= 419.52 Ks125=0.8420 Ta= 235.82 Ha= 508.83
Dh1a= 300.45
G1= 2.4541 G1m= 2.4716
----- G1 loop: Ta= 230.70
Pa= 420.02 Ks125=0.8420 Ta= 230.70 Ha= 493.90
Dh1a= 315.39
G1= 2.4716 G1m= 2.4876

End of G1 loop: G1= 2.4876
T3= 235.70

P38 = 122.31 Ks138 =0.8420 T38 = 170.00 H38 = 499.00 G38 = 1.0000

P29 = 122.01 Ks129 =0.8420 T29 = 126.30 H29 = 389.35 G29 = 1.0000

Dha66= 392.62 Dh166= 392.34 Difference= .28
T9= 170.00 Dt4= .11 T4= 170.11

End of ksi 14 loop

P68 = 415.02 Ks168 =0.8420 T68 = 334.47 H68 = 735.07 G68 = 1.0000
G6= 1.4576 G5= 1.0300 G64= .5758 G63= .4242

=====

Sum of turbine enthalpy drops= 85.38 Turbine work= 83.24
Hin= 482.10
Dhp= 1.45
Dwp= 1.81 Wnet= 81.44

Nte= 16.89 %

Weight flow 25 176876.82

Net Output 4220.43KWe

Second law limit 24.97 %

Second law efficiency 67.66 %

=====

12 Jan 1989 14:56:45

!	#	P(psia)	X	T deg F	H (Btu/lb)	G	!
	1	-	BRINE	367.00	-	2.6269	
	2	-	BRINE	333.96	-	2.6269	
	3	-	BRINE	222.26	-	2.6269	
	4	-	BRINE	170.22	-	2.6269	
	5	-	BRINE	367.00	-	.9851	
	6	-	BRINE	367.00	-	1.6418	
	7	-	BRINE	333.96	-	.9851	
	8	-	BRINE	333.96	-	1.6418	
	9	-	BRINE	170.00	-	2.6269	
	14	89.30	0.8305	60.00	-20.33	1.0000	
	21	432.44	0.8305	60.00	-18.78	1.0000	
	23	-	WATER	53.00	-	16.8761	
	24	-	WATER	78.07	-	16.8761	
	25	402.44	0.8305	352.00	776.80	1.0000	
	26	232.20	0.8305	273.78	736.73	1.0000	
	29	89.60	0.8305	121.93	402.82	1.0000	
	30	227.20	0.8305	352.00	792.06	1.0000	
	36	90.90	0.8305	222.26	726.46	1.0000	
	38	89.90	0.8305	170.00	519.86	1.0000	
	60	422.44	0.8305	165.00	98.25	1.0000	
	61	422.44	0.8305	165.00	98.25	.4029	
	62	422.44	0.8305	165.00	98.25	.5971	
	63	412.44	0.8305	217.26	444.29	.4029	
	64	412.44	0.8305	217.26	444.29	.5971	
	66	412.44	0.8305	217.26	444.29	1.0000	
	68	407.44	0.8305	307.75	743.60	1.0000	

Sum of turbine enthalpy drops = 105.66 Btu/lb

Turbine net output = 103.02 Btu/lb

Heat input = 527.85 Btu/lb

Pump heat input = 1.55 Btu/lb

Pump power = 1.93 Btu/lb

Net power output = 101.09 Btu/lb

Net thermal efficiency = 19.15%

Geothermal fluid weight flow = 440000 lbs/hr

Working fluid weight flow 25 = 167496.06 lbs/hr

Net output 4960.85 KWe

Second law efficiency limit	27.48%
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Second law efficiency	69.68%
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APPENDIX B
STATE POINTS FOR HCRP REGENERATIVE CYCLE

CASE I

*****INPUT DATA CYCLE CALC*****

FLUID-----IC4/C7 94/6 01-04-1980 02:32:37

TG1= 360 TF1= 312.29 PF1= 600 GPP= 5 TG2LMT= 160
 TF4= 80 PF4= 51.77 TC1= 70 CPP= 5 TWB= 60.5
 MAXS= 0 S= 0 OPT2= 0 RPP= 9
 ET= .86 EEP= .8

*****WITHOUT RECUPERATOR*****
 *****PERFORMANCE IN W-H/LB GEOFLUID*****

	PINCH POINT	GEO OUT LIMIT
GROSS TURBINE PWR=	12.24193	10.67085
PUMP PWR=	1.217902	1.061602
WET COOLING TOWER=	.9514036	.8293042
NET POWER WITH WET COOLING=	10.07262	8.779941
SECOND LAW EFF. (KALINA)	.7566579	.6595513

WORKING FLUID/GEOFLUID WT. FLOW RATIO BASED ON PINCH= 1.160506
 WORKING FLUID/GEOFLUID WT FLOW RATIO BASED ON GEOFLUID LIMIT= 1.011571
 COOLING WATER/WORKING FLUID WT FLOW RATIO= 11.17885

*****SYSTEM STATE POINTS*****

	P (PSIA)	T (F)	H (BTU/LB)	S (BTU/LB R)	B (BTU/LB)
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*****GEOFLUID*****

STATE 1 HEATER INLET	0.0	360.0	332.35	0.5162	64.51
STATE 2 HEATER OUTLET	0.0	129.9	97.87	0.1814	4.28
STATE 3 HEATER PINCH PT	0.0	265.0	233.87	0.3889	32.26
STATE 2L HEATER OUTLET (GEOFLUID TEMP LIMIT)	0.0	160.0	127.96	0.2313	8.39

*****WORKING FLUID*****

STATE 1 TURBINE INLET	600.0	312.3	207.73	0.8389	50.08
STATE 2 TURBINE OUTLET	51.8	144.4	171.73	0.8486	9.01
STATE 3 CONDENSER INLET	51.8	144.4	171.73	0.8486	278.99
STATE 4 CONDENSER OUTLET	51.8	80.0	2.10	0.5396	0.23
STATE 5 FEED PUMP OUTLET	600.0	84.5	5.68	0.5409	3.12
STATE 6 HEATER INLET	600.0	84.5	5.68	0.5409	278.99
STATE 7 HEATER PINCH PT	600.0	260.0	122.87	0.7259	24.03
STATE 9 COND PINCH PT	51.8	84.0	102.76	0.7256	4.07

*****COOLANT*****

STATE 1 COND INLET	0.0	70.0	38.09	0.0744	0.17
STATE 2 COND OUTLET	0.0	85.2	53.26	0.1027	0.62
STATE 3 PINCH POINT	0.0	79.0	47.09	0.0914	0.32

*****WITH RECUPERATOR*****
 *****PERFORMANCE IN W-H/LB GEOFLUID*****

	PINCH POINT	GEO OUT LIMIT
GROSS TURBINE PWR=	12.24193	12.23937
PUMP PWR=	1.217902	1.217648
WET COOLING TOWER=	.9285199	.9283259
NET POWER WITH WET COOLING=	10.09551	10.0934
SECOND LAW EFF. (KALINA)	.7517501	.751593

WORKING FLUID/GEOFLUID WT. FLOW RATIO BASED ON PINCH= 1.160506
 WORKING FLUID/GEOFLUID WT FLOW RATIO BASED ON GEOFLUID LIMIT= 1.16Q263
 COOLING WATER/WORKING FLUID WT FLOW RATIO= 11.17885

*****SYSTEM STATE POINTS*****

	P (PSIA)	T (F)	H (BTU/LB)	S (BTU/LB R)	B (BTU/LB)
*****GEOFLUID*****					
STATE 1 HEATER INLET	0.0	360.0	332.35	0.5162	64.51
STATE 2 HEATER OUTLET	0.0	160.0	127.92	0.2312	8.39
STATE 3 HEATER PINCH PT	0.0	265.0	233.87	0.3889	32.26
STATE 2L HEATER OUTLET (GEOFLUID TEMP LIMIT)	0.0	160.0	127.96	0.2313	8.39
*****WORKING FLUID*****					
STATE 1 TURBINE INLET	600.0	312.3	207.73	0.8389	50.08
STATE 2 TURBINE OUTLET	51.8	144.4	171.73	0.8486	9.01
STATE 3 CONDENSER INLET	51.8	102.4	145.84	0.8039	6.39
STATE 4 CONDENSER OUTLET	51.8	80.0	2.10	0.5396	0.23
STATE 5 FEED PUMP OUTLET	600.0	84.5	5.68	0.5409	3.12
STATE 6 HEATER INLET	600.0	128.4	31.58	0.5866	5.24
STATE 7 HEATER PINCH PT	600.0	260.0	122.87	0.7259	24.03
STATE 8 RECUP PINCH PT	51.8	116.6	159.28	0.8276	7.50
STATE 9 COND PINCH PT	51.8	84.0	102.76	0.7256	4.07
STATE 10 RECUP PINCH PT	600.0	80.4	19.12	0.5711	0.86
*****COOLANT*****					
STATE 1 COND INLET	0.0	70.0	38.09	0.0744	0.17
STATE 2 COND OUTLET	0.0	82.9	50.94	0.0985	0.49
STATE 3 PINCH POINT	0.0	79.0	47.09	0.0914	0.32

*****INPUT DATA CYCLE CALC*****

FLUID-----IC4/C7 94/6 01-04-1980 02:34:01

TG1= 360	TF1= 316.11	PF1= 600	GPP= 5	TG2LMT= 160
TF4= 80	PF4= 51.77	TC1= 70	CPP= 5	TWB= 60.5
MAXS= 0	S= 0	OPT2= 0	RPP= 9	
ET= .86	EFP= .8			

*****WITHOUT RECUPERATOR*****

*****PERFORMANCE IN W-H/LB GEOFLUID*****

	PINCH POINT	GEO OUT LIMIT
GROSS TURBINE PWR=	11.86779	10.69457
PUMP PWR=	1.152524	1.038589
WET COOLING TOWER=	.90334	.8140382
NET POWER WITH WET COOLING=	9.811926	8.841946
SECOND LAW EFF. (KALINA)	.7361332	.663361

WORKING FLUID/GEOFLUID WT. FLOW RATIO BASED ON PINCH= 1.098209
 WORKING FLUID/GEOFLUID WT FLOW RATIO BASED ON GEOFLUID LIMIT= .9896427
 COOLING WATER/WORKING FLUID WT FLOW RATIO= 11.17885

*****SYSTEM STATE POINTS*****

	P (PSIA)	T (F)	H (BTU/LB)	S (BTU/LB R)	B (BTU/LB)
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*****GEOFLUID*****

STATE 1 HEATER INLET	0.0	360.0	332.35	0.5162	64.51
STATE 2 HEATER OUTLET	0.0	137.6	105.54	0.1944	5.18
STATE 3 HEATER PINCH PT	0.0	275.0	244.08	0.4029	35.20
STATE 2L HEATER OUTLET (GEOFLUID TEMP LIMIT)	0.0	160.0	127.96	0.2313	8.39

*****WORKING FLUID*****

STATE 1 TURBINE INLET	600.0	316.1	212.21	0.8447	51.54
STATE 2 TURBINE OUTLET	51.8	152.2	175.33	0.8546	9.51
STATE 3 CONDENSER INLET	51.8	152.2	175.33	0.8546	278.99
STATE 4 CONDENSER OUTLET	51.8	80.0	2.10	0.5396	0.23
STATE 5 FEED PUMP OUTLET	600.0	84.5	5.68	0.5409	3.12
STATE 6 HEATER INLET	600.0	84.5	5.68	0.5409	278.99
STATE 7 HEATER PINCH PT	600.0	270.0	131.83	0.7383	26.53
STATE 9 COND PINCH PT	51.8	84.0	102.76	0.7256	4.07

*****COOLANT*****

STATE 1 COND INLET	0.0	70.0	38.09	0.0744	0.17
STATE 2 COND OUTLET	0.0	85.5	53.58	0.1033	0.63
STATE 3 PINCH POINT	0.0	79.0	47.09	0.0914	0.32

*****WITH RECUPERATOR*****
 *****PERFORMANCE IN W-H/LB GEOFLUID*****

	PINCH POINT	GEO OUT LIMIT
GROSS TURBINE PWR=	11.86779	0
PUMP PWR=	1.152524	0
WET COOLING TOWER=	.8786762	0
NET POWER WITH WET COOLING=	9.83659	0
SECOND LAW EFF. (KALINA)	.7306952	0

WORKING FLUID/GEOFLUID WT. FLOW RATIO BASED ON PINCH= 1.098209
 OVERFLOW DUE TO ZERO FLO RATIO
 WORKING FLUID/GEOFLUID WT FLOW RATIO BASED ON GEOFLUID LIMIT= 0
 COOLING WATER/WORKING FLUID WT FLOW RATIO= 11.17885

*****SYSTEM STATE POINTS*****

	P (PSIA)	T (F)	H (BTU/LB)	S (BTU/LB R)	B (BTU/LB)
*****GEOFLUID*****					
STATE 1 HEATER INLET	0.0	360.0	332.35	0.5162	64.51
STATE 2 HEATER OUTLET	0.0	170.0	137.93	0.2471	10.12
STATE 3 HEATER PINCH PT	0.0	275.0	244.08	0.4029	35.20
STATE 2L HEATER OUTLET (GEOFLUID TEMP LIMIT)	0.0	160.0	127.96	0.2313	8.39

*****WORKING FLUID*****					
STATE 1 TURBINE INLET	600.0	316.1	212.21	0.8447	51.54
STATE 2 TURBINE OUTLET	51.8	152.2	175.33	0.8546	9.51
STATE 3 CONDENSER INLET	51.8	102.4	145.84	0.8039	6.39
STATE 4 CONDENSER OUTLET	51.8	80.0	2.10	0.5396	0.23
STATE 5 FEED PUMP OUTLET	600.0	84.5	5.68	0.5409	3.12
STATE 6 HEATER INLET	600.0	134.3	35.17	0.5927	5.69
STATE 7 HEATER PINCH PT	600.0	270.0	131.83	0.7383	26.53
STATE 8 RECUP PINCH PT	51.8	116.6	159.28	0.8276	7.50
STATE 9 COND PINCH PT	51.8	84.0	102.76	0.7256	4.07
STATE 10 RECUP PINCH PT	600.0	80.4	19.12	0.5711	0.86

*****COOLANT*****					
STATE 1 COND INLET	0.0	70.0	38.09	0.0744	0.17
STATE 2 COND OUTLET	0.0	82.9	50.94	0.0985	0.49
STATE 3 PINCH POINT	0.0	79.0	47.09	0.0914	0.32

CASE II

*****INPUT DATA CYCLE CALC*****

FLUID-----IC4/C7 94/6 01-04-1980 02:35:46

TG1= 367	TF1= 323.23	PF1= 600	GPP= 5	TG2LMT= 170
TF4= 65	FF4= 40.43	TC1= 53	CPP= 5	TWB= 60.5
MAXS= 0	S= 0	OPT2= 0	RPP= 9	
ET= .86	EFP= .8			

*****WITHOUT RECUPERATOR*****
*****PERFORMANCE IN W-H/LB GEOFLUID*****

	PINCH POINT	GEO OUT LIMIT
GROSS TURBINE PWR=	13.51024	11.25871
PUMP PWR=	1.154581	.9621659
WET COOLING TOWER=	.7915991	.6596765
NET POWER WITH WET COOLING=	11.56406	9.636871
SECOND LAW EFF. (KALINA)	.7691621	.6409786

WORKING FLUID/GEOFLUID WT. FLOW RATIO BASED ON PINCH= 1.087417
WORKING FLUID/GEOFLUID WT FLOW RATIO BASED ON GEOFLUID LIMIT= .9061951
COOLING WATER/WORKING FLUID WT FLOW RATIO= 9.520578

*****SYSTEM STATE POINTS*****

	P (PSIA)	T (F)	H (BTU/LB)	S (BTU/LB R)	B (BTU/LB)
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*****GEOFLUID*****

STATE 1 HEATER INLET	0.0	367.0	339.79	0.5251	67.31
STATE 2 HEATER OUTLET	0.0	129.6	97.61	0.1810	4.25
STATE 3 HEATER PINCH PT	0.0	275.0	244.08	0.4029	35.20
STATE 2L HEATER OUTLET	0.0	170.0	137.98	0.2472	10.13
(GEOFLUID TEMP LIMIT)					

*****WORKING FLUID*****

STATE 1 TURBINE INLET	600.0	323.2	219.85	0.8545	54.10
STATE 2 TURBINE OUTLET	40.4	154.8	177.45	0.8659	5.78
STATE 3 CONDENSER INLET	40.4	154.8	177.45	0.8659	279.01
STATE 4 CONDENSER OUTLET	40.4	65.0	-6.48	0.5236	-0.00
STATE 5 FEED PUMP OUTLET	600.0	69.4	-2.86	0.5250	2.91
STATE 6 HEATER INLET	600.0	69.4	-2.86	0.5250	279.01
STATE 7 HEATER PINCH PT	600.0	270.0	131.83	0.7383	26.56
STATE 9 COND PINCH PT	40.4	68.0	88.94	0.7051	0.95

*****COOLANT*****

STATE 1 COND INLET	0.0	53.0	21.06	0.0418	0.14
STATE 2 COND OUTLET	0.0	72.3	40.38	0.0788	0.21
STATE 3 PINCH POINT	0.0	63.0	31.08	0.0612	0.04

*****WITH RECUPERATOR*****
 *****PERFORMANCE IN W-H/LB GEOFLUID*****

	PINCH POINT.	GEO OUT LIMIT
GROSS TURBINE PWR=	13.51024	0
PUMP PWR=	1.154581	0
WET COOLING TOWER=	.7608641	0
NET POWER WITH WET COOLING=	11.5948	0
SECOND LAW EFF. (KALINA)	.7617743	0

WORKING FLUID/GEOFLUID WT. FLOW RATIO BASED ON PINCH= 1.087417
 OVERFLOW DUE TO ZERO FLO RATIO
 WORKING FLUID/GEOFLUID WT FLOW RATIO BASED ON GEOFLUID LIMIT= 0
 COOLING WATER/WORKING FLUID WT FLOW RATIO= 9.520578

*****SYSTEM STATE POINTS*****

	P (PSIA)	T (F)	H (BTU/LB)	S (BTU/LB R)	B (BTU/LB)
*****GEOFLUID*****					
STATE 1 HEATER INLET	0.0	367.0	339.79	0.5251	67.31
STATE 2 HEATER OUTLET	0.0	170.1	138.08	0.2474	10.15
STATE 3 HEATER PINCH PT	0.0	275.0	244.08	0.4029	35.20
STATE 2L HEATER OUTLET (GEOFLUID TEMP LIMIT)	0.0	170.0	137.98	0.2472	10.13
*****WORKING FLUID*****					
STATE 1 TURBINE INLET	600.0	323.2	219.85	0.8545	54.10
STATE 2 TURBINE OUTLET	40.4	154.8	177.45	0.8659	5.78
STATE 3 CONDENSER INLET	40.4	87.7	140.24	0.8012	2.25
STATE 4 CONDENSER OUTLET	40.4	65.0	-6.48	0.5236	-0.00
STATE 5 FEED PUMP OUTLET	600.0	69.4	-2.86	0.5250	2.91
STATE 6 HEATER INLET	600.0	133.0	34.35	0.5913	5.61
STATE 7 HEATER PINCH PT	600.0	270.0	131.83	0.7383	26.56
STATE 8 RECUP PINCH PT	40.4	104.1	154.92	0.8276	3.17
STATE 9 COND PINCH PT	40.4	68.0	88.94	0.7051	0.95
STATE 10 RECUP PINCH PT	600.0	65.3	11.82	0.5585	0.15
*****COOLANT*****					
STATE 1 COND INLET	0.0	53.0	21.06	0.0418	0.14
STATE 2 COND OUTLET	0.0	68.4	36.47	0.0714	0.14
STATE 3 PINCH POINT	0.0	63.0	31.08	0.0612	0.04

Case I $T_{\text{geo out}}=160^\circ \text{ F}$, $T_{\text{cond out}}=80^\circ \text{ F}$, $P_H=600 \text{ psia}$, $P_C=51.8 \text{ psia}$

Turbine calculation

$$P_2 = 51.8 + 2 = 53.8 \quad h_{2s}=166.30 \text{ Btu/lb} \quad h_2=172.10 \text{ Btu/lb.}$$

$$\frac{\dot{W}_{T_{\Delta P}}}{\dot{W}_{T_{\Delta P=0}}} = \frac{207.73 - 172.10}{207.73 - 171.73} = 0.9897$$

Corrected Power Calculation

$$\dot{W}_{T_{\Delta P}} = (.9897)(12.2419) = 12.1161 \text{ whr/lb geo}$$

$$\dot{W}_{P_{\Delta P}} = \left[\frac{600-51.8+30}{600-51.8} \right] (1.2179) = 1.2845 \text{ whr/lb geo}$$

$$\dot{Q}_H = \frac{332.35-127.92}{3.413} = 59.8974 \text{ whr/lb}$$

$$\dot{Q}_{R_{\Delta P=0}} = 59.8974 - 12.2419 + 1.2179 = 48.8735$$

$$Q_{R_{\Delta P}} = 59.8974 - 12.1161 + 1.2845 = 49.0659$$

$$\dot{W}_{CT_{\Delta P}} = \frac{49.0659}{48.8735} (.9285) = 0.9322 \text{ whr/lb geo}$$

$$\dot{W}_{\text{Net Plant}} = 12.1161 (.975) - 1.2845 = 10.5287 \text{ whr/lb geo}$$

$$\dot{W}_{\text{Net}} = 10.5287 - .9322 = \underline{9.5964} \text{ whr/lb geo}$$

Second Law Efficiency

$$T_{o_{\text{eff}}} = 530 \left[2 - \frac{530 \text{ hr} \frac{542.9}{530}}{12.9} \right] = 536.3^\circ \text{R}$$

$$\Delta_{\text{avail geo}} = \frac{(332.35-127.92) - 536.3(.5162-.2312)}{3.413}$$

$$= 15.1141 \text{ whr/lb geo}$$

$$\eta_{2\text{nd law}} = \frac{10.8316}{15.1141} = 0.6966$$

Case IA $T_{\text{geo out}}=170^\circ \text{ F}$, $T_{\text{cond}}=80^\circ \text{ F}$, $P_H=600 \text{ psia}$, $P_C=51.8 \text{ psia}$

Turbine calculation

$$P_2 = 51.8 + 2 = 53.8 \quad h_{2s}=169.76 \text{ Btu/lb} \quad h_2=175.70 \text{ Btu/lb.}$$

$$\frac{\dot{W}_{T_{\Delta P}}}{\dot{W}_{T_{\Delta P=0}}} = \frac{212.21 - 175.70}{212.21 - 175.33} = 0.9900$$

$$\dot{W}_{T_{\Delta P=0}}$$

Corrected Power Calculation

$$\dot{W}_{T_{\Delta P}} = (.9900)(11.8678) = 11.7487 \text{ whr/lb geo}$$

$$\dot{W}_{P_{\Delta P}} = \left[\frac{600-51.8+30}{600-51.8} \right] (1.1525) = 1.2156 \text{ whr/lb geo}$$

$$\dot{Q}_H = \frac{332.35 - 137.93}{3.413} = 56.9645 \text{ whr/lb}$$

$$\dot{Q}_{R_{\text{No}\Delta P}} = 56.9645 - 11.8678 + 1.1525 = 46.2492$$

$$\dot{Q}_{R_{\Delta P}} = 56.9645 - 11.7487 + 1.2156 = 46.4314$$

$$\dot{W}_{CT_{\Delta P}} = \frac{46.4314}{46.2492} (.8787) = 0.8822 \text{ whr/lb geo}$$

$$\dot{W}_{\text{Net Plant}} = 11.7487 (.975) - 1.2156 = 10.2394 \text{ whr/lb geo}$$

$$\dot{W}_{\text{Net}} = 10.2394 - .8822 = \underline{9.3572} \text{ whr/lb geo}$$

Second Law Efficiency

$$T_{\text{oeff}} = 536.3^\circ \text{ R}$$

$$\Delta_{\text{avail geo}} = \frac{(332.35 - 137.93) - 536.3(.5162 - .2471)}{3.413}$$

$$= 14.6797 \text{ whr/lb geo}$$

$$\eta_{2\text{nd law}} = \frac{10.5331}{14.6797} = 0.6975$$

Case II $T_{\text{geo out}}=170^\circ \text{ F}$, $T_{\text{cond}}=65^\circ \text{ F}$, $P_H=600 \text{ psia}$, $P_C=40.4 \text{ psia}$

Turbine calculation

$$P_2 = 40.4 + 2 = 42.4 \quad h_{2s}=171.14 \text{ Btu/lb} \quad h_2=177.95 \text{ Btu/lb.}$$

$$\frac{\dot{W}_T}{\Delta P} = \frac{219.83 - 177.95}{219.83 - 177.43} = 0.9877$$

$$\dot{W}_T \Delta P=0$$

Corrected Power Calculation

$$\dot{W}_{T\Delta P} = (.9877)(13.51265) = 13.3459 \text{ whr/lb geo}$$

$$\dot{W}_{P\Delta P} = \left[\frac{600-40.4+30}{600-40.4} \right] (1.154957) = 1.2169 \text{ whr/lb geo}$$

$$\dot{Q}_H = \frac{339.79 - 138.02}{3.413} = 59.1181 \text{ whr/lb}$$

$$\dot{Q}_{R_{\text{No}\Delta P}} = 59.1181 - 13.5127 + 1.1550 = 46.7604$$

$$\dot{Q}_{R\Delta P} = 59.1181 - 13.3459 + 1.2169 = 46.9891$$

$$\dot{W}_{CT\Delta P} = \frac{46.9891}{46.7604} (.76111) = 0.7648 \text{ whr/lb geo}$$

$$\dot{W}_{\text{Net Plant}} = 13.3459 (.975) - 1.2169 = 11.7954 \text{ whr/lb geo}$$

$$\dot{W}_{\text{Net}} = 11.7954 - .7648 = \underline{11.0306} \text{ whr/lb geo}$$

Second Law Efficiency

$$T_{\text{oeff}} = 513 \left[2 - \frac{513 \text{ hr} \frac{528.4}{513}}{15.4} \right] = 520.5^\circ \text{ R}$$

$$\Delta_{\text{avail geo}} = \frac{(339.79 - 138.02) - 520.5(.5251 - .2473)}{3.413}$$

$$= 16.7522 \text{ whr/lb geo}$$

$$\eta_{2\text{nd law}} = \frac{1}{16.1290} = 0.7313$$