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OVERVIEW OF RECENT, SUPERCRITICAL BINARY GEOTHERMAL CYCLE EXPERIMENTS FROM THE HEAT CYCLE RESEARCH PROGRAM

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ABSTRACT

The Heat Cycle Research Program, which is being conducted for the Department of Energy, has as its objective the development of the technology for effecting improved utilization of moderate temperature geothermal resources. Testing at the Heat Cycle Research Facility located at the DOE Geothermal Test Facility East Mesa, California involves supercritical vaporization and counterflow in-tube condensing in an organic Rankine cycle. Results of the experiments are given for both pure and mixed-hydrocarbon working fluids. The heater and condenser behavior predicted by the Heat Transfer Research, Inc. computer codes used for correlation of the data was in excellent agreement with experimental results. A special series of tests, conducted with propane and up to approximately 40% isopentane concentration, indicated that a close approach to "integral" condensation was occurring in the vertically-oriented condenser. Preliminary results of tests in which the turbine expansion "passed through the two-phase region" did not indicate efficiency degradation assignable to these metastable expansion processes.

BACKGROUND

The purpose of the Heat Cycle Research Program is to develop technology which will result in more effective utilization of moderate temperature geothermal resources; a major emphasis of the program has been directed toward binary cycle technology. The overall program has been discussed briefly in (1*). As indicated in that reference several concepts have been investigated analytically in earlier program efforts which have shown the potential for effecting significant performance gains for binary plants. Use of non-adjacent hydrocarbon mixtures for working fluids, which are vaporized at supercritical pressures, and a counterflow in-tube condenser to provide a close approach to integral condensation, are two such concepts. (Integral condensation refers to the maintaining of thermal equilibrium between phases during condensation.) Additional performance gains were predicted through use of turbine exhaust recuperation, and through modification of turbine inlet state points to achieve super-saturated-vapor turbine-expansion processes. These advances, in total, were projected to increase present levels of net plant geofluid effectiveness (Wh/lbm geofluid) by as much as 28% using 360°F hydrothermal resources, and to double the use of moderate-temperature geothermal energy. An overview of the recent experiments for confirming the assumptions made in the performance projections, and for developing the technology needed to achieve counterflow integral condensation, is given in the present paper. Initial results of these experiments were presented in (2), and a more detailed discussion of the current results is given in (3).

*Numbers in parentheses designate references at the end of the paper.

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EXPERIMENTAL APPROACH

To accomplish the objective of developing technology for advanced binary geothermal plants, a number of Rankine-cycle experiments were conducted using working fluids consisting of both pure and mixed hydrocarbons of the propane-isopentane and isobutane-hexane families at nominally 0, 5, 10% isopentane or hexane, by mass. Vaporization of the working fluids at supercritical pressures was investigated, as was condensing of the pure and mixed-hydrocarbon vapors in a vertical counterflow in-tube condenser. The test parameters varied were heater pressure and working-fluid, geofluid, and cooling-water flows. A series of tests was run with special propane-isopentane mixtures with isopentane mass factions of up to 40% to investigate the departure from integral condensing exhibited by the condenser. Finally, the effect of allowing turbine expansion processes to cross the saturation line and "pass through the two-phase region" was studied during several tests using isobutane-hexane working fluids.

The experiments were conducted in the Heat Cycle Research Facility (HCRF), in which the energy from the geothermal fluid is transferred to a secondary working fluid, which is in turn expanded through a turbine driving an electrical generator. The

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facility is currently located at the DOE Geothermal Test Facility (GTF) at East Mesa in the Imperial Valley of Southern California. The HCRF in its current configuration is shown schematically in Figure 1; a photograph is included in (1). In this configuration the facility is operated as a supercritical cycle; that is, the working fluid vapor leaving the heaters is at a temperature and pressure higher than its critical point. As indicated in Figure 1, there are two supercritical heat exchangers, a preheater and a vapor generator, with which the energy from the geothermal fluid is used to vaporize a hydrocarbon working fluid. (The geothermal fluid was supplied from GTF Well 6-2, and entered the HCRF at a temperature between about 312 and 322°F.) The high-pressure working fluid vapor leaving the supercritical heaters can either be expanded through a turbine which drives an electrical generator or be expanded through a turbine bypass valve. The low-pressure vapor leaving the turbine or bypass valve is discharged to the condenser where it is desuperheated and condensed by rejecting heat to the GTF cooling-water system.

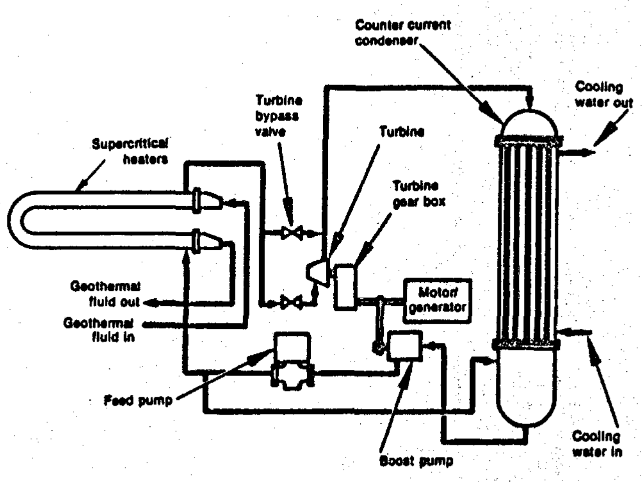

Figure 1. Schematic of the Heat Cycle Research Facility

HEATERS - The working-fluid heaters are arranged in a hairpin configuration with the preheater on the bottom and vapor generator on top, as sketched in Figure 1. The geothermal fluid and the working fluid have countercurrent flow paths through the heaters with the working fluid flow on the outside of the tubes. The preheater tube length is 28.21 feet (tubesheet face-to-face) with an outside shell diameter of 5.56 inches. It contains 27, 1/2-inch OD, 19 fins/inch, low-fin tubes made of admiralty brass having an outside-to-inside area ratio of 4.17. The vapor generator contains 39 of the same type of tube with a 29.21 foot length, and an outside shell diameter of 6.63 inches.

CONDENSER - The condenser in its present orientation is a vertical unit (as sketched in Figure 1) also having countercurrent flow paths. The condensation occurs on the inside of 1/2-inch OD, internally-finned tubes made of 90/10 cupro-nickel (Noranda forge fin No. 6, with ten straight longitudinal fins inside each tube giving an inside-to-outside area ratio of 1.3). The vessel is 18 inches in diameter and contains 419 of the tubes which have a length of 18.54 feet (tubesheet face-to-face). The cooling water enters the shell-side just above the lower tube-sheet and leaves the vessel just below the upper tubesheet.

TURBINE-GENERATOR - The turbine-generator assembly was designed and built by Barber-Nichols Engineering, and consists of an axial-flow impulse turbine driving an induction motor/ generator through a 6.135:1 speed-reduction gearbox. The generator's rated speed is 3600 rpm, and for the present nozzle area and turbine inlet state points, the turbine power produced is on the order of 40kW.

RESULTS

The analysis of the heat exchanger data from these experiments had a two-fold purpose. First, data were obtained and verified for the phenomena of supercritical heating in a finned tube heat exchanger and the condensation of hydrocarbon mixtures inside finned tubes. Second, the data were used to determine how well a heater or condenser similar to those tested could be designed using standard techniques. To achieve these purposes, it was decided to use the computer codes developed by Heat Transfer Research, Inc. (HTRI) to rate the exchangers, because these codes are commonly used for heat exchanger design, and a direct comparison between experiment and calculation gives a measure of how well the codes serve as design tools for this application. Several references containing general descriptions of methods used by HTRI in their development of the codes are listed as (4) through (8).

PROPERTIES - Working-fluid thermodynamic properties used in analysis of data were determined by a computer program named "EXCST" (9), developed by J. Ely at the National Bureau of Standards (NBS), which is based on an "Extended Corresponding States" theory. The NBS program resulted in more consistent energy balances than other properties codes available to us. Overall, these properties appeared satisfactory; no major deficiencies were detected.

HEATERS - For the heaters the HTRI computer code ST-4 MOD 5.4, the shell-and-tube code with no phase change, was used. This code uses average properties for a given exchanger; however, for our application, the thermodynamic and transport properties change quite rapidly with temperature, and very non-linear temperature distributions result within each heat exchanger along with large variations in the transport properties. In order to account for these variations the heater was divided into six separate increments and the vaporizer into nine.

The clean overall heat transfer coefficients and temperature differences were determined, and fouling values consistent with the measured temperatures for each heat exchanger were calculated. Fouling resistances for each heat exchanger were plotted as functions of time, and predicted temperature distributions through both heat exchangers were compared with measured temperatures. Agreement between the calculated and measured temperature distributions in the heater-vaporizer heat exchangers was quite good. Measured temperatures are within a few degrees F of the calculated values for all cases, as can be seen from the plots presented in Appendix B of (3).

Figure 2 shows the calculated fouling resistances referred to the inside area (geothermal-side), where the major portion of the fouling was expected. The time at which each fluid was tested is noted across the top of the plot. Boiling tests (slightly subcritical working fluid pressures) and those points with energy balances in error greater than 7% were omitted from the plot. The faired line is a least-squares fit of a quadratic curve for all of the data plotted between zero and about 320 days. The fouling resistance exhibits an increasing magnitude up to this point, at which time the heater tubes had shown evidence of some plugging (increased geofluid pressure drop); the heads were then removed, and the tubes unplugged. It is expected that the unplugging operation may have removed some of the fouling products, particularly in the preheater, and disturbed the fouling trend at that time. Therefore, the faired line in Figure 2 is shown only to the time of the unplugging. The fact that the calculated fouling factor is negative at early times in the test series is an indication that the computer code was slightly conservative, and under-predicted the heat transfer coefficient in the clean configuration. The clean overall heat transfer coefficient for these fluids at nominal conditions is between 100 and 140 Btu/hr ft^2°F, based on the finned outside area, and the code under-predicted the heat transfer by about 20% in the clean condition.

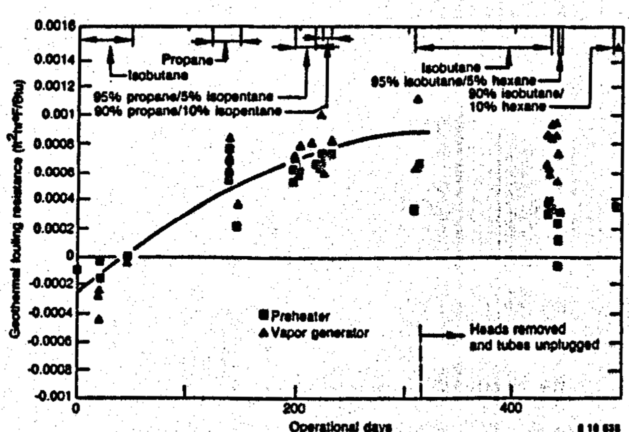

Figure 2. Fouling in Geothermal Heat Exchangers

CONDENSER

Temperature and Pressure Correlations - The condenser results were analyzed using CST-1 MOD 2.0 (the HTRI condenser code). The thermodynamic properties (from the EXCST code) used in the analysis assumed completely mixed phases during the condensation (integral condensation). This code treats variable working-fluid properties by dividing the condenser into twenty "constant-property" nodes. Corrections were made, however, to account for the presence of the internal fins on the working-fluid side of the tubes; the code normally assumes circular tubes. The relationship between measured condenser pressure and bubble point temperature contains some uncertainty in a number of items such as: pressure measurement accuracy, working fluid composition, accuracy of thermodynamic properties defining the saturation line, presence of noncondensibles, and the magnitude of condenser subcooling. Because of the combination of very close approach temperature differences between working fluid and cooling water in the condenser (≈ 1.5°F), and the uncertainties in the bubble-point temperature as a function of measured condenser pressure, it was found that measured temperatures rather than pressure, had to be used as code input quantities to best represent actual condenser conditions. The code was input assuming zero subcooling, zero pressure drop in the tubing, and with the measured working fluid inlet state-point and flow conditions. Measured cooling-water-inlet and outlet temperatures were input, and the code was used to calculate a bubble-point temperature for which the required condenser heat-transfer area equalled the actual surface area.

Values of calculated bubble-point temperature minus measured outlet working fluid temperature were plotted versus operational time for the working fluids tested as shown in Figure 3. No significant trends are indicated, although the plotted differences for the pure fluids seem to be slightly more positive than for the 90% - 10% mixtures. This trend will be carefully watched as more data are correlated for the next series of condenser tests. No evidence of fouling was shown. The correlation of Figure 3 indicates that if one designed a similar condenser using the HTRI code CST-1, the NBS thermophysical properties, and with the assumption that the fluid leaves the condenser at the bubble point, the resulting condenser would produce a bubble-point temperature which is, on the average, 0.4°F lower than the design value. The standard deviation of the

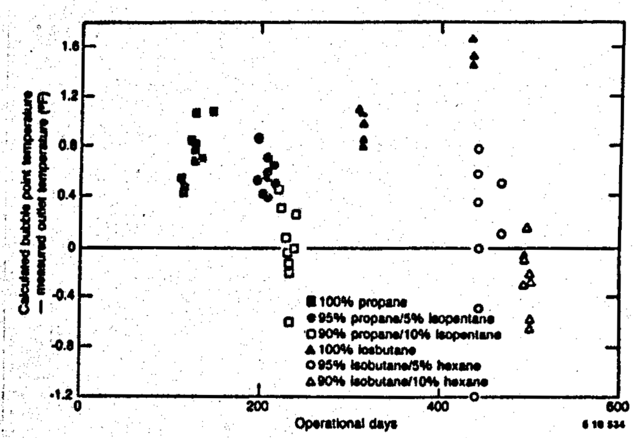

Figure 3. Correlation of Calculated Condensing Temperature

plotted points of Figure 3 from this average is 0.6°F. The code, therefore, provides a slightly conservative design, with the extent of conservatism comparable to the uncertainty in the experimental measurements.

A separate correlation for condenser pressure is shown in Figure 4 as the calculated bubble-point temperature minus the bubble-point temperature corresponding the actual condenser pressure, plotted as a function of operational time. Again, no significant trend is apparent. The average temperature difference for these plotted points is +1.4°F with a standard deviation from the average of 0.7°F. Therefore, a condenser designed using the HTRI code with the NBS properties would be expected to be slightly conservative; the condenser pressure produced would be equal to the bubble-point pressure for a temperature which is, on the average, 1.4°F lower than the design bubble-point temperature.

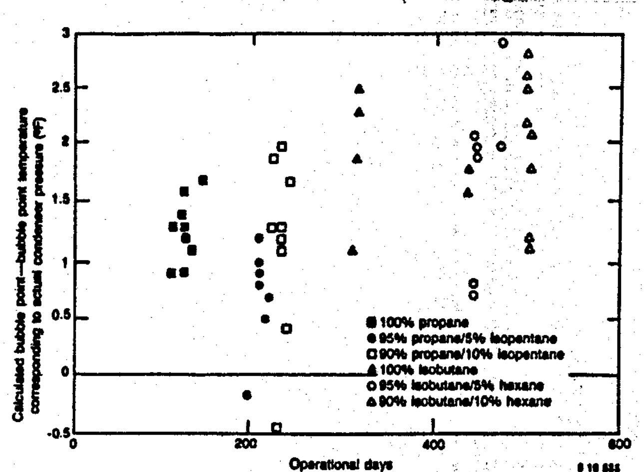

Figure 4. Correlation of Condenser Pressure

Temperature Distributions - Temperature distributions during condensing showed excellent agreement between calculated and measured values (within about 2°F). These results are given in (3). The average overall heat transfer coefficient for the 90% propane/10% isopentane mixture was 81.2 Btu/hr ft^2°F, and the average mean temperature difference was 9.2°F. For condensing pure propane the minimum approach or pinch-point temperature difference between working fluid and cooling water was much closer than for the mixture (1.5°F compared to 6°F) and created an average mean temperature difference of 7.4°F. The average overall heat transfer coefficient was 92.8 Btu/hr ft^2°F for the propane. Although the overall heat transfer coefficient was higher for the pure fluid by 14.3%, the lower mean temperature difference more than compensated for it, and more heat was transferred to the cooling water for the mixture (9% more) for this example. As a result, the bubble-point temperature approached the inlet cooling water temperature more closely for the mixture than for the pure propane. A similar trend was observed for the isobutane-hexane family of working fluids. It was concluded that despite the lower condensing heat transfer coefficients for the mixtures (about 30% lower for 90% propane -10% isopentane than for pure propane), mixtures can be as "easy or even easier" to condense than pure fluids, depending on the specific application.

Integral Condensation - As mentioned earlier, integral condensation refers to maintaining thermal equilibrium between phases during condensation, and results in the minimum achievable condenser pressure (maximum cycle performance) for a given working-fluid bubble-point temperature. As departure from integral condensation increases, the condenser pressure increases, (for a given condensing temperature), and in the limit reaches the condensing pressure of the light hydrocarbon constituent of the mixture. By increasing the isopentane content to 40% in the special series of propane-isopentane mixtures tested, the difference in bubble-point temperatures between the mixture and pure propane (at a given bubble-point pressure) was increased to more than 20°F, making any departure from integral condensation more observable. Preliminary results of these special tests, conducted with the condenser in the vertical attitude, are shown in Figure 5 as values of the bubble-point temperature (corresponding to the measured condenser pressure) minus the measured condensate outlet temperature (ordinate) plotted versus isopentane concentration. The plot shows generally negative values of the ordinate with no real trend evident. Any departure from integral condensation would tend to result in positive values of the ordinate; therefore, these preliminary results as well as similar comparisons for the nominal working fluids did not indicate departure from integral condensation.

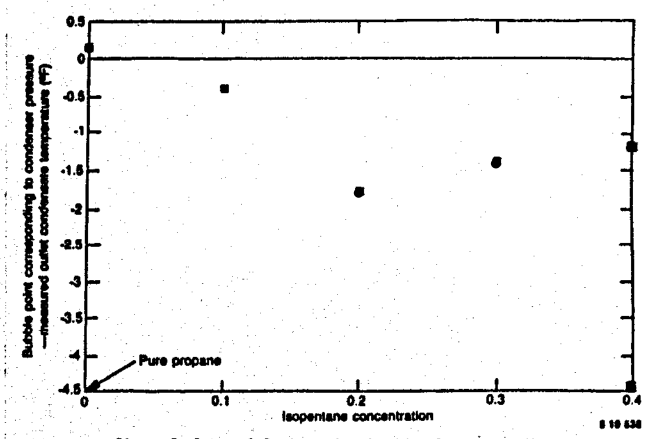

Figure 5. Integral Condensation for High-Isopentane Mixtures

TURBINE - A major objective of the turbine tests in this phase of experiments was to investigate the effect of expanding through the two-phase region (supersaturated expansions) on turbine efficiency. Three series of turbine-expansion tests with isobutane have been run, each at a constant pressure ratio. A fourth series of the same type was conducted with a 95% isobutane -5% hexane mixture. For the first series, two of the expansions remained in pure-vapor equilibrium states throughout;during the other four expansions the turbine inlet entropy was reduced such that the isentropic nozzle flow (outside of the boundary layer) reached equilibrium moisture conditions ranging from about 7 to 22%. Figure 6 is a schematic temperature - entropy diagram for isobutane showing the six turbine expansion processes (assuming metastable expansions). The vertical lines through the six inlet state points (at 560 psia) represent the potential-flow regions of the nozzle expansions to the exit pressure (60 psia), which correspond to the part of the nozzle flow most likely to undergo some condensation. Since the turbine is the impulse type, pressure in the blading is approximately constant; the constant pressure lines connecting the lower end of the vertical line and the lower end of the inclined line for each inlet condition, represent the irreversible flows through the blading. The inclined lines, themselves, only connect the turbine inlet and outlet state points.

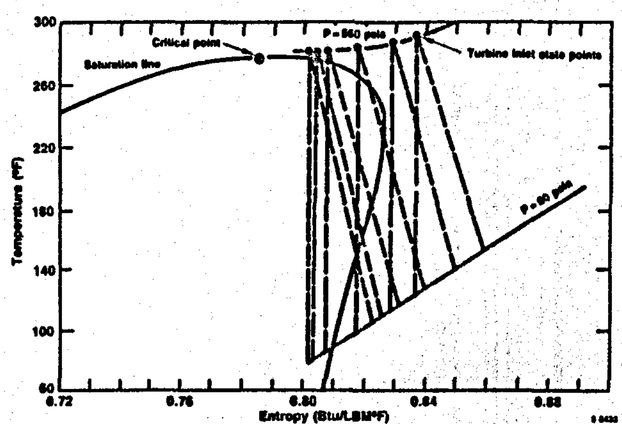

Figure 6. Supersaturated Expansion Conditions, Isobutane

Turbine isentropic efficiency has been plotted as a function of turbine inlet entropy in Figure 7 for these six tests. Values of maximum equilibrium moisture reached at the temperature of maximum dew-point entropy during each of the nozzle expansions can be estimated from the auxiliary scale. Efficiencies have been calculated, as described in (3), using the wattmeter readings (solid-faired triangles), and working-fluid state-point measurements (dash-faired circles). Repeatability of the triangles has been estimated, preliminarily, to be within ±1.4%, and the circles within about ±5.6%. Unknowns that can affect the relative levels of the two efficiency curves include the precise magnitudes of gear-box losses, generator efficiency, and the fraction of turbine-bearing and windage losses that find their way into the exhaust gas enthalpy. Because of these considerations, interpretation of the results is based on the trends shown by the two efficiency curves rather than their relative magnitudes.

A factor which must be considered in the interpretation of the dashed curve, is that the curve was calculated assuming no moisture in the turbine exhaust. Moisture present would reduce the calculated efficiency below its real value, and could result in a misleading trend on the plot. As an example, the presence of 1% moisture in the turbine exhaust for the test at 0.808 inlet-entropy would raise the circle about 5% in efficiency to the level indicated for the highest inlet-entropy test. A second factor which must be considered is the relatively large uncertainty estimated for the efficiencies based on fluid properties; the plotted circles could have resulted from an actual efficiency which is independent of turbine inlet entropy. Therefore, two interpretations come to mind for the trend shown by the efficiencies based on fluid properties. The first is that some moisture was present for the low inlet-entropy expansions in the form of sub-micron droplets which are essentially in complete thermal and velocity equilibrium with the vapor; then no degradation of actual efficiency would have resulted from the presence of moisture. A second interpretation is that no moisture was present, and the actual efficiency is independent of turbine inlet entropy, but the measurement uncertainties have introduced major scatter in the plotted results. The plotted triangles, based on the wattmeter readings, are seen to indicate no significant effect of inlet entropy on turbine efficiency.

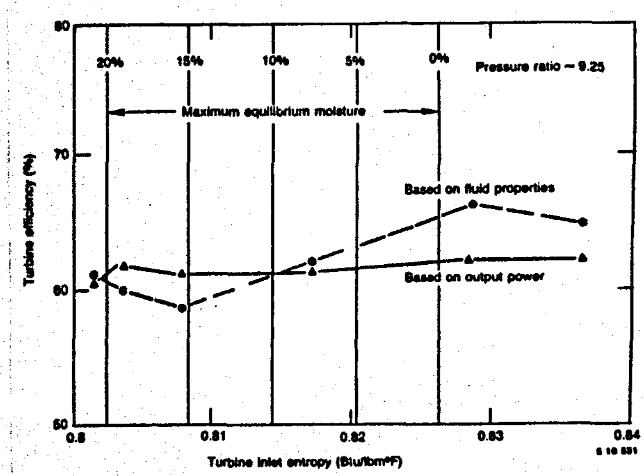

Figure 7. Turbine Performance for Supersaturated Expansions, Isobutane

Substantially the same behavior was shown for the other isobutane series, and for the 95% isobutane -5% hexane series (which reached an "equilibrium-moisture" state point of over 30%) as was discussed relative to Figure 7. It was concluded that the present results do not show turbine-efficiency degradation associated with expanding through the two-phase region. Future two-dimensional nozzle tests, which are planned with extensive wall pressure instrumentation and with a LASER system to illuminate condensate droplets formed, should provide a valuable supplement to these and other supersaturated-expansion turbine tests which will be conducted during the next several months.

CONCLUSIONS

The results and conclusions reached can be summarized as follows:

1. Good agreement between predicted and observed temperature distributions for both heaters and condenser was obtained using the state-of-the-technology HTRI heat-exchanger design computer codes.
2. The HTRI heater and condenser codes with NBS properties can be used for design of supercritical vaporization systems and condensers for pure and mixed-hydrocarbon binary geothermal plants; the designs will tend to be slightly conservative. The heater designs will be, on the average, about 20% larger than required if suitable fouling resistances are included. The condenser pressure will be equal to the bubble-point pressure for a temperature which is, on the average, about 1.4°F lower than the design bubble-point temperature.
3. Departure from integral condensation was not detected for the vertical condenser orientation.
4. Preliminary turbine results suggest that the turbine efficiency is not affected by supersaturated expansion processes.

Overall, it is concluded that the results presented here are favorable and support previous projections of potential performance gains approaching 28% for advanced binary plants.

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