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DEMONSTRATION OF A ROTARY SEPARATOR
FOR TWO-PHASE BRINE AND STEAM FLOWS

Final Report

By
Donald J. Cerini

January 1978

Work Performed Under Contract No. EY-76-C-03-1228

Biphase Energy Systems
Research-Cottrell
Santa Monica, California

MASTER

U. S. DEPARTMENT OF ENERGY
Geothermal Energy

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DONALD J. CERINI

Prepared for the
DEPARTMENT OF ENERGY
Geothermal Division

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CHANGE OF NAME

As of December 30, 1977, Biphase Engines, Inc., Santa Monica, California joined with Research-Cottrell, Inc. of New Jersey. Biphase Engines, Inc., is now *BIPHASE ENERGY SYSTEMS, INC.*

BIPHASE ENERGY SYSTEMS, INC. will remain a wholly independent operation in Santa Monica, California, with even broader staff and resources devoted to the technical research and development.

ACKNOWLEDGMENTS

Technical direction of this program was provided by Clifford MacFarland of the Geothermal Division of ERDA.

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We thank the Union Oil Company for their cooperation in allowing us to use their Brawley Wellsite and for their assistance in installation and operation of the Biphase test system.

ILLUSTRATIONS

Fig. No.		Page No.
1	Geothermal Separator Nomenclature	3
2	Separator/Pickup Probe Schematic	9
3	Geothermal Separator Test System Installation at Brawley, California Well Test Site	10
4	Geothermal Test Schematic	11
5	Rotary Separator System in Test	13
6	Rotary Separator Control and Instrumentation Console	14
7	Rotary Separator Resource Utilization Efficiency Data	19
8	Predicted Diffuser Exit Pressures	20
9	Multi-rim Separator Design Concept	23
10	Flow Pick-up Probe Size Requirement	27
11	Rotary Separator/Single Flash Steam System Comparison	34
12	2.2 Meter Rotary Separator Design Concept	39
13	Two Stage Rotary Separator/Steam Turbine	41

TABLE OF CONTENTS

ABSTRACT.....	1
I. INTRODUCTION.....	2
Background.....	2
Nozzle.....	4
Separator.....	4
Liquid Pickup.....	4
II. FIELD TESTS.....	6
Test System Description.....	7
Nozzle Description.....	7
Separator Description.....	7
Liquid Pickup Description.....	8
Well Flow System Description.....	8
Test Operations.....	12
III. TEST RESULTS AND DISCUSSION.....	16
Nozzle.....	16
Rotary Separator.....	16
Component Efficiencies.....	18
Nozzle Efficiencies.....	22
Separator Admission.....	22
Separator Efficiency.....	24
Flow Pickup Efficiency.....	24
Diffuser Efficiencies.....	25
Separator Performance.....	28
Scale Tolerance.....	30
Materials Evaluation.....	31
IV. ROTARY SEPARATOR STEAM FLASH COMPARISON.....	33
V. ROTARY SEPARATOR/TURBINE SYSTEM DESIGN.....	38
VI. CONCLUSIONS AND RECOMMENDATIONS.....	44
VII. TABLES.....	45
VIII. REFERENCES.....	55

APPENDICES

Appendix A - Rotary Separator Laboratory Tests.....	A-1
B - Two-Phase Nozzle Description.....	B-1
C - Performance Prediction Analysis.....	C-1
D - Geothermal Rotary Separator Fabrication Drawing Index.....	D-1

ABSTRACT

The application of a two-phase rotary separator for geothermal energy conversion was demonstrated. Laboratory tests were conducted with clean water and steam at Biphase Energy Systems, Inc., Santa Monica, California. Field tests were conducted at the Union Oil Co., Tow #1 wellsite near Brawley, California. The system tested consisted of the major components of a total flow rotary separator/turbine conversion system. A nozzle converted the brine wellhead enthalpy to two-phase flow kinetic by impinging the nozzle flow tangentially on the inside of the separator. The flow was therefore subjected to the high centrifugal force field in the separator. This caused the liquid phase to collect as a film on the separator drum with very little energy loss. The steam was allowed to flow radially inward to the central steam discharge. Potable water was obtained by condensing the steam exhaust. The brine collection system converted the liquid film kinetic energy to static pressure head.

The system was operated for 116 hours in a high salinity environment (115,000 ppm TDS). The system operated properly with no adverse effects from solids precipitation or scale build-up. The rotary separator produced separate flows of pure liquid and steam of greater than 99.5% quality. At an operating condition with 0.28 MPa (40psi) inlet pressure to the rotary separator, the liquid collection system delivered the separated brine at a pressure of 2.76 MPa (400 psia) with a steam discharge at atmospheric pressure and velocity of 96 m/s (316 ft/s). This represents a hydraulic power output of 3.96 kW/kgm/sec (1.8 kW/lbm/sec) for the brine flow and 4.6 kW/kgm/sec (2.1 kW/lbm/sec) of steam kinetic energy. This corresponds to a 34% conversion of isentropic enthalpy change for expansion from 0.55 MPa (80 psia) to atmospheric pressure. Further expansion of the steam to 0.014 MPa (2 psia) through a steam turbine of 70% efficiency would increase the system output to 7.4 kW/kgm/sec (3.4 kW/lbm/sec) at a system overall efficiency of 54.7%. This is 158% of the system efficiency for a single flash steam system.

Application of the rotary separator/turbine to a two-stage flash steam system showed a calculated power increase of 18%. A lower specific capital cost would result due to increased output power and elimination of the reinjection pump.

The system tests identified the necessary feasibility required for further development. Additional field tests were recommended on the existing system with the addition of liquid and steam turbines. Such a full system demonstration would then permit a complete design of a pilot plant in the multi-megawatt range.

I. INTRODUCTION

BACKGROUND

The national need for alternate energy sources places the development of geothermal energy conversion technology in the position of ever-increasing importance. There are numerous types of geothermal energy conversion systems in the demonstration phase aimed at electrical power production from liquid-dominated geothermal resources. The majority of these programs utilize some form of flash process as a means of eliminating the problems related to scaling, erosion and corrosion resulting from salt precipitation. The utilization of single-phase steam power generation technology, however, represents an inefficient use of the geothermal resource which exists as a two-phase steam/brine mixture in majority of known geothermal reserves.

There is a type of conversion equipment referred to as a total flow system designed specifically for the two-phase geothermal application. In this type of system, the maximum energy extraction is permitted when both the steam and liquid streams are used. Studies have shown that the power output from a geothermal energy conversion system with a single-stage flash process can be increased by two-thirds (Ref. 1 and 2) with the utilization of a total flow system. There are several types of total flow equipment that are currently being developed. The helical rotor type expander (Ref. 3) has demonstrated efficiencies in the 50% range and operability in the high saline environment. The low flow velocity typically will require extremely large expanders to operate at subatmospheric pressures as required for high thermodynamic efficiency. The large physical sizes will result in high capital cost and increased surface area and frictional losses above the values reported for the small units operated at atmospheric pressure.

Analyses of conventional steam impulse turbines performance operating as total flow expanders show low efficiencies due to high frictional losses. The losses result from the liquid droplets being unable to pass through a blade row without impinging and undergoing frictional losses proportional to the product of the entire surface area of all the blades and the relative velocity between rotor and flow. Some analyses have shown the maximum efficiency to be approximately 40%. Recent tests of impulse turbines with steam and water gave overall efficiencies of 20%, Ref. 4.

The separation of liquid droplets and vapor and power extraction from the liquid and vapor phases occur simultaneously in the case of the impulse turbine. It is the flow separation at high relative velocity (turbine blade velocity of approximately one-half the liquid velocity) that produces the high losses and low efficiency of liquid kinetic energy conversion to rotor torque. The rotary separator, Ref. 5, the type of total flow expander to be described in the following sections, eliminates

FIGURE 1 GEOTHERMAL SEPARATOR

the limitations ascribed to the impulse turbine by decoupling the separation and power extraction steps. The separation of steam and liquid takes place in a high centrifugal force field on the rotor rim, Fig. 1, where the relative surface to liquid velocity is near zero. The power extraction from the separated liquid takes place in a liquid turbine with flow wall surface areas proportional to liquid densities as compared to flow areas proportional to steam densities in the impulse turbine. The frictional losses associated with the brine kinetic energy conversion to output torque can thus be reduced by as much as the liquid to steam density ratio of 3000 for the rotary separator/turbine system compared to the impulse turbine.

The rotary separator performance tests to be described represent the first tests of the three primary components of a rotary separator with a brine flow. The components are the nozzle, rotary separator and the liquid pickup. The primary goals for these three components are summarized as follows:

Nozzle

1. demonstration of brine enthalpy conversion to kinetic energy steam/liquid flow at nozzle exit.
2. correlation of measured conversion efficiencies by theoretical model.

Separator

1. demonstration of ability to separate liquid brine from steam.
2. evaluation of steam purity.
3. evaluation of kinetic energy transfer from input brine/steam mixture to output brine.

Liquid Pickup

1. demonstration of liquid removal from rotary separator.
2. evaluation of removal efficiency.
3. evaluation of kinetic energy transfer of separated liquid to pressurized liquid flow.

The system used in performing these tests is shown schematically in Fig. 1. The liquid flow from the wellhead enters the two-phase nozzle and undergoes a partial flash as it flows to the lower pressure inside the separator housing. The mixture leaving the nozzle has a steam quality in the 5 to 20% range and a velocity of 200 m/s (650 ft/s). The two-phase jet impinges

tangentially on the inside of the cylindrical rotor mounted on bearings so that it can rotate freely. The drum is kept in motion by the impinging liquid and revolves at about 4000 rpm. Because of the high centrifugal force the liquid forms an annular layer only a few millimeters thick on the inside of the rotor rim.

The liquid is removed from the rotor and discharged from the separator by a "pitot tube" type collector immersed in the liquid layer. Pressure recovery in the collector is accomplished by incorporating a convergent/divergent diffuser contour in the collector inlet.

The results to be described show a nozzle efficiency that is 70% of the isentropic value for an expansion from 0.55 MPa (80 psia) to atmospheric pressure. This value agrees with a theoretical model that shows an increase in nozzle efficiency to 85% at 2.4 MPa (350 psia) nozzle inlet pressure, Appendix B, Fig. B-5. The separator evaluation shows the necessary degree of steam and liquid separation to permit steam condensation resulting in clean water output. The separation takes place while transferring 63% of the nozzle exit kinetic energy to kinetic energy of the liquid film at the inlet of the liquid pickup. The liquid recovery capability of the pickup probe was demonstrated with discharge pressures up to 2.75 MPa (400 psia). For the system operating condition of 0.55 MPa (80 psia) wellhead pressure and a separator speed of 3950 rpm liquid film stagnation pressure at the pickup inlet is 11.4 MPa (1650 psia) therefore a reinjection pressure requirement of 2.1 MPa (300 psia) requires less than 1/5 of the pressure rise available. The diffuser efficiency for this modest pressure rise is 85%. This combined with a nozzle efficiency of 69%, separator energy transfer efficiency of 66% and pickup inlet efficiency of 90% results in an overall efficiency of 34%. The addition of a turbine and subatmospheric condenser to the separator steam discharge would increase the system output to 57% of the isentropic enthalpy drop available. The trouble free operation with the highly saline brine flow from the Union Oil Co., Brawley, California well for 117 hours provides an important first field-test demonstration.

II. FIELD TESTS

The rotary separator system was operated on the highly saline well flow typical of the California Imperial Valley to provide the first demonstration of the system's total flow capability. The most important feature to be demonstrated was the ability to operate for a period of at least 100 hours since many other systems had experienced performance degradation due to precipitate deposition in shorter periods. The other performance goals to be achieved were evaluation of steam/brine separation, brine enthalpy conversion to separated-liquid kinetic energy, and two-phase nozzle efficiency. The system performance was to be evaluated over a range of inlet conditions from 0.28 to 1.0 MPa (40 to 150 psia).

The detailed performance results will be described in following sections. In general the test demonstrated several important characteristics. The two-phase nozzle converted the input flow enthalpy to kinetic energy over a range of inlet qualities from 2 to 20 percent. Based on the nozzle laboratory testing with steam and water the nozzle efficiency was estimated to be 50 to 70 percent over the range of test conditions.

For these range of conditions the separator speed ranged from 1,000 to 3,500 rpm. For the assumption of liquid film velocity equal to the separator rim speed, the conversion of nozzle exit kinetic energy to separator liquid film kinetic energy varied from 60 to 70 percent. Measurement of the brine flowrate in the liquid collection system showed complete liquid recovery. Steam quality at the separator discharge was found to be greater than 99.5%. The liquid collection system was evaluated by measurement of the liquid flow collected for return to the reinjection system. The return flow was found to agree closely with the estimated nozzle exit liquid flowrate. The amount of liquid carryover with the steam was measured to be approximately 0.5 percent of the liquid flowrate. This flowrate was also equivalent to the amount of steam calculated to be condensing on the steam discharge piping. The measurement of liquid discharge pressure up to 2.75 MPa (400 psia) was accomplished for nozzle inlet pressures of 0.28 MPa (40 psia). These values correspond to a diffuser efficiency of 85%. Over the range of conditions tested this varied from 75 to 85%. The product of the three component efficiencies quoted is 22% to 42% for the very low wellhead pressure tested. This value is the system efficiency for conversion of wellhead enthalpy to hydraulic output power. Such a system does not make use of the available steam enthalpy at the nozzle exit. Expanding the steam through a turbine to 0.014 MPa (2 psia) sink would increase the system output by 68% to an overall efficiency value of 57%.

Test System Description

The rotary separator system consisting of two-phase nozzle, separator and liquid pickup probe was installed for test in a semi-trailer van. This enabled laboratory testing at the Biphasic Energy Systems, Inc. (BES), test facility followed by geothermal well tests at Brawley. The laboratory tests are described in Appendix A. Fig. 1 shows schematically the primary components and design features of the rotary separator. The components are described in the following sections.

Nozzle Description

The flow enters the rotary separator through the nozzle. The exit aperture of the nozzle is shown in Fig. 1 with the direction of the flow indicated. The nozzle utilized for the steam/water and brine tests was 63.5 cm (25 in) long. It had a convergent/divergent cross-section with 5.59 cm (2.22 in) diameter inlet, 2.88 cm (1.133 in) diameter throat, and 5.08 cm (2.0 in) diameter exit. The nozzle wall contours were calculated for steam/water mixtures according to Ref. 6. The nozzle wall contour was fabricated by a composite fiber-glass epoxy casting around a metal mandrel made to the calculated contour which is given in Table B-11 of Appendix B. The nozzle centerline was inclined at an angle $\theta = 17.5$ degrees to the separator housing and located radially outward from the separator centerline a distance $R_n = 34.3$ cm (13.5 in). The flow upon leaving the nozzle impinges on the inside of the separator rotor rim which has an inside diameter, $D_s = 76.2$ cm (30 in). V_n , the nozzle velocity vector is inclined at a compound angle $(\alpha + \theta)$ to the rim such that the velocity vector, V_s , tangential to the separator rim is given by: $V_s = V_n \cos \theta \cos \alpha$.

For the particular test setup, $\alpha = 25.8$ degrees, was evaluated as $\alpha = \cos^{-1} (2 \times R_n / D_s)$.

Separator Description

The rotor rim shape is shown in cross-sectional detail in Fig. 2. The rotor has a lip of height $F = 1.1$ cm (0.43 in) which forms a reservoir for separated liquid flow. The rim is shown attached to a circular disc which is in turn attached to the shaft assembly. The cantilevered shaft is supported by a double bearing assembly as shown in Fig. 1. The rotor is enclosed by the aluminum separator housing consisting of two circular halves spaced 12.7 cm (5.0 in) apart and an aluminum sheet metal section placed circumferentially around the rotor and between the separator housing halves. Fig. A-4 of Appendix A shows the rotary separator assembly installed in the test van which is shown in Fig. A-3.

Attached to the separator drum were material test coupons exposed to the high velocity steam-brine environment. Eight material samples were installed representing a variety of corrosion

resistance and strength properties. The materials samples consisted of:

1. Inconel 600	5. Titanium (Ti-7AL-4V)
2. Hastalloy X	6. Tungsten
3. Tantalum	7. Molybdenum
4. Columbium (Niobium/1% Zirconium)	8. Zirconium

Aluminum and stainless steel were not included in the selection of test specimens since the rotor disc and rotor rim were fabricated from stainless steel (Type 304) and aluminum (Type 6061-T6) respectively.

Liquid Pickup Description

The liquid pickup probes used to remove the liquid layer from within the rotary separator are shown schematically in Fig. 2. The inlet is shown to be rectangular in cross-section, 2.03 x 15.8 mm (0.08 x 0.62 in) which converges at a 3 degree angle to a 1.02 x 15.8 mm (0.04 x 0.62 in) throat section. The walls then diverge at a 10 degree total included angle to a circular cross-section of 10.2 mm (0.40 in) diameter. The probe assembly was fabricated from three tube sections. The inlet with diffuser contour was fabricated from stainless steel tube stock, 1.27 cm (0.5 in) diameter, by flattening the tube stock onto an aluminum mandrel machined to the desired diffuser contour. The edges of the flattened tube were machined to a uniform width, $w_0 = 1.27$ cm (0.5 in), and welded closed. The aluminum mandrel was then melted out of the inlet section. The inlet of the probe is shown to have beveled walls which worked satisfactorily with the clean water in the laboratory tests. In order to eliminate possible erosion in the geothermal well tests of the sharpened wall edges, a 1.14 mm (0.045 in) diameter weld bead of stellite was added to protect the leading edges of the probe inlet.

Well Flow System Description

Fig. 3 is a photograph of the test system and van installed at the wellsite. Shown on the side of the van are four connections to the well flow system to provide single phase steam and brine. The third and fourth lines are used to return the separator brine and steam outputs to either the brine injection or atmospheric steam discharge. A schematic of the well flow system is shown in Fig. 4. The Union Oil flow system is of multi-purpose capability which was operated in a manner to produce separate brine and steam flows at 1.03 MPa (150 psia). These separate steam and brine flows were metered by orifice metering systems installed in the Biphasic Engines test van and mixed at the nozzle inlet. The rotary separator operating on the two-phase flow resulted in separate steam and pressurized brine outputs. The separated steam was vented to the atmosphere. The brine output was returned to Union Oil's system to be flashed at

FIGURE 2 SEPARATOR/PICKUP PROBE SCHEMATIC

FIGURE 3
GEOHERMAL SEPARATOR TEST SYSTEM
INSTALLATION AT BRAWLEY, CALIFORNIA
WELL TEST SITE

FIGURE 4 GEOTHERMAL TEST SCHEMATIC

the inlet to the cyclone separator No. 3. The return brine flow was metered by a third orifice metering section. The liquid return valve was used to throttle the brine flow during diffuser performance evaluations.

Test Operations

Fig. 5 is a photograph of the rotary separator in operation as installed in the test van at the Union Oil Co., Brawley, California geothermal wellsite. The photograph shows the separator with the aluminized thermal insulation installed to minimize steam condensation. The steam discharge is shown out of the central 12.7 cm (5 in) diameter pipe section. The brine output line is the small diameter 1.9 cm (3/4 in) tube partially visible at the top-center.

The system startup was very simple requiring a short warm up period in which the steam and brine flows were initiated through the purge valves with the flow metering valves closed. Steady state operation was achieved after a 15 minute warm up of the rotary separator by initiating a low steam and brine flow through the system. During this period the separator drum was held stationary by means of an electro-magnetic brake while steam and brine flowed through the system and out through a separator housing drain to the brine waste pump. A proximity probe placed between the separator drum and housing was used to indicate thermal equilibrium between the rotor and the housing. With both the steam and brine metering valves set to pressurize the nozzle inlet to 0.03 MPa (5 psig), de-energizing the brake would allow the separator to come to a rotational speed of 800 rpm. At this condition the purge valves were closed and liquid return valve was opened, there being sufficient separator velocity head at the inlet to the liquid pickup probe to return the flow to the brine disposal system. This was the initial condition from which the performance survey was conducted. The nozzle inlet pressure and nozzle inlet qualities were the two prime independent parameters varied. Each test condition was maintained for a period of 15 minutes to several hours to verify steady state operation before the next condition was set. The operating conditions were recorded from the visual displays of temperature, pressure, and speed. The displays were mounted on a partition shown in Fig. 6 between the separator test area and the control room in the test van. The partition contained also the remote valve controls for the brine and steam systems.

Measurement of the input flowrates of steam and brine permitted comparison of measured and predicted nozzle performance, i.e., nozzle flow as a function of inlet pressure and quality. Continued agreement between the measured and predicted nozzle performance was used as an indication of no significant change in nozzle contour from scale deposition. Comparison of measured and predicted separator speed was considered an important indication of proper operation of several separator processes. Nozzle to separator energy transfer and liquid pickup probe flow

FIGURE 5 ROTARY SEPARATOR SYSTEM IN TEST

FIGURE 6 ROTARY SEPARATOR CONTROL AND INSTRUMENTATION CONSOLE

recovery were the most important. The concern was whether a scale deposit on either the separator drum or liquid pickup probe would occur to alter their performance. Proper operation of the liquid recovery system was also indicated by observing a constant liquid level in the surge tank between the liquid pickup discharge and the flowmeter. A decrease in level would indicate either plugging of the probe or increase in noncondensable gases. Normal operation was to maintain the vent closed. Calculation of nozzle exit quality permitted an estimation of the liquid flowrate that should be returned and thus measured by the pickup-probe output flowmeter. Steam quality measurements were also by means of an installed throttling calorimeter and with chloride analysis technique* which provided an alternate and independent steam quality measurement.

* Developed by Union Oil Field Analysis Group

III. TEST RESULTS AND DISCUSSION

Nozzle

The rotary separator system as installed in the mobile test van, Fig. 5, was tested to characterize its performance with clean steam and water in preparation for high salinity geothermal brine tests. The preliminary tests with steam and water are described in Appendix A. Included in these tests were a separate series of nozzle calibration tests to measure separately the nozzle performance. The results agreed to within $\pm 3\%$ of the values predicted by the computer program of Ref. 6. Over the range of expected well head pressures, the nozzle calibration with steam and water showed a conversion of input flow enthalpy to nozzle exit kinetic energy of 50 to 70% at 1 atm discharge. The same nozzle was used in both test series with no deleterious effect on the epoxy-fiberglass nozzle wall. The nozzle construction details are described in Appendix B. The relationship of the predicted and measured nozzle exit velocity is shown in Fig. B-2. Although the measured efficiencies are quite low at the 70% level, the close agreement with the predicted gives encouragement that the predicted efficiencies in the 80 to 90% range can be achieved when wellhead pressures in the 1.38 to 2.76 MPa (200 to 400 psia) range are utilized or when advanced nozzle design techniques are used for lower pressures, Ref. 8. Nozzle efficiencies have been measured in the 90% range with steam and water, Refs. 6 and 7. The nozzle efficiency for the reduced discharge pressure is shown by the dashed curves in Fig. B-5 to be reduced from 75% at 1 atm nozzle exit pressure and 0.69 MPa (100 psia) inlet pressure to 70%. This represents the fact that the nozzle contour was optimized for the atmospheric discharge. The increased nozzle length from 63.5 cm (25 in.) to 130 cm (51 in) required for the subatmospheric discharge pressure produces proportionally higher frictional losses as compared to higher output kinetic energy which results in the slightly reduced nozzle efficiency. The calculation of nozzle performance and optimum nozzle contour were performed by a computer program according to Ref. 6.

A similar nozzle contour optimization, Ref. 8, has shown a 10 to 15 percentage point increase by means of an increased pressure gradient in the convergent inlet section. This has the effect of increasing the relative velocity between the gas and liquid phases in the region where the absolute velocities are low. This shifts the maximum shear and droplet breakup to the region of low frictional losses.

Rotary Separator

In general, the system performance was very good. There was no effect on performance from the high salinity brine scale deposition, corrosion or erosion. There was no plugging of any

of the flow or instrumentation systems. The only hardware operational problem was the required replacement of the rotor bearings on two occasions after only two hours of operation. After the second failure it was found that the new bearings had not been repacked by the supplier with high temperature (175°C) water resistance grease as requested. The third bearing set with the correct grease packing served throughout the remainder of the 115 hours of testing with no apparent change in bearing condition or loss of grease packing.

The field test data is shown tabulated in Table I including the calculated performance parameters based on the test data. Each line of data was assigned a sequential scan number with a data scan being taken on approximately one hour intervals. The scan numbers and elapsed test time being shown in columns 1 and 2. The next eight columns represent the measured separator system pressures. The first three being the differential pressure outputs from the brine input, steam input and brine recovery orifice flowmeters. The relative locations of the pressure measurements are indicated in Fig. A-2. The next eight columns indicate the temperatures measured at the locations indicated also in Fig. A-2. The measured separator speed is indicated in the last column of measured data. The remaining nine columns summarize the calculated values for: brine input ($\dot{m}_l$), steam input ($\dot{m}_s$), and brine recovery ($\dot{m}_r$), flowrates, nozzle inlet quality (x_1), nozzle exit quality (x_2), separated liquid kinetic energy (P_o), conversion efficiencies (η_1) and (η_2) and the ratio (R) being recovered liquid to input liquid flowrate, $\dot{m}_r/\dot{m}_l$.

The liquid film power (P_o) was evaluated as the kinetic energy of the flow ($\dot{m}_r$) at the velocity corresponding to the separator rim speed. The ratio of that power to the isentropic nozzle enthalpy drop defines the efficiency term η_1 . Division of this efficiency value by the enthalpy-ratio factor from Fig. C-7 yields the corrected efficiency (η_2) which considers the reduced enthalpy extraction due to the brine concentration, $C=0.115$. The variation of this calculated efficiency with nozzle inlet pressure is shown in Fig. 7. The last column in Table I contains the probe mass recovery ratio (R) calculated as the ratio of the measured probe output liquid flowrate to the nozzle exit liquid flowrate, $\dot{m}_r = (x_2 \dot{m}_l)$. The value of $R = 1.0$ indicating complete liquid recovery. These values are based on two pairs of pressure and temperature measurements for calculation of $\dot{m}_l$ and $\dot{m}_r$, each measurement pair having a $\pm 2\%$ accuracy. The recovery ratio error level is $\pm 4\%$.

An examination of one of the high speed operating points, #112, shows a nozzle inlet pressure of $P_s + 14.7 = 79.7$ psia with a 5.1% inlet quality. Using a 68% nozzle efficiency the system achieved a 38.6% conversion efficiency of the isentropic nozzle enthalpy drop to brine kinetic energy in the separator liquid film (at the probe inlet). For an assumed diffuser efficiency of $85\% \times 0.386 = 0.33$. The diffuser efficiency was evaluated at

lower power levels to be 85% but was not measured at this condition since the recovery pressure would have exceeded 8.9 MPa (1300 psia), (Fig. 8)*; which exceeded the pressure rating of the liquid pickup system. The following section gives a detailed examination of the separator component efficiencies.

Component Efficiencies

One result of the field tests to be reviewed in this section is an estimation of the performance of the rotary separator in converting the available enthalpy of the brine to kinetic energy of the separated steam and brine flows at the separator discharge.

This conversion will be a measure of the overall efficiency of the system consisting of nozzle, separator and liquid pickup probe. The most direct evaluation of overall efficiency is thus based on the relationship:

$$\eta_o = \frac{\text{kinetic energy of liquid \& steam}}{\dot{m}_t \Delta h} = \frac{\frac{1}{2} \dot{m}_t V^2}{\dot{m}_t \Delta h_i} = \frac{N_s^2 \times 7.773 \times 10^{-7}}{\Delta h_i} \quad (1)$$

where $\dot{m}_t$ = total input nozzle flowrate, kgm/s
 V = mean steam and liquid exit velocity, m/s
 N_s = separator speed, rpm
 Δh_i = isentropic enthalpy change for expansion from nozzle inlet pressure to atmospheric condition, j/gm

This calculation is then based on the measured separator speed and measured nozzle input and exit pressures and temperatures. The isentropic enthalpy change from the nozzle inlet state point to nozzle exit conditions is then based on the procedure in Appendix C.

For test condition No. 112 of Table I, the measured separator speed was 3375 rpm. The nozzle inlet and exit conditions were:

Inlet:	Pressure	0.51 MPa	(74.7 psia)
	Temperature	156°C	(312°F)
	Quality	0.051	
Exit:	Pressure	0.11 MPa	(15.9 psia)
	Temperature	109°C	(228°F)
	Quality	0.17	

The corresponding isentropic enthalpy drop is 26.5 j/gm (11.4 BTU/lbm). The brine concentration for this series of runs was measured at 115,000 ppm. Using a concentration of $C = 0.115$, the calculation procedure for brine properties evaluation in Appendix C the enthalpy drop is reduced to 22.9 J/gm (9.9 BTU/lbm). The overall efficiency is thus evaluated as:

$$\eta_o = \frac{3375^2 \times 7.773 \times 10^{-7}}{22.9} = 0.386$$

*See Fig A-7 for comparison of measured and predicted probe discharge pressure with clean steam and water.

FIGURE 7 ROTARY SEPARATOR RESOURCE UTILIZATION EFFICIENCY
VERSUS NOZZLE INLET PRESSURE

FIGURE 8 PREDICTED DIFFUSER EXIT PRESSURES

This is a conservative estimate of efficiency since the steam is assumed to have the same velocity as the liquid. The nozzle computer program shows a steam velocity of 30% in excess of the mean liquid velocity (Table B-I, $V_g = 1.3V_L$) which correlated the previous test data with clean water and steam as shown in Fig. B-2 of Appendix B. If the additional energy of the steam were included, assuming it could be extracted efficiently with a radial inflow turbine or impulse blading, the added power would increase the overall efficiency from 34 to 54.7% for a steam expansion to 2 psia.

It is of interest to estimate the component efficiencies for the nozzle, separator and liquid pickup. Although the field test was designed primarily to demonstrate separator effectiveness, some estimates of component efficiencies can be made utilizing the component calibration tests results with clean water described in Appendix A as an upper bound for the brine flow tests.

The overall system performance parameter, η_o , will be analyzed as being a function of five component efficiencies. This subdivision, where:

$$\eta_o = \eta_n \times \eta_{sa} \times \eta_s \times \eta_p \times \eta_d, \quad (2)$$

is somewhat arbitrary but based on functional groups, whose individual efficiencies can be either calculated or measured. The five component efficiencies are defined in Table II. They are evaluated as the ratio of power output from the component under consideration to the power input from the preceeding component. For this analysis, the rotary separator is viewed as a five step, series-power-transfer function. This is somewhat of an oversimplification since most of the terms are not independent of the others but this approach gives some insight into the relative importance of the individual components. This arrangement also provides a more direct means of evaluating overall system efficiencies for other operating conditions or modified designs.

In the case of the liquid turbine when the power is extracted not as hydraulic ($\dot{V}\Delta P$) power but as rotational power in using the probe as a turbine, then the diffuser losses are reduced in proportion to the power extracted via the turbine. The diffuser is thus used to pressurize the liquid to meet the centrifugal-force pumping requirement and to provide reinjection pressure. For the nozzle inlet pressure of 0.55 MPa (80 psia) and separator speed of 3800 rpm these two pressure requirements are 2.6 and 2.1 MPa (382 and 300 psia) respectively as compared to an available stagnation pressure of 10.5 MPa (1528 psia). The diffuser efficiency related loss with the liquid turbine will be approximately one-fourth of the loss with the stationary probe.

Nozzle Efficiency

The source of nozzle efficiency is shown in Table II to be a calculation of nozzle exit kinetic energy based on the nozzle exit velocity. The two values of efficiencies shown represents the calculated or measured value for the test system and the second value being an estimation of an achievable efficiency. For the nozzle, both efficiency values come from Fig. 5, corresponding to: (1) current nozzle operation at 0.55 MPa (80 psia) source pressure and (2) advanced nozzle design at 2.1 MPa (300 psia) and improved atomization with two micron droplets. This is a large extrapolation from the nozzle inlet pressure of 0.55 MPa (80 psia), where good agreement between predicted and measured performance was obtained. Testing of other two-phase nozzles, Ref. 7, have shown measured efficiencies in the 80 to 90% range attributable to increased droplet atomization from operation at high pressures.

The variation of nozzle efficiency with nozzle inlet pressure and nozzle inlet quality is summarized in Fig. B-5. The efficiency is seen to increase with increasing pressure and steam quality. Both of these parameters relating directly to improved nozzle efficiency as a result of increased liquid atomization.

Separator Admission

The separator admission efficiency is a consideration of nozzle exit kinetic losses due to the geometrical non-alignment of the nozzle center line with respect to the separator direction of motion. There are two components of this geometrical calculation; one is the inclination of the nozzle at an angle θ with respect to the plane of rotation of the separator disc and the second is the inclination of the nozzle exit velocity component at an angle α to the rim-tangential velocity vector (Fig. 1). For the test system, the two angles were $\theta = 17.5$ and $\alpha = 25.8$ degrees. The velocity transfer from the nozzle to the separator liquid film based on these two alignment considerations is the product of the cosine of the two angles ($\cos \theta \cos \alpha = 0.85$) and the energy transfer efficiency is the square of this product, therefore $\eta_{sa} = (\cos \theta \times \cos \alpha)^2 = 0.737$. This efficiency level is thus a function of design and fabrication complexity with respect to making the nozzle inclination angle θ as small as possible. A feasible rotary separator design is one with a full admission nozzle where an angle $\theta = 10^\circ$ is used. The angle α influence on efficiency is also a separator design function since the angle α is defined by distance of the nozzle center line from the separator rim. An improved design would use an angle $\alpha = 18.7^\circ$ which corresponds to a nozzle center line to separator-rim distance 2.03 cm (0.8 in.) as compared to 3.8 cm (1.5 in.) for the current design. For such an improved design, the separator admission efficiency is increased to $\eta_{sa} = 0.87$. The total nozzle flowrate for a full admission design decreases as θ and α are decreased to improve efficiency. Therefore the maximum energy output per system will represent an optimization of efficiency and total flowrate and power output.

FIGURE 9 MULTI-RIM SEPARATOR DESIGN CONCEPT

Another means of improving the separator admission efficiency without reducing the α angle which requires making the nozzle smaller is to use a multi-rim separator design. This is shown schematically in Fig. 9. A four-rim separator design would reduce the nozzle segment to rim distance from 1.5 in. to 0.5 in. thus the angle $\alpha = \cos^{-1} (15-.5)/15 = 14.8$ degrees and separator admission efficiency η_{sa} would increase from 0.74 to 0.85. Reducing the inclination angle, θ , to 10° from 17.5° would further increase η_{sa} to 0.907.

Separator Efficiency

The separator efficiency term, η_s , of Table II is a measure of the separator flow kinetic energy lost due to bearing friction and separator windage losses. It was not possible to measure these losses during separator operation with well flow, however, an estimation of their magnitude was obtained by measurement of separator speed decay rates upon turning the brine and steam flows off. The power loss as a function of separator speed is summarized in Fig. A-8, which shows the data obtained for both the steam/water calibration and brine flow tests. A fit to the data gives the relationship of power loss (P_{SL} , kW) to separator speed (N_s , rpm) as:

$$P_{SL} = 8.413 \text{ E-10 } N_s^{2.6545}, \text{ kW W} \quad (3)$$

This represents an underestimate of the bearing losses since they would be higher under load of the separator flow. It is an overestimate of the windage losses, since the relative velocity between rotor and flow is higher during the no-flow speed decay measurement than during flow tests when the steam and rotor speeds are more equal. The performance prediction program to be described in Appendix C calculates the bearing and windage losses directly based on bearing characteristics and conventional rotating surface frictional losses. Both methods indicate about an 11% loss in separator film kinetic energy loss or a separator efficiency, η_s , of 89%. By a system change to a full admission nozzle, the ratio of the separator losses (P_{SL}) to separator film power (P_{sa}) is reduced from 0.11 to 0.05 which increases the separator efficiency:

$$\eta_s = 1 - P_{SL}/P_{sa}$$

from 0.89 to 0.95.

Flow Pickup Efficiency

The flow pickup efficiency term, η_p , describes the transfer of liquid film jet power to flow jet power inside the probe inlet. The two terms are different in proportion to the amount of the flow intercepted by the probe wall. This flow undergoes a momentum loss, $\dot{m}_p V_s \cos \theta$. If this flow is deflected

back into the separator film flow this momentum loss is extracted from the separator film jet power in accelerating the intercepted flow from velocity, $V_s \cos \beta$ to V_s . If some of the intercepted flow is deflected not into the separator but into the steam and carried out of the system as liquid carryover, the probe flow, $\dot{m}_p$, will be less than the separator flow, $\dot{m}_s$, by that amount and the probe power reduced proportionally. As seen in Table II, the probe power loss is evaluated as the fraction (W/r_o) of total flow intercepting the probe wall times the momentum loss of that flow, $\dot{m}_p V_s \cos \beta$, times the product of the terms: separator speed and separator radius which converts the momentum change to power. The flow pickup efficiency, η_p , is thus calculated from:

$$\eta_p = 1 - \frac{(\dot{m}_r \times \pi^2 \times R_s^2 \times N_s^2) \times (W/r_o) \times (1 - \cos^2 \gamma)}{2 \times (P_{sa} - P_{sL}) \times g} \quad (4)$$

where the denominator has been calculated according to the equations of Table II and the numerator terms;

$$\begin{array}{lll} F = 0.43 & R_s = 1.25 \text{ ft} & L_o = 0.66 \text{ in.} \\ w = 0.049 \text{ in.} & R_p = 0.5 \text{ ft} & L_s = 3.5 \text{ in.} \\ r_o = 0.8 \text{ in.} & \alpha = 15 \text{ deg.} & \end{array}$$

are geometrical shape constants for the probe and separator as shown in Fig. 2. The separator speed, N_s , and probe liquid flowrate, $\dot{m}_r$, were measured. The data in Table I shows the liquid flowrate recovered by the probe, $\dot{m}_r$, to be within the measurement accuracy, $\pm 2\%$, of the maximum recoverable flow, where the maximum is the product of total liquid input, $\dot{m}_L$, and the nozzle exit quality, X_2 . Table II shows the flow pickup efficiency was 90%. For the design with full admission nozzle, the total flowrate would be increased such that the liquid film thickness, r_o , entering the probe would be increased so the probe flow cross-section would be nearly square, $r_f = L_o$. This would reduce the loss from flow impingement on the upper probe wall in proportion to the flow area increase relative to the probe wall cross-section. For the test system probe the ratio r_f/w , was $0.50/0.49 = 1.02$, which would increase to $9.8/0.049 = 16.3$, for a square probe. Thus for a full admission system, the impingement loss would be reduced by one-third of $16.3/1.02 = 5.3$. The one-third term results from additional flow impingement on the two vertical sides of the probe. The net effect is to increase the pickup efficiency term, η_p , from 0.90 to 0.93.

Diffuser Efficiency

The diffuser efficiency term, η_d , was evaluated on the basis of the pressure recovery capability for the probe diffuser. The pressure recovery was measured by back-pressuring the brine discharge from the probe to determine the pressure at which a portion of the flow entering the probe was expelled from the probe inlet. This is evidenced by a reduction in separator speed and a reduction in liquid recovery flowrate, $\dot{m}_r$.

This limiting pressure recovery level is then used to evaluate a diffuser efficiency by comparing it to the theoretical pressure output for an isentropic two-phase diffuser with the same inlet velocity and inlet ratio of gas to liquid volume. The maximum diffuser exit pressure can then be calculated from the expression (Ref. 9):

$$V_1^2 = P_1 - P_2 + \frac{rv_1}{r} \ln \frac{P_2}{P_1} \quad (5)$$

where the exit isentropic pressure, P_2 , is obtained by successive substitution of values for P_2 until the calculated value for V_1 equals the separator flow velocity, based on the test conditions for inlet pressure and inlet gas to liquid volume ratio, P_1 and rv_1 , respectively.

Fig. 8 shows the results of this calculation where pressure recovery, P_2 , is shown as a function of nozzle inlet pressure at three nozzle inlet quality values. Measured data for clean water tests is shown in comparison to the calculated curves in Fig. A-7. An empirical fit to two-phase diffuser test data, Ref. 9, yields a relationship of diffuser efficiency as a function of inlet volume ratio given by:

$$\eta_d = .86 - 0.09rv_1 \quad (6)$$

The rv_1 term is calculated for the test conditions from:

$$rv_1 = (L_0 \times r_0 \times p \times V_1 / \dot{m}_r) - 1. \quad (7)$$

For the test conditions the efficiency varied between 71 to 85%, which agrees closely to the efficiency calculated from Eq.(6). This shows a limiting efficiency of 85% for a full-flowing probe, $r_f = r_0$, Fig. 2. A characteristic of the two-phase rotary separator/turbine is that as the power is reduced to 50% of full power, the diffuser efficiencies varies between 83 and 85%, corresponding to variations in the degree of probe-flow fullness from 85% to 100%. Fig. 10 shows the probe height requirements for the test system. For a 10% nozzle inlet quality, the same probe will flow full over the nozzle inlet pressure range from 0.3 to 0.69 MPa (50 to 100 psia).

FIGURE 10 FLOW PICK-UP PROBE SIZE REQUIREMENT

Separator Performance

One of the primary goals of the geothermal field tests was to demonstrate the effectiveness of the rotary separator for steam and brine separation. The effect of the high saline brine flow representative of the Brawley, California area was to be assessed by comparison with system tests of the same system with steam and clean water, Ref. 10. The separation process to be evaluated takes place after pressurized brine and steam are expanded to atmospheric pressure through a two-phase nozzle. The resulting high velocity (200 m/s) two-phase flow consists of liquid droplets in a steam continuum with typically a 0.1 to 0.2% steam quality (ratio of steam to liquid plus steam flows). This flow then impinges at an effective angle of 30.8 degrees on the rim where the cosine of this angle times the nozzle exit flow momentum is retained by the flow along the rim (85.9% to nozzle momentum). The component of the flow momentum, $(\dot{m}_S + \dot{m}_L) V_N \times \cos \theta \sin \alpha$, which is normal to the separator rim is lost and the component of flow momentum, $(\dot{m}_S + \dot{m}_L) V_N \times \sin \theta$ parallel to the separator rim surface tends to cause a non-uniform film depth across the 8.25 cm (3.25 in) width of the rim in the vicinity of the nozzle. As the flow proceeds around the circumference of the separator the liquid droplets are exposed to a radial acceleration of 5200 g's for a rotor speed of 3500 rpm. This achieves a very rapid migration of the liquid droplets into a liquid film flowing at the same speed as the separator after impingement. The separation of the steam and liquid is now complete with what would be 100% effectiveness if the liquid could be removed without re-entrainment of the liquid by the steam. There is some re-entrainment that comes about as a result of liquid splash from the pick-up probe placed in the liquid film to collect the liquid flow. There are other means of removing the liquid flow that would not provide a re-entrainment loss such as eliminating the pick-up and letting the liquid layer build up and spill over the lip of the separator into a circumferential collection site.

This would eliminate the re-entrainment possibility but also would eliminate the possibility of efficient recovery of the liquid flow kinetic energy as pressure head or as torque. The liquid pickup probe design shown in Fig. 2 represents a compromise between these two ends of efficient separation and kinetic energy conversion. The results from the brine flow tests, Table I, and the earlier clean steam and water tests, Appendix A, show liquid recovery rates of 98 to 99.5% of the nozzle exit liquid flowrate. For the best liquid recovery rate operating condition with 0.5% indicated liquid carryover, the nozzle input flowrate was typically 1.86 kgm/s (12,700 lbm/hr). The liquid flowrate available for liquid pickup recovery, calculated on the basis of the amount of brine flash to steam in flowing through the nozzle, is a liquid flow rate of 1.58 kgm/s (12,530 lbm/hr). This shows a 2% higher liquid recovery than calculated available liquid. This, however, represents good

agreement since an error analysis shows that a measurement error of $\pm 2\%$ is possible in making either the nozzle input flowrate or brine return flowrate measurements. Both measurements require two pressure and one temperature measurements with one additional pressure and temperature to establish nozzle inlet and exit qualities. With a $\pm 1\%$ and $\pm 2\%$ accuracy on temperature and pressure measurements, respectively, the total error possibility greatly exceeds the observed error between the input and return liquid flowrates. The stated liquid recovery of 99.5% is based not on the difference between these two relatively large magnitude input and return flowrates, but on the measurement of steam quality. Two types of steam quality measurement were used. These consisted of a throttling calorimeter installed in the steam discharge line. In addition, a technique utilizing the comparison of chloride contents in the input brine and steam condensate was used to evaluate the steam quality. A steam operated ejector was used to draw a steam sample across a throttling orifice. Measurement of input and output temperature and pressure, assuming isenthalpic expansion of the steam into the superheat region, permitted calculation of the input quality. This measurement accuracy was also dependent on the temperature and measurement accuracies which resulted in a steam quality of $99\% \pm 2\%$. The chloride analysis showed a total chloride concentration content $C_b = 70,500$ ppm in the input brine and concentration $C_s = 435$ ppm in the steam discharge. The steam condensate was taken by drawing a steam sample into a water cooled copper tube. The liquid samples were analyzed by the GHT Laboratories of Imperial Valley, Inc. located in Brawley, California, with a 3% TDS variation between samples taken over a range of operating conditions. The readings were converted into a steam quality value, X_s , by the following procedure:

For a total nozzle flowrate of $\dot{m}_t$ kgm/s steam plus $(1-X_2)$ $\dot{m}_t$ kgm/s of liquid flow. A liquid carryover of one kgm/s would include a chloride carryover of $C_b \times 10^{-6}$ kgm/s. Since there was a chloride concentration of $C_s \times 10^{-6}$ kgm per kgm of steam and the amount of steam was $X_2 \times \dot{m}_t$, then the total amount of chloride in the steam, m_c , was $C_s \times 10^{-6} \times X_2 \times \dot{m}_t$. The liquid carryover rate, LC, corresponding to the mass of chloride, m_c , then is the ratio of the total chloride to the chloride concentration in the brine. C_b , therefore:

$$LC = \frac{m_c}{C_b \times 10^{-6}} = \frac{C_s \times 10^{-6} \times X_2 \times \dot{m}_t}{C_b \times 10^{-6}} \quad (8)$$

The steam quality, X_s , was then calculated as the ratio of the steam mass, $X_2 \times \dot{m}_t$, to the steam plus liquid carryover or:

$$X_s = \frac{X_2 \dot{m}_t}{X_2 \dot{m}_t + (C_s/C_b) X_2 \dot{m}_t} = \frac{1}{1 + (C_s/C_b)} = \frac{1}{1 + (435/70500)} \quad 0.994 \quad (9)$$

Scale Tolerance

This brine test series provided the first opportunity to investigate the characteristics of the rotary separator system in a potentially severe scaling environment. The brine was analyzed by the GHT Laboratories and found to contain 115,000 ppm of total dissolved solids. The brine output from the liquid pickup was analyzed also. The concentration was the same within the analysis accuracy of 100 ppm. Assuming there had been a 100 ppm scale deposition rate, for an average flow-rate of 0.91 kgm/s (2 lbm/s), for 120 hours a total of 39 kgm (86 lbm) of scale would have formed. An estimate of the scale actually deposited shows only 1/20th of that amount distributed in a 0.76 mm (0.03 in) thick layer on the inside of the separator rim and the outer 10% of the separator disc. From this it is concluded the deposition rate was an order of magnitude less than the TDS analysis accuracy. An analysis of the scale shows it to be a mixture of iron sulfide and silica. It formed a soft and easily dislodged coating on the aluminum separator rim but a very hard and tenacious scale formed on the immediately adjacent stainless steel separator disc. Samples had to be removed by a hammer and chisel from the latter whereas one's fingernail would dislodge scale flakes from the rim. It should be noted the rotor rim had a 0.0002 electrode-less nickel coating on which the soft deposit formed.

The soft scale on the separator rim was easily dislodged by the pickup probe. The probe had been positioned to be 0.76 to 1.07 mm (0.03 to 0.05 in) from the separator rim. As the scale reached that approximate thickness, it was removed by the probe. The probe was fabricated from stainless steel, type 304, with the addition of a stellite bead, 1.14 mm (0.045 in) in thickness, on the leading edge of the probe inlet. Earlier probes without this hardened tip became nicked and damaged from apparent solids in the flow. The hardened tip protected the probe from any apparent damage. The same type of scale formed on the nozzle wall but was generally two-thirds of the separator rim scale thickness. The only evidence of the typical silica deposits was in areas of stagnated flow. It was found that the dissolution of silica from brine samples taken from the liquid pickup probe outlet began within two minutes after sampling but continued for several hours. It is the long dissolution period for the majority of dissolved solids as compared to the short residence time of the flow in the rotary separator typically (30 to 45 msec) that provides degree of protection of the rotary separator from the potentially heavy scale deposits. Several well system shutdowns were occasioned by scale plugging. In contrast there was no plugging of any of the rotary separator components which includes the 2.03 x 16.8 mm (0.08 x 0.66 in) cross-section aperture to the inlet of the liquid pickup probe.

Materials Evaluation

A series of materials test coupons were attached to the separator disc for exposure to the brine/steam flows. Of the eight test coupons all acquired an iron sulfide scale coating that was similar in thickness and strength to the scale on the stainless separator disc to which they were attached. This scale bond strength to the coupons was very high requiring removal by a sharp edged chisel. This was in marked contrast to the bond strength to the separator rim. None of the test coupons suffered any corrosion or erosion effects from the brine exposure as visible under microscopic comparison to similar unexposed samples. The test coupons were 2.5 cm x 2.5 cm (1 in x 1 in) in size. The material types are listed on page 8. Five of the eight coupons were 1.5mm (0.06 in.) thick. The remaining three were 0.25 mm (0.01 in) thick. The latter three samples (no. 6, 7, 8) were apparently of inadequate thickness to provide the strength needed for the dynamic loads from the two-phase flow. This was evidenced by the brittle fracturing of the three thin specimens numbered 6, 7 and 8. The coupons were reduced in size from the 2.5 cm square to a disc of 0.9 cm dia which is slightly larger in diameter than the screw used to secure them to the separator disc. The failure may have been induced by hydrogen embrittlement as the edges under microscopic examination were sharp and quite irregular, this indicating erosion or corrosion had little to do with the failure of the three samples. The only evidence of corrosion was seen on the aluminum surfaces. The aluminum rotor rim had been coated with an electroless nickel application of about 0.005mm (0.0002 in) thickness. The nickel coating remained intact, however the inherent porosity of the nickel coating permitted localized corrosion of the aluminum. This was evidenced by aluminum oxide scale sites about 0.5 mm (0.02 in) in diameter spaced randomly about 1.3 cm (0.5 in) apart. This point type corrosion occurred on the separator rim and disc on the sides away from the brine flow. These surfaces were exposed to a predominantly steam flow. The separator surface exposed to the brine flow was free of the localized corrosion sites since it had a uniform layer of scale. There was a higher amount of silica in this scale with respect to the iron sulfide which is most likely the reason for the scale being softer.

There were several areas in the separator system where non-metals were used. The most important one with respect to system performance was the nozzle wall. To avoid the expensive process of machining a contoured wall inside a 63.5 cm (25 in) long section of 6.5 cm (2.5 in) diameter pipe, the inside nozzle contour was fabricated of epoxy. The nozzle contour was first machined on a metal mandrel which was wrapped with three layers of fiberglass cloth. This assembly was installed in the pipe section and epoxy filled the space around the mandrel.

The epoxy was cured at 120°C (250°F) with a vacuum applied to maximize epoxy impregnation of the fiberglass. The surface finish achieved was equivalent to the 32 micro-finish of the machined mandrel surface. This process of nozzle fabrication was developed in the event a full admission system were to be built, then as many as a dozen nozzles would be needed. In that case, since the mandrel fabrication is approximately six times the cost of one nozzle, the price per nozzle would be reduced to \$500 each as compared to about five times that for an all metal technique. Only one of the nozzles was used throughout the 80 hours of steam/water calibration tests and the 117 hours of brine flow tests. There was no deterioration of the epoxy or fiberglass surface. The throat diameter remained unchanged at 2.88 cm (1.133 in) diameter. There was a similar iron sulfide scale buildup which was 0.51 mm (0.02 in) thick. Removal of the scale showed an epoxy surface similar in appearance to the untested nozzle.

There were two other applications of nonmetallic materials. One was used as a coating to protect the aluminum housing that enclosed the separator rotor. The coating is a fluorocarbon resin sold under the product name Kynar Coatings by the Pennwalt Corp. of Philadelphia, Pa. The material was applied as a spray with a 0.5 mm (0.02 in) coating thickness with a cure at 150°C (300°F) which is the maximum service temperature. The other material was silicone rubber sealant applied between the three aluminum sections that assemble to form the separator housing. Both applications remained serviceable throughout the brine test series. Both the rubber and Kynar coatings were exposed to the full separator flow velocities up to 150 m/s (500 ft/s) although the latter had a larger surface area exposure, 0.44 m² (4.71 ft²) compared to the edge-wise exposure of the rubber sealant bead. Both applications achieved protection of the aluminum surfaces to which they were applied without loss of bond strength.

IV. ROTARY-SEPARATOR FLASH-STEAM COMPARISON

One of the primary advantages apparent for the rotary separator over a conventional flash system is the elimination of the gravity type flash separators. It is these large tanks that have shown scale plugging tendencies which can be eliminated by a rotary separator system. The rotary separator would have an additional power output by conversion of the brine enthalpy drop to output hydraulic or rotary shaft power, depending on whether a stationary or rotating liquid pickup is used. Both the rotary and flash system would have an approximately equal power output from conventional turbines operating on the separated steam. The following description will show how the brine energy conversion will increase the output of the rotary separator 27% to 51% above the flash system.

For this comparison both the rotary separator and flash system will utilize a single stage flash from wellhead conditions (pressure, $P_1 = 0.55$ MPa (80 psia), steam quality, $x_1 = 0.05$) to 0.9 MPa (14.7 psia) with steam quality, $x_2 = 0.13$. These particular conditions were chosen to provide an equivalent basis for comparison but primarily to permit use of the performance factors measured in the rotary separator field tests. The comparison of the two system flow schematics and thermodynamic state points are summarized in Fig. 11. The flash steam uses a throttling orifice to expand from the wellhead pressure to atmospheric pressure which serves only to generate steam. The rotary separator expansion from wellhead to atmospheric pressure occurs in the convergent/divergent nozzle where in addition to steam generation, a portion of the enthalpy drop is converted to brine kinetic energy. Thus, more energy is extracted from this expansion process. This is shown in the temperature-entropy diagram where the rotary separator expansion (point 2 to 3) is closer to an isentropic expansion (point 2 to a), than the flash process expansion (point 2 to 3). After the expansion the steam and brine are separated by the respective separators in each system. In both cases the steam flows to a steam turbo-generator where a power output of 5.91 kWe per kgm/s (2.68 kW/lbn/s) is obtained with a 70% efficient turbine. There is 11% less power from the steam turbine in the rotary separator because of the 6% lower steam quality. The flash system steam quality is calculated assuming a constant enthalpy expansion at the throttle orifice where enthalpy is evaluated as:

$$h_n = x_n h_{g_n} + (1 - x_n) h_{f_n} \quad (10)$$

for the conditions of Table V, therefore solving the expression:

$$h_{\frac{2}{2}} = h_{\frac{2}{3}} \quad (11)$$

for $x_{\frac{2}{3}}$ yields:

Single Flash System

Rotary Separator

FIGURE 11
ROTARY SEPARATOR/SINGLE FLASH STEAM SYSTEM COMPARISON

$$X_{\bar{3}} = \frac{X_{\bar{2}} h g_{\bar{2}} + (1 - X_{\bar{2}}) h f_{\bar{2}} - h f_{\bar{3}}}{h g_{\bar{3}} - h g_{\bar{2}}} \quad (12)$$

Substituting the state point conditions above yields:

$$X_{\bar{3}} = \frac{0.5(1183.1) + 0.95(282) - 180.2}{1150.5 - 180.2} = 0.151 \quad (13)$$

For the rotary separator system, the nozzle exit quality, X_3 , is evaluated as an isentropic process which then defines an enthalpy change, Δh_i :

$$\Delta h_i = h_2 - h_i \quad (14)$$

Since the nozzle expansion is not isentropic, but achieves an enthalpy extraction, Δh_{23} , which is a fraction of the isentropic enthalpy change defined by the nozzle efficiency, η_n . Thus:

$$h_{23} = \eta_n \times \Delta h_i \quad (15)$$

For the state point conditions of Fig. 11, the isentropic enthalpy change is calculated by first evaluating the nozzle exit quality for an isentropic expansion. Equating the entropy at points 2 and 3 yields:

$$s_2 = s_3 \quad (16a)$$

where:

$$s_n = X_n s_{gn} + (1 - X_n) s_{fn} \quad (16b)$$

$$x_3 = \frac{x_2 s_{g_2} + (1-x_2)s_{f_2} - s_{f_3}}{s_{g_3} - s_{f_3}} \quad (17)$$

Substitution of the state point conditions from Table III yields:

$$x_3 = \frac{0.05 \times 1.6209 + 0.95 \times 0.4533 - 0.3121}{1.7568 - 0.3121} = 0.138 \quad (18)$$

The enthalpy change corresponding to the isentropic expansion is then found by evaluating the enthalpy for state points 2 and a. The difference according to Eq. (14) is:

$$\begin{aligned} \Delta h_i &= 0.05 \times 1183.1 + 0.95 \times 282 - 0.1381 \times 1150.5 - (1-0.1381) \times 180.2 \\ \Delta h_i &= 29.87 \text{ J/gm (12.86 BTU/lbm)} \end{aligned} \quad (19)$$

For a nozzle efficiency, $\eta_n = 0.7$, the actual enthalpy change will be according to Eq. (15):

$$\Delta h_{2,3} = 0.7 \times 29.87 = 20.91 \text{ J/gm (9.00 BTU/lbm)} \quad (20)$$

The nozzle exit enthalpy is found by subtracting the values of $\Delta h_{2,3}$ from the value of h_2 from Eq. (10) in the relationship:

$$h_3 = h_2 - \Delta h_{2,3} = 738.8 \text{ J/gm (318.06 BTU/lbm)} \quad (21)$$

The power output from the rotary separator is then evaluated by Eq. (23) which is the product of the machine efficiency (η_e), enthalpy change ($\Delta h_{2,3}$) and total flow rate ($\dot{m}_+$). The machine efficiency is defined by Eq. (22).

$$\eta_e = (\eta_{sa} \times \eta_s \times \eta_p \times \eta_d) \quad (\text{Ref. Table II}) \quad (22)$$

$$PRS = \eta_e \times \Delta h_{2,3} \times \dot{m}_+ \text{ or } = \eta_o \times \Delta h_{2,3} \times \dot{m}_+ \quad (23)$$

For a machine efficiency of:

$$\eta_e = (0.74 \times 0.89 \times 0.90 \times 0.85) = 0.504 \quad (24)$$

and mass flowrate of $\dot{m}_1 = 2.2 \text{ kgm/s (1.00 lbm/s)}$, the rotary separator specific power output is:

$$P_{RS} = \frac{0.504 \times 9 \times 1}{0.948} = 4.65 \text{ kW per lbm/s} \quad (25)$$

The power output from the steam turbine in both the rotary separator and flash system is calculated in a similar manner

by first evaluating the turbine exit quality for an isentropic expansion from point 4 to 5:

$$x_{5i} = x_{5i} = \frac{s_{g_4} - s_{f_5}}{s_{g_5} - s_{f_5}} = \frac{1.7566 - 0.1749}{1.9200 - 0.1749} = 0.91 \quad (26)$$

The exit enthalpy, h_5 , is evaluated:

$$h_5 = x_{5i} h_{g_5} + (1 - x_{5i}) h_{f_5}$$

$$h_5 = 0.91 \times 116.2 + (1 - 0.91) \times 94$$

$$h_5 = 1024.2 \text{ BTU/lbm}$$

The enthalpy extraction in the turbine, with efficiency, $\eta_t = 0.7$, then is:

$$\Delta h_{45} = \eta_t \times (h_4 - h_5) = 0.7 (1150.4 - 1024.2) = 88.34 \text{ kW/lbm/s} \quad (27)$$

The power output from the steam turbo-generator, P_s , is:

$$P_s = \dot{m}_1 \times x_{\frac{1}{3}} \times \Delta h_{45} = 1 \times 0.151 \times 88.34 = 14.07 \text{ kW/lbm/s}$$

$$P_s = \dot{m}_1 \times x_{\frac{1}{3}} \times \Delta h_{45} = 1 \times 0.142 \times 88.34 = 13.23 \text{ kW/lbm/s}$$

The power output from the rotary separator thus is the sum of the rotary separator and steam turbines or:

$$P_{RS+} = P_{RS} + P_s = 4.65 + 13.23 = 17.88 \text{ kW/lbm/s}$$

or the rotary separator extracts:

$$(17.88/14.07) \times 100 = 27\%$$

more power than a single flash steam system. For rotary separator of optimized design where the efficiencies of Table II are achieved, $\eta_o = 0.62$, the power output would be increased to:

$$P_{RS+} = 8.41 + 12.88 = 21.3$$

Thus, the rotary separator system would add 51% resource conversion to a flashed steam system.

The following section performs a similar comparison for a two-stage flash steam system for a 50 MW power plant design, Ref. 11, and a two-stage rotary separator system.

V. ROTARY SEPARATOR/TURBINE SYSTEM DESIGN

The primary emphasis on geothermal energy conversion has been in the area of central-station electric power generating systems. This direction has somewhat been forced by the energy crisis and the need to develop geothermal energy as an alternate source of electrical power at a time when the only technology sufficiently developed is that of central-station turbo-generators. Considering the nature of the majority of geothermal resources being high-salinity water-dominated resources and of low power level (tens of megawatts) as compared to the thousand megawatt size technology being applied, there appears to be a need for a power conversion system in the low power levels and capable of operating in the high salinity scaling environment. The rotary separator system is one of several new technologies being developed with the goal of obtaining more complete resource utilization by energy extraction from the brine in addition to the flashed steam. It will be shown that the low cost of the rotary separator with both a liquid turbine and subatmospheric steam turbine will provide a near-term means of utilizing single or multiple wellhead geothermal reserves. This can be done without the major expense of a network of wellhead delivery systems to a central geothermal power station. Individual 5 to 50 MWe modular power systems would provide a means of proving a new well site to be operational while still producing electrical power and fresh water from the condensed steam.

The preliminary design of a rotary separator with liquid turbine shows the system for a 50 MWe output would have a full admission nozzle and a rotor of 2.16 m (7.1 ft) diameter. The system is shown schematically in Fig. 12. Shown is the full admission nozzle consisting of nine individual nozzles with 19.1 cm (7.53 in) diameter throat and 127 cm (50 in) overall length. The divergent portion diverges to a rectangular cross-section, 21.6 x 22.6 cm (8.5 x 8.9 in) at a 24 degree angle.

Each nozzle is separately valved to provide power level variation. The control valve at the inlet of each nozzle would be used to stop flow for partial admission operation or adjust nozzle inlet quality by a partial expansion of the brine to steam across the valve. Fig. 12 also shows the details of the rotor and liquid turbine. The liquid probe is supported by bearings and a hollow shaft which ducts the flow from the probe inlet of 13.2 x 13.2 cm (5.2 x 5.2 in) cross-section to the stationary liquid outlet block. The shaft connects to the electrical generator. The separator is bearing mounted concentrically to the probe shaft to provide independent rotation of the separator probe. The separator rim is 30 cm (12 in) wide which is 50% larger than the width of geometric nozzle impingement zone on the rim. The width, w , is calculated from the following equation, as a function of nozzle inclination angle, T_2 , rotor

FIGURE 12 2.2 METER ROTARY SEPARATOR DESIGN CONCEPT

diameter, D, and angle, B, shown in Fig. 12:

$$w = 2(\sin B \times D/2 \times \tan T_2)$$

The circumferential probe barricade serves to prevent nozzle flow from impinging on the probe. The flow tranverses from the impingement zone to the liquid pickup through the holes in the probe barricade. A rotary separator of this size, 2.16m (85 in) diameter, with full admission nozzles and operating from a 5% quality, 0.55 MPa (80 psia) wellhead pressure would have an output of 5.78 MW. This is based on the nozzle exit area of 488 cm² (75.6 in²) for each of the nine nozzles. The throat to exit area required for the expansion to atmospheric pressure is 2.6 from Fig. B-3. Therefore, the throat area is 488/2.6 = 187cm² (75.6/2.6 = 29 in²). From Fig. B-4 the area required is 135 cm² (21 in²) per 45.5 kgm/s (100 lbm/s) of flow, therefore, the total flowrate will be:

$$\dot{m}_+ = 9 \times (29/21) \times 100 = 564.5 \text{ kgm/s (1242 lbm/s).}$$

For the system efficiency of $\eta_o = 0.343$, Table II, and isentropic enthalpy drop, $\Delta h_i = 12.83$, the output power according to Eq. (23) is $P_s = .343 \times 12.83 \times 1242 = 5.76 \text{ MW}$. A single flash system with the same flow rate would have an output of 3.33 MW.

A recent design study, Ref. 11, shows a detailed analysis of a two-stage flash system with a 33 kW/kgm/s (15 kW/lbm/s) specific output operating from a wellhead pressure of 0.79 MPa (115 psia). With a well flowrate of 1515 kgm/s (3333 lbm/s) the system would have a 50 MWe net output. The design shows a partial flash from 0.79 to 0.34 MPa (115 to 50 psia) which yields 125 kgm/s (275 lbm/s) steam flowrate and 30 MWe output from the high pressure turbine. The remaining brine is flashed in the second stage separator, 0.34 to 0.10 MPa (50 to 15 psia) which produces an additional 93 kgm/s (205 lbm/s) of steam and 20 MWe power output in the low pressure stage of the steam turbine. This is a gross output of 50 MW with 9 MW supplied to auxiliaries. This represents a 37.6% utilization of the available enthalpy extraction for an isentropic process system operating from 0.73 to 0.14 MPa (115 to 2 psia). Replacement of the two gravity type flash separator tanks with rotary type separators would provide a 16% increase in output power. In addition, the replacement of the static separators eliminates the component most likely to be adversely effected by scale deposits. Fig. 13 shows schematically a two-stage flash system with rotary separator/turbine replacing the flash separator tanks.

For the first stage rotary separator, the component efficiencies calculated according to Table II are:

1. $\eta_n = 0.71$ from Fig. B-5 for 0.79 MPa (115 psia)
2. $\eta_{sa} = [\cos 24 \times (41.3/42.5)]^2 = 0.782$ which utilizes a nozzle inclination of 24° and a four rim separator
3. $\eta_s = 0.95$ a direct result of the low separator speed of 800 rpm

FIGURE 13 TWO STAGE ROTARY SEPARATOR/STEAM TURBINE

$$4. \quad \eta_p = 0.9$$

5. $\eta_d = 0.9$ the pressure recovery is negligible and the 0.1 diffuser loss is based on frictional losses

$$\eta_o = 0.71 \times 0.786 \times 0.95 \times 0.90 \times 0.90 = 0.429$$

For the expansion from 115 to 50 psia at $X_1 = 0.025$, the isentropic enthalpy drop is 3.65 BTU/lbm. Therefore, from Eq. (23) the power output:

$$P_{R1} = \eta_o \times \dot{m} \times \Delta h_i = 0.429 \times 3333 \times (3.65/0.948)$$

$$P_{R1} = 5.4 \text{ MWe}$$

assuming a 98% efficient electrical generator.

For the two second-stage rotary separators, the efficiency terms are the same except for a lower nozzle efficiency of $\eta_n = 0.6$ and lower diffuser efficiency $\eta_d = 0.83$, because of a pressure requirement for reinjection:

$$\eta_o = 0.6 \times 0.786 \times 0.95 \times 0.90 \times 0.83 = 0.335$$

The power output for an isentropic enthalpy drop of 2.78 BTU/lbm and flowrate of $3333 \times (1 - 0.085) = 3049.7$ lbm/s is:

$$P_{R2} = 0.335 \times 3049.7 \times (2.78/0.948) = 3.0 \text{ MWe}$$

The high pressure steam turbine stage would have a power output quite similar to that of Ref. 1, differing by the slight reduction in steam output from the rotary separator:

$$P_{S1} = \eta_e \times \dot{m}_t \times X_2 \times \Delta h_i = 0.7 \times 3333 \times 0.085 \times 181.6$$

$$P_{S1} = 38 \text{ MWe}$$

Similarly for the two second-stage turbines:

$$P_{S2} = 0.7 \times 0.09 \times (1 - 0.085) \times 3333 \times 103.7 = 21 \text{ MWe}$$

The total net system output, P_o , is the sum of the three rotary separator/turbines and two steam turbines less the auxiliary power requirements, P_a , which from Ref. 11 is 9 MWe:

$$P_o = P_{R1} + P_{R2} + P_{S1} + P_{S2} - P_a = 5.4 + 3.0 + 38 + 21 - 9$$

$$P_o = 58.4 \text{ MWe}$$

This represents for the same resource flowrate an increase in output power of $58.4/50 = 1.168$. The system characteristics are summarized in Table IV.

COST ESTIMATE

A cost estimate for the two-stage combined rotary separator and steam turbines shows an installed cost of \$664 per kWe. The estimate is based primarily on the estimate of the two-stage flash steam system of Ref. 11, modified by elimination of the flash tank costs and addition of the rotary separator costs. The flash tank costs represented only 1% (\$290,000) of the total. The rotary separator system cost according to Table V is \$1,595,000 for the three rotary separator/turbine systems. This gives a total cost \$38,795,000 for the rotary separator system with the steam turbine system of Ref. 11. The specific cost of the system is reduced from \$744 per kWe for the two-stage flash system to \$664 per kWe for the combined rotary separator and steam turbine systems, representing a 12% cost reduction.

VI. CONCLUSIONS AND RECOMMENDATIONS

The operation of the rotary separator in the high saline environment for 117 hours has demonstrated the characteristics necessary to warrant further development as a total flow geothermal energy conversion system. Some of these characteristics are:

1. Two-phase nozzle design that provide steam/brine enthalpy conversion to kinetic energy with 70% efficiency measured at 0.55 MPa (80 psia) and 85% efficiency predicted at nozzle inlet pressures above 2.07 MPa (300 psia).
2. Rotary separator capability of steam/brine separation while achieving 86% brine velocity transfer and a steam quality of 99.5%.
3. A tested liquid pickup design that incorporates a two-phase diffuser for efficient conversion of separated brine kinetic energy to pressure head. The pickup demonstrated scale handling capabilities in brine flows with 115,000 ppm total dissolved solids.
4. The short residence time of the brine in the rotary separator provides an important means of eliminating large scale deposits characteristic of flash systems.
5. The relative simplicity of the rotary separator compared to conventional steam turbines or other total flow conversion systems provides inherent reliability and low cost.
6. Based on the existing 76 cm (30 in) diameter rotary separator, a cost projection for a 216 cm (85 in) diameter separator with 5 MWe liquid turbine would show a capital cost of \$160,000. The comparison with a single stage flash system shows a 48.6% increase in resource utilization with a rotary separator. For a double stage flash system of 50 MWe capacity, the replacement of the plugging prone flash tanks with rotary separators provides an increase in resource utilization of 16% and installed cost reduction from \$744 to \$664 per kWe.

TABLES

Table
No.

I	Test data and results summary for geothermal field tests of rotary separator
II	Rotary separator component efficiencies
III	Performance calculation comparison for rotary separator/turbine system and flashed steam system
IV	System description summary for two stage rotary separator/turbine system
V	Cost estimate for two stage rotary separator/turbine system with steam turbine

TABLE I
TEST DATA

Scan No.	Hrs.	Pressure								Temperature								SPEED RPM									
		ΔP_1	ΔP_4	ΔP_{11}	P_2	P_3	P_5	P_6	P_{10}	T_1	T_2	T_3	$T_{4_{OF}}$	T_5	T_6	T_7	T_8		M_L	M_S	M_R	X_1	X_2	P_{O_1}	η_1	η_2	$\backslash R$
		in H ₂ O			in psig					in °F									in lbm/s			in kw					
1	0	0.5	0	1.0	147	10.0	8.0	0	17.0	-	-	-	-	-	-	-	-	1067	0.354	0	0.30	12.6	15.0	0.143	.10	.105	.98
2		5.0	0	3.5	147	16.0	12.0	0	17.0	349	-	-	-	-	-	-	221	1496	1.12	0	0.94	13.1	18.0	0.89	.145	.15	.99
3	1	6.5	0	4.5	146	18.0	15.0	0	17.0	352	-	-	-	-	-	-	222	1725	1.28	0	1.06	11.0	17.0	1.35	.176	.18	.97
4	2	10.0	0	6.5	147	24.0	19.5	0	27.0	350	-	-	-	-	-	-	222	2064	1.58	0	1.28	10.0	14.3	2.40	.253	.26	.95
5	3	14.0	0	9.5	147	31.0	25.0	0	29.0	353	-	-	-	-	-	-	222	2335	1.87	0	1.54	9.5	14.6	3.63	.243	.26	.97
6	4	14.0	0	9.3	146	31.0	25.0	0	30.0	353	-	-	-	-	-	-	222	2374	1.87	0	1.54	9.5	14.6	3.75	.26	.26	.97
7	5	18.8	0	13.0	147	38.0	30.3	0	36.0	355	-	-	-	-	-	-	223	2576	2.17	0	1.80	8.9	14.8	5.12	.26	.28	.98
8	6	10.5	0	7.0	148	26.0	20.0	0	25.0	355	-	-	-	-	-	-	223	1775	1.60	0	1.32	10.4	14.8	1.82	.17	.18	.96
9	7	8.0	0	5.5	150	19.0	15.5	0	13.0	339	-	-	-	-	-	-	222	1657	1.40	0	1.17	9.5	13.2	1.38	.18	.19	.96
10	8	7.8	0	5.2	148	19.0	15.5	0	13.0	339	-	-	-	-	-	-	221	1533	1.40	0	1.14	9.5	13.2	1.17	.152	.16	.94
11	9	11.8	0	8.0	149	27.0	21.0	0	11.0	343	-	-	-	-	-	-	221	1936	1.72	0	1.41	9.0	13.6	2.29	.20	.22	.95
12	10	11.5	0	7.8	150	25.0	20.0	0	20.0	341	270	265	220	224	-	223	-	1953	1.70	0	1.40	9.0	13.4	2.30	.208	.22	.95
13	11	15.0	0	10.5	150	31.0	25.0	0	20.0	343	-	-	-	-	-	-	-	2160	1.94	0	1.02	8.4	13.6	3.20	.22	.24	.97
14	12	14.5	0	10.2	150	31.0	24.5	0	19.0	344	-	-	-	-	-	-	-	2150	1.90	0	1.60	8.4	13.6	3.13	.22	.24	.97
15	13	20.0	0	15.0	150	0	32.0	0	0	347	282	280	-	225	-	-	-	2160	2.24	0	1.94	7.8	14.0	3.71	.20	.22	1.00
16	14	10.0	0	6.0	150	0	18.0	0	0	315	-	-	-	-	-	-	-	1200	1.58	0	1.22	6.6	10.7	.81	.10	.105	.87
17	15	10.0	0	6.0	150	0	18.0	0	0	315	-	-	-	-	-	-	-	1200	1.58	0	1.22	6.6	10.7	.81	.10	.105	.87
18	16	6.0	9.0	3.5	147	20.0	15.0	0	8.0	334	259	-	-	-	-	-	219	1819	1.22	.051	.93	12.7	16.2	1.44	.174	.18	.88
19	17	5.5	32.0	3.5	148	29.0	20.0	0	8.0	-	-	-	-	-	-	-	-	2370	1.17	.108	.94	16.0	20.0	2.34	.22	.23	.92
20	18	8.2	28.0	5.5	150	34.0	25.0	0	9.0	337	279	-	-	-	-	-	223	2569	1.43	.11	1.17	14.0	19.0	3.36	.23	.24	.94
21		11.9	53.0	8.0	150	51.0	38.0	0	11.0	344	296	287	219	-	-	-	227	3227	1.72	.14	1.41	15.5	20.4	6.39	.23	.24	.95
22		16.0	31.0	11.0	150	52.0	39.0	0	15.0	343	300	291	220	-	-	-	227	3195	2.00	.13	1.66	12.2	19.0	7.26	.23	.24	.96

TABLE I Cont'd

Scan No.	Hrs.	Pressure								Temperature								SPEED RPM									
		ΔP_1	ΔP_4	ΔP_{11}	P_2	P_3	P_5	P_6	P_{10}	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8		M_L	M_S	M_r	K_1	K_2	P_{01}	η_1	η_2	R
		in H ₂ O			in psig					in °F									in lbm/s			in kw					
23	19	4.0	10.0	2.5	148	16.0	11.5	0	10.0	327	250	245	217	-	-	-	220	1508	1.00	.051	.79	13.5	16.2	0.81	.147	.15	.90
24	20	4.0	11.0	2.5	148	17.0	13.0	0	7.0	328	253	247	217	221	169	-	219	1485	1.00	.051	.79	13.5	16.2	0.78	.143	.15	.90
25	21	4.5	11.0	3.0	149	18.0	14.0	0	10.0	333	256	250	217	221	-	-	220	1544	1.06	.051	.87	13.2	16.0	0.90	.152	.16	.93
26	22	4.5	13.3	3.0	149	19.0	13.5	0	9.0	329	257	250	218	222	168	-	220	1547	1.06	.059	.87	13.8	16.6	0.90	.15	.16	.93
27	23	4.0	13.5	2.8	149	18.5	13.5	0	8.0	331	256	249	217	221	168	-	220	1553	1.00	.059	.87	14.1	16.9	0.86	.15	.156	.98
28	25	3.0	13.5	2.5	149	18.0	12.0	0	8.0	329	256	248	218	222	164	-	220	1465	0.87	.059	.79	15.0	18.0	0.66	.13	.133	1.04
29	27	4.0	10.5	3.5	158	20.0	14.0	0	14.0	321	262	253	218	222	194	-	220	1444	1.00	.061	.935	13.3	16.5	0.74	.11	.11	1.06
30	29	5.0	10.0	3.5	148	22.0	15.5	0	10.0	334	262	254	218	222	164	-	221	1647	1.11	.055	.935	13.3	16.8	1.08	.14	.15	.96
31	31	4.8	10.5	3.3	146	21.0	14.5	-.05	10.0	331	259	251	217	221	169	-	220	1565	1.10	.056	.91	13.3	16.6	0.96	.124	.13	.96
32	33	4.4	11.5	3.0	147	20.0	14.0	-.05	10.0	333	258	250	217	220	173	-	220	1533	1.05	.059	.97	13.7	17.0	0.88	.12	.12	.94
33	34	4.3	11.2	3.0	147	20.0	13.5	-.05	10.0	332	257	248	217	220	172	-	219	1503	1.05	.059	.97	13.7	17.0	0.88	.12	.12	.94
34	35	4.4	11.0	3.0	145	20.5	14.0	-.05	10.0	334	259	250	217	221	172	-	220	1554	1.05	.059	.87	13.7	17.0	0.90	.12	.12	.94
35	36	4.5	10.5	3.0	144	21.0	14.5	-.05	6.0	336	260	251	217	221	176	-	220	1594	1.05	.059	.87	13.7	17.0	0.90	.12	.12	.94
36	37	4.4	10.8	3.0	145	21.0	14.0	-.05	6.0	335	259	250	217	220	178	-	220	1546	1.05	.057	.87	14.0	17.3	0.89	.12	.12	.95
37	38	4.2	11.2	3.0	145	20.5	13.5	-.05	6.0	331	258	249	217	220	178	-	219	1500	1.02	.057	.87	14.2	17.4	0.82	.12	.12	.97
38	39	4.1	11.4	2.9	147	20.0	13.5	-.05	6.0	331	258	248	217	221	177	-	220	1476	1.01	.057	.87	14.2	17.4	0.78	.12	.12	.98
39	40	4.0	11.4	2.7	146	19.5	13.0	-.05	6.0	331	258	248	218	221	177	-	220	1448	1.00	.057	.82	14.3	17.5	0.75	.12	.12	.94
40	46	5.0	13.5	3.2	150	24.0	15.0	-.05	12.0	326	264	252	219	221	164	-	220	1558	1.12	.066	.89	14.3	17.7	0.966	.12	.12	.92
41	50	5.0	12.5	3.2	146	26.0	16.0	-	10.0	335	269	255	218	222	154	-	221	1601	1.12	.064	.89	14.0	17.5	1.02	.12	.12	.92
42	52	4.8	12.6	3.0	147	25.0	15.0	-	6.0	335	268	254	218	222	162	-	220	1530	1.09	.064	.89	14.2	17.6	0.91	.12	.12	.94
43	53	4.5	12.6	3.0	147	25.0	14.5	-	6.0	333	267	253	219	222	166	-	221	1435	1.06	.064	.89	14.4	17.8	0.78	.12	.12	.97
44	55	5.7	13.0	3.8	144	29.0	18.0	-	8.0	331	274	259	218	222	168	-	221	1526	1.19	.064	.98	13.8	17.2	0.99	.12	.12	.94
45	56	5.7	13.1	3.8	146	29.0	18.0	-	8.0	331	289	259	217	221	169	-	220	1475	1.19	.064	.98	13.8	17.2	0.92	.12	.12	.94
46	57	5.4	13.2	3.6	146	29.0	17.5	-	8.0	333	297	257	217	221	171	-	220	1450	1.16	.064	.95	13.9	17.3	0.87	.12	.12	.94

Table I Cont'd

Scan No.	Hrs.	ΔP_1	ΔP_4	ΔP_{11}	Pressure					Temperature								SPEED RPM	M_L	M_s	M_r	X_1	X_2	P_{01}	η_1	η_2	R
		in H ₂ O			in psig					in °F									in lbm/s			in kw					
47	58	6.0	12.5	3.9	146	31.0	18.5	-	8.0	335	298	260	217	221	173	-	220	1517	1.22	.064	.99	13.7	17.1	1.00	.12	.12	.93
48	59	5.7	12.3	3.7	145	31.0	18.0	-	8.0	332	275	258	216	220	174	-	220	1465	1.19	.064	.96	13.7	17.2	.91	.12	.12	.93
49	60	5.0	20.0	3.4	144	31.0	17.5	-	8.0	332	275	257	216	220	176	-	219	1533	1.12	.087	.92	15.0	19.0	.94	.12	.12	.94
50	68	4.0	8.0	1.0	150	20.0	12.0	-	0.0	323	259	244	211	219	178	-	217	704	1.0	.048	.90	13.6	16.5	.176	.03	.03	1.03
51	71	7.2	0	5.5	180	32.0	16.0	-	3.0	341	235	256	217	222	178	-	217	1444	1.407	0	1.173	9.7	13.4	1.044	.124	.13	.96
52	72	6.8	0	5.3	180	31.0	15.0	-	3.0	340	233	254	217	222	169	-	220	1380	1.367	0	1.151	9.9	13.4	.927	.113	.12	.97
53	73	6.5	0	5.0	182	30.0	14.5	-	2.0	339	236	253	217	221	172	-	220	1314	1.337	0	1.118	9.8	13.2	.821	.102	.11	.96
54	74	6.5	0	5.0	178	30.0	14.5	-	2.5	340	241	253	218	222	173	-	221	1322	1.337	0	1.11	9.8	13.2	.832	.104	.11	.96
55	75	6.2	0	5.0	179	29.0	14.0	-	2.0	337	234	251	217	221	171	-	220	1274	1.306	0	1.118	9.7	13.0	.754	.96	.10	.98
56	76	6.8	0	5.0	178	31.0	15.0	-	3.0	340	232	254	218	222	170	-	221	1357	1.367	0	1.118	9.8	13.3	.896	.119	.11	.94
57	77	6.3	0	5.0	178	30.0	14.5	-	3.0	340	246	253	217	221	171	-	220	1355	1.316	0	1.118	9.9	13.3	.86	.109	.113	.98
58	78	6.2	0	4.7	178	29.0	13.8	-	3.0	339	237	249	216	219	172	-	218	1264	1.306	0	1.084	10.0	13.3	.742	.95	.10	.96
59	79	6.8	0	5.2	176	31.0	15.0	-	5.0	339	238	253	217	220	172	-	219	1328	1.367	0	1.14	9.8	13.3	.858	.105	.11	.96
60	80	6.5	0	5.0	176	31.0	15.0	-	5.0	339	237	251	216	219	172	-	219	1344	1.337	0	1.118	9.6	13.3	.859	.107	.11	.96
61	81	6.5	10.0	5.4	139	41.0	19.0	0	5.0	317	288	259	216	220	168	-	219	1504	1.337	.067	1.162	11.1	15.2	1.076	.115	.12	.98
62	83	3.1	43.5	2.9	159	23.5	16.5	0	5.0	316	262	254	213	219	168	-	218	2000	1.019	.118	.85	16.6	20.2	1.450	.190	.19	.94
63	84	4.5	40.0	3.0	159	27.0	16.0	0	5.0	317	261	253	213	219	129	-	217	1968	1.230	.113	.87	14.8	18.4	1.690	.190	.19	.79
64	85	3.0	41.0	2.9	159	3.0	16.0	0	5.0	319	261	253	214	220	131	-	217	1965	1.000	.114	.85	16.5	20.1	1.340	.190	.19	.95
65	86	3.4	41.0	3.0	160	3.0	16.0	0	6.0	324	262	253	215	220	131	-	218	1999	1.070	.114	.87	16.7	20.3	1.500	.190	.19	.92
66	87	3.2	40.0	2.8	159	23.0	16.0	0	6.0	329	263	284	216	221	137	199	219	2033	1.040	.113	.84	16.9	20.5	1.520	.190	.19	.92
67	88	4.0	36.0	3.6	16	28.0	19.5	0	8.0	332	269	261	216	221	137	199	219	2245	1.160	.110	.95	15.6	19.2	1.520	.190	.19	.93
68	89	4.0	35.5	3.5	?	28.0	19.0	0	7.0	335	270	261	216	221	137	199	219	2253	1.160	.110	.95	15.6	19.2	2.090	.190	.19	.93
69		4.0	35.5	3.5		28.0	25.0	0	7.0	335	270	261	216	221	137	199	219	2555	1.160	.110	1.05	15.6	19.2	3.090	.190	.19	.96
70	90	4.9	32.0	4.5		23.0	24.0	0	9.0	341	277	269	216	223	137	196	221	2510	1.280	.100	1.06	15.6	19.2	3.090	.190	.19	.96

TABLE I Cont'd

Scan No.	Hrs.	Pressure								Temperature								SPEED RPM									
		ΔP_1	ΔP_4	ΔP_{11}	P_2	P_3	P_5	P_6	P_{10}	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8		M_L	M_S	M_R	X_1	X_2	P_{01}	η_1	η_2	R
		in H_2O			in psig					in $^{\circ}F$									in lbm/s			in kw					
71	91	4.6	61.0	4.4	160	41.0	27.5	.3	9.0	340	286	274	216	224	137	200	222	2850	1.24	.100	1.06	14.7	19.2	3.6	.29	.30	.99
72	92	4.4	61.0	4.5	160	41.0	27.0	.3	8.0	338	286	274	216	224	168	201	223	2824	1.21	.100	1.06	14.9	20.1	3.45	.28	.29	1.01
73	93	4.0	62.0	4.5	160	41.0	27.0	.3	8.0	340	286	274	216	224	155	201	223	2829	1.16	.100	1.06	15.2	20.4	3.53	.28	.29	1.06
74	94	4.5	65.0	5.0	160	43.0	30.0	.4	7.0	342	291	278	217	225	158	201	224	2970	1.23	.170	1.12	19.0	24.5	3.84	.27	.28	1.06
75	95	4.5	60.0	5.0	159	46.0	30.0	.3	7.0	341	290	277	216	224	157	200	224	2902	1.22	.170	1.12	19.0	24.5	3.67	.26	.26	1.06
76	96	4.0	60.0	5.0	159	45.0	30.0	.3	7.0	344	291	277	217	225	158	200	224	2909	1.16	.170	1.12	20.0	25.0	3.49	.25	.26	1.12
77	97	6.0	36.0	7.3	158	48.0	32.0	.3	6.5	350	294	282	217	225	161	203	224	2905	1.42	.130	1.35	15.6	22.0	4.26	.25	.26	1.14
78	98	6.2	35.0	7.3	158	48.0	32.0	.3	6.5	347	294	282	217	225	167	205	224	2874	1.44	.130	1.35	15.9	21.4	4.26	.25	.26	1.10
79	99	8.0	33.5	9.0	156	52.0	36.0	.4	7.0	350	299	287	217	226	167	195	225	3011	1.64	.133	1.50	14.6	20.9	5.3	.27	.28	1.07
80	100	6.5	28.5	7.5	156	46.5	32.0	0	6.5	346	292	279	216	224	161	195	223	2782	1.48	.118	1.37	15.0	20.8	4.06	.23	.24	1.08
81	103	6.5	0	10.0	160	34.0	25.0	0	6.0	345	264	271	214	221	178	-	221	2061	1.74	0	1.58	8.6	13.8	2.62	.21	.232	1.056
82		9.0	0	10.0	160	33.0	25.0	0	12.0	346	233	269	215	221	182	-	221	2025	1.73	0	1.58	8.6	13.8	2.53	.206	.22	1.05
83		7.0	4.0	7.0	160	35.0	25.0	0	8.0	345	275	269	214	221	186	-	220	2207	1.53	.040	1.32	11.0	16.0	2.65	.214	.224	1.00
84	104	7.0	4.0	7.3	160	35.0	25.0	0	8.0	342	273	268	214	220	188	-	220	2174	1.53	.050	1.35	11.0	16.0	2.58	.218	.218	1.02
85		12.0	0	12.0	160	39.0	29.0	0	11.0	348	265	277	215	221	188	-	221	2290	2.00	0	1.73	8.4	14.0	3.74	.23	.254	1.005
86		11.5	0	12.0	160	39.0	29.0	0	11.0	347	248	277	215	222	185	-	221	2254	2.00	0	1.73	8.4	14.1	3.74	.256	.28	1.005
87		7.8	11.1	8.0	160	41.0	28.0	0	9.0	344	284	275	214	221	184	-	221	2527	1.60	.070	1.41	12.0	17.4	3.67	.22	.23	1.014
88	105	7.5	11.2	7.8	160	41.0	28.0	0	9.0	343	285	275	215	222	186	-	221	2508	1.58	.070	1.40	12.0	17.5	3.55	.21	.22	1.02
89		16.0	0	16.0	160	47.0	35.0	0	10.0	350	289	287	216	223	191	-	222	2488	2.30	0	2.00	8.0	14.3	5.10	.23	.25	1.01
90		16.0	0	16.0	160	47.0	35.0	0	10.0	350	269	286	215	223	188	-	222	2499	2.30	0	2.00	8.0	14.3	5.15	.24	.26	1.01
91		11.0	10.4	11.0	160	49.0	35.0	0	10.0	347	294	283	215	223	180	-	222	2758	1.90	.70	1.66	10.8	17.2	5.20	.25	.26	1.005
92	106	11.0	10.4	11.0	160	49.0	35.0	0	10.0	347	294	283	215	223	179	-	222	2760	1.90	.70	1.66	10.8	17.2	5.20	.25	.26	1.005
93		19.5	0	19.5	160	53.0	40.0	0	11.0	353	289	291	216	224	190	-	223	2650	2.60	0	2.21	10.8	14.6	6.40	.23	.25	1.01
94		19.0	0	19.0	160	53.0	40.0	0	11.0	353	259	291	216	224	193	-	223	2639	2.50	0	2.18	7.5	14.6	6.20	.234	.25	1.01
95		13.5	8.0	14.0	160	55.0	40.0	.5	8.0	350	299	289	216	224	177	-	223	2883	2.10	.07	1.87	10.0	17.0	6.30	.31	.33	1.03

TABLE I Cont'd

Scan No.	Hrs.	Pressure								Temperature								SPEED RPM	M_L M_S M_r in lbm/s			X_1	X_2	P_{O_2} in kw	η_1	η_2	R
		ΔP_1 in H_2O	ΔP_4	ΔP_{11}	P_2	P_3	P_5	P_6	P_{10}	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8										
96	107	13.5	8.5	13.5	160	55.0	40.0	.5	9.0	350	299	289	215	223	175	-	223	2882	2.10	.070	1.84	10.0	17.0	6.30	.31	.33	1.01
97		22.5	0	23.0	160	59.0	45.0	.5	10.0	355	299	296	216	225	190	-	224	2805	2.70	0	2.40	7.1	14.8	7.70	.27	.30	1.02
98		23.0	0	25.0	160	59.0	45.0	.5	8.0	354	280	297	217	225	174	-	224	2719	2.80	0	2.50	7.1	14.0	7.30	.25	.27	1.06
99		17.0	10.5	17.5	160	62.0	45.0	.6	8.0	351	306	296	216	225	169	-	225	2979	2.40	.080	2.10	9.7	17.0	7.50	.23	.25	1.02
100	108	17.5			Well Pressure Fluctuating																						
101		27.0	0	28.5	160	66.0	50.0	.6	10.0	355	305	303	217	225	191	-	225	2819	3.01	0	2.67	6.5	14.8	8.50	.23	.25	1.04
102		27.0	0	29.0	160	65.0	49.5	.6	10.0	355	304	302	217	225	185	-	225	2823	3.01	0	2.69	6.6	14.8	8.50	.23	.25	1.05
103		19.5	11.0	20.0	160	67.0	49.0	.6	8.0	353	311	300	217	226	178	-	225	3096	2.56	.084	2.24	9.4	17.4	8.72	.24	.25	1.025
104	108	23.0	9.0	24.0	158	72.0	52.0	.8	9.0	355	314	304	217	226	179	-	226	3224	2.78	.078	2.45	8.9	17.2	10.30	.27	.29	1.036
105		33.0	0	33.5	158	70.0	54.0	.9	13.0	356	308	307	217	226	187	-	226	3017	3.32	0	2.89	6.2	14.9	10.80	.248	.279	1.023
106		33.0	0	34.5	158	72.0	55.0	.9	12.0	357	292	308	217	226	185	-	227	3052	3.32	0	2.94	6.2	15.0	11.00	.253	.285	1.039
107	109	24.0	8.5	24.0	156	74.0	54.0	.9	9.0	354	316	305	218	227	173	-	227	3271	2.84	.077	2.45	8.5	17.0	10.80	.272	.298	1.013
108	110	23.0	8.5	23.5	160	73.0	53.0	.9	9.0	354	315	304	217	227	175	-	-	2335	2.776	.080	2.42	6.1	17.0	10.30	.306	.348	1.02
109	111	38.0	0	38.5	155	76.0	58.0	1.0	11.0	357	317	311	218	228	182	-	-	3090	3.56	0	3.02	5.9	15.0	12.10	.279	.325	1.02
110	112	38.0	0	38.5	155	77.0	59.0	1.1	11.0	357	309	312	218	228	181	-	228	3102	3.57	.010	3.10	5.8	15.0	12.20	.281	.326	1.02
111	113	30.0	8.5	32.0	156	82.0	60.0	1.2	10.0	356	322	312	218	228	180	-	228	3375	3.17	.080	2.83	5.6	17.0	12.80	.33	.386	1.049
112	114	30.0	8.5	32.0	156	82.0	60.0	1.2	10.0	356	322	312	218	228	180	-	228	3375	3.17	.080	2.83	5.6	17.0	12.80	.33	.386	1.049
113	115	50.0	0	40.5	155	85.0	65.0	1.2	14.0	356	308	316	218	228	186	-	227	3204	4.09	0	3.54	5.1	14.9	14.90	.279	.323	1.015
114	116	47.0	0	47.5	155	85.0	65.0	1.2	13.0	356	304	316	218	228	182	-	227	3194	3.90	0	3.45	5.0	14.9	14.40	.278	.322	1.02
115	117	35.0	14.0	34.0	156	89.0	65.0	1.0	11.0	355	326	314	219	228	180	-	228	3542	3.42	.110	2.92	7.9	17.4	15.3	.298	.333	1.00

TABLE II

TABLE II

Efficiency	Component	Power Term to be Evaluated for Efficiency Calculation	Efficiency Calculation	Efficiencies Existing Advanced	
η_n	Nozzle	kinectic energy of two phase nozzle exit flow	$P_n = \frac{1}{2} m V_n^2$	$\eta_n = P_n / m_T \Delta h_1$	0.68 0.86
η_{sa}	Separator Admission	kinectic energy component tangential to separator surface	$P_{sa} = P_n (\cos \theta \cos \alpha)^2$	$\eta_{sa} = (\cos \theta \cos \alpha)^2$	0.74 0.907
η_s	Separator	kinectic energy loss due to bearing friction & windage	$P_{SL} = (8.413E-10 x N_s^{2.6545})$	$\eta_s = (P_{sa} - P_{sL}) / P_{sa} = 1 - \frac{P_{sL}}{P_{sa}}$	0.89 0.95
η_p	Flow pickup	kinectic energy loss due to flow disruption by probe	$P_{pL} = \left(\frac{m}{2} R \times \frac{\pi^2 x R^2 x N_s^2}{30^2 g 738} \right) x (W/r_o) (\cos^2 B)$	$\eta_p = 1 - \frac{P_{pL}}{P_{sa} - P_{sL}}$	0.90 0.93
η_d	Diffuser	hydraulic output power at diffusion exit	$P_d = \frac{m_p}{\rho} \Delta P_p$ or $P_d = \eta_d x P_{pi} = \eta_d x (P_{sa} - P_{sL} - P_{pL})$ (where $\eta_d = .85 - r_v x K$)	$\eta_d = \dot{V} \Delta P / (P_{sa} - P_{si} - P_{pi})$ for measured pressure recovery values = ΔP or $\eta_d = 0.85 - r_v x$	0.85 0.91
		$\eta_e = \eta_{sa} x \eta_p x \eta_d = .74 x .89 x .9 x .85 = 0.504, 0.729$			
				$\pi = .343$	0.62

TABLE III

Calculated Performance Comparison for a Rotary Separator/Turbine System & Flashed Steam System

	Symbol	Definition	Units	Rotary Separator Full Admission (85" dia)	Flashed Steam
Total input flowrate	$\dot{m}$		lbm/s	1225	1225
Wellhead pressure	$P_2^{(2)}$		psia	80	80
quality	X_2		-	0.05	0.05
enthalpy	h_2		BTU/lbm	327	327
Separator/Turbine					
inlet pressure	P_3		psia	14.7	14.7
quality	X_3		-	⁽⁴⁾ 0.138/0.142	0.151
enthalpy	h_3		BTU/lbm	$314^2/318$	327
isentropic enthalpy drop	$\Delta h_{2s} = h_2 - h_3$		"	12.83	0
enthalpy extraction	$\Delta h_{2s} = \eta_n \Delta h_{23}$		"	⁽¹⁾ ($\eta_n = .7/.86$) 8.98/11.03	0
flash energy conversion output	$P_s = \frac{\Delta h_{23}}{23} \eta_e$		Mw	($\eta_e = .49/.72$) 5.69/9.75	
resource utilization efficiency	$\eta_1 = \frac{P_s}{\dot{m} \Delta h_{23}}$			0.372	
Steam Turbine					
input flowrate	$\dot{m}_4 = \dot{m} X_3$		lbm/s	174/170	1.85
quality	X_4, X_5		-	0.196	0.204
enthalpy	h_4, h_5		BTU/lbm	1150.4, 1024.2	1150.4, 1024.2
isentropic enthalpy drop	$\Delta h_{4s} = h_4 - h_5$		"	126.2	126.2
turbine enthalpy extraction	$\Delta h_{4s} = \eta \Delta h$		"	88.3	88.3
turbine power output	$P_t = \Delta h_{4s} \dot{m}_4$		Mw	16.2/15.8	17.2
Total power output	$P_o = P_t + P_s$		Mw	21.9/25.55	17.2
Total Resource Utilization efficiency	$\eta = \frac{P_o}{(\dot{m} \cdot \Delta h_1)^{(3)}}$		%	0.47/0.55	0.37

(4) demonstrated performance/advanced design

(3) isentropic enthalpy drop 80 to 2 psia=35.9

(2) state point numbers refer to Fig. 11

(1) η_n =nozzle efficiency, η_e =machine efficiency, eg. (23)

Existing Rotary Separator (30 in. dia.)

TABLE IV

System Description Summary For Two Stage Rotary Separator/Turbine System

		1st Stage	2nd Stage
No. of separator turbine per stage		1	2
No. of nozzles per separator		9	9
Nozzle inlet diameter	in.	16.3	11.0
" throat "	in.	7.53	6.23
" exit cross section	in. x in.	8.5 x 8.9	8.5 x 8.9
" length	in.	50	50
Rotor diameter	in.	85	85
Rotor rim width	in.	12	12
Probe inlet cross-section	in. x in.	5.2 x 5.2	5.6 x 5.6
No. of probe inlets		2	2
Nozzle inlet pressure	psia	115	40
Nozzle exit pressure	psia	50	15
Separator housing pressure	psia	50	15
Probe discharge pressure	psia	60	90
Probe stagnation pressure	psia	145	100
Nozzle exit velocity	ft/s	380	300
Separator speed	rpm	820	656
Turbine speed	rpm	410	328
Nozzle inlet flowrate	lbm/s	3333	3074
Nozzle inlet quality	%	0.025	1.3
Nozzle exit quality	%	0.082	13.
Liquid turbine output power	Mw _e	5.4	3.0

TABLE V

Cost Estimates For Two Stage Rotary Separator/Steam Turbine System

	Cost			
	Single Rotary Separator/Turbine		Two stage Rotary Separator/Turbine & Two Stage Steam Turbine	
	No req'd.	Cost, K\$	No req'd.	Cost K\$
A Rotary Separator System				
1. Separator housing			3	
(a) high pressure	-		1	200
(b) low pressure	1	120	2	240
2. Rotor	1	90	3	270
3. Nozzles	9	20	7	54
4. Liquid Turbine	1	77	3	231
5. Rotor/turbine bearing assembly	1	50	3	150
6. Control & Instrumentation Sys.	1	150	1	270
7. Electrical generators				
(a) 5 mwe	1	300		
(b) 3 mwe			1	180
		<u>807</u>		<u>1,595</u>
B Steam Turbine System (Ref. 11)				
1. 60 mwe turbine/generator			1	10,690
2. Condenser			1	1,070
3. Cooling towers			1	1,690
4. Auxiliary systems			1	430
				<u>13,880</u>
C Installation & Misc. site costs				<u>23,320</u>
				<u>37,200</u>

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APPENDIX A

ROTARY SEPARATOR

LABORATORY TESTS

TEST SYSTEM DESCRIPTION

The rotary separator system to be described was designed and built specifically for geothermal well tests. Preliminary tests to be described included laboratory calibration tests with steam and water to measure component performances on the nozzle, rotary separator, and liquid pick up. The primary performance parameters to be measured were (1) nozzle conversion of wellhead enthalpy to nozzle exit two-phase kinetic energy, (2) rotary separator conversion of the nozzle exit flow to single-phase liquid and steam flows, with each separate stream retaining the maximum kinetic energy content, (3) the rotating liquid film kinetic energy transfer.

The design of the experimental rotary separator was based on a wellhead pressure and temperature of $P_o = 0.55$ MPa (80 psia) and $T_o = 155^\circ\text{C}$ (312°F) representing available well conditions. The nozzle exit velocity corresponding to the expansion to atmospheric pressure is 165 m/s for an 0.55 MPa, 5% quality mixture at the nozzle inlet. The selection of separator rotor diameter of $D_s = 0.76$ M (30 in.), Fig. 1, was based on a desired rotor speed of approximately 5,000 rpm which in turn permits the use of aluminum as the rotor material, considering maximum rim stresses of 138 MPa (20,000 psia). The rotor speed, N_s , is related to the nozzle exit velocity, V_n , and the separator flow velocity, V_s , by the relation $N_s = V_s 60 / (2\pi R_s) = V_n 60 / (\cos\theta \cos\alpha / (\sin R_s))$ where the $V_n \cos\theta$ term is the component of nozzle exit velocity in the plane of separator rotation. The angle θ is the angle of inclination of the nozzle to the separator plane of rotation. The angle α is then the angle which the velocity component $V_n \cos\theta$ intersects the rotor rim. Fig. 1 shows these relationships where the $\cos\alpha$ term is equal to the ratio of nozzle distance from rotor center, R_n , to the rotor radius $D_s/2$. The square of the ratio V_s/V_n represents the maximum kinetic energy transfer from the nozzle. The energy transfer to the separator is thus proportional to $(120 V_n \cos\theta R_n / D_s^2)^2$.

Since $R_n = R_s - \frac{D_n}{2}$ where D_n is the nozzle exit diameter, then the energy transfer varies inversely to the square of the nozzle diameter. The maximum efficiency configuration is thus a full admission, minimum width nozzle distributed on the rotor circumference with nozzle size, D_n , determined by the nozzle flowrate m_n required for the desired output power, P_o . $P_o = m_n V_s R_s N_s K$ where the efficiency of the output probe or turbine is represented by K which may be a function of flowrate and velocity as in the stationary liquid pick up probed used. In this case, the output power would be measured in the form of hydraulic power, P_h , evaluated as the product of brine flowrate collected times the product of discharge pressure less the pressure required for reinjection. The predicted output power would be calculated as the fraction of the input kinetic energy to the probe that would be converted by the diffuser to hydraulic power output. Fig. 1 shows the diffuser in the liquid pickup inlet to have a convergent/divergent contour. This probed configuration represents a supersonic diffuser contour required for the two-phase flow at the probe inlet. A Mach number of 3.5 results if the liquid film occupies one-half the probe flow area. For this condition of two-phase flow entering the diffuser, the diffuser efficiency is given by $\eta_d = 0.86 - 0.09r_a$ which was a fit to experimental data for water and nitrogen (Ref. 5), where r_a is the ratio steam to liquid flow areas at the probe inlet. The predicted output power would be $P = (m_n/2)(1 - X_n)V_s^2\eta_d$.

Nozzle. The design of the two-phase nozzle was based on a calculation of two-phase flow (Ref. 6) which solves the nozzle flow equations, including the effects of liquid droplet breakup and vapor-liquid slip. The calculation results include the nozzle exit velocity and flowrate with the corresponding nozzle contour and throat diameter. For the well conditions chosen of 0.55 MPa (80 psia) and 5% inlet quality, a 63.5 cm long nozzle with 2.79 cm diameter throat and 5.1 cm diameter exit is required. The corresponding steam and liquid flowrates are 0.11 and 2.07 kgm/s at the inlet and 0.42 and 1.76 kgm/s at the exit. The nozzle flowrate characteristics are shown in Fig. A-2 as a function of nozzle inlet pressure and quality. A pair of nozzles were fabricated with the specified contour with 11 deg. convergence angle and 3.4 deg. divergence angle by machining this contour on a metal mandrel that was used as a form to cast a high-temperature epoxy fiberglass-

lined nozzle wall inside a 63.5 cm length of pipe. Flanges welded at each end of the pipe provide the attachment point for the mixer on the inlet and the separator at the exit. The mixer used is a commercially available chemical process mixer from the Koch Co. that utilizes the interwoven metal elements to pre-mix the steam and liquid at the nozzle inlet.

Rotary Separator. For a nozzle with 5.08 cm (2 in.) diameter exit and an inclination angle $\theta = 17.5$ degrees, Fig. 1, the flow impingement pattern on the inside of the cylindrical separator surface is 11.4 cm (4.5 in.) wide. To simplify the fabrication of the separator, it was made as an assembly of four parts: rotor rim, rotor stiffener and shaft assembly. The rotor is a 6.35 mm (0.25 in.) thick stainless steel plate 0.762 m (30 in.) in diameter. Attached by 32 screws at the edge of the rotor is the rim which forms the cylindrical separator surface. It has an outside diameter of 0.813 m (32 in.) and is 14 cm (5.52 in.) wide and was machined from an aluminum type-6061 plate. The rim was sandblasted, etched, and coated with a 0.005 mm (0.0002 in.) electro-less nickel plating. The rotor stiffener is a stainless steel conically shaped spinning that attaches at the rim and rotor connection line and extends radially inward to the bearing shaft assembly to form the triangular stiffened cross-section. The cantilevered bearing shaft assembly has one single-row bearing and a second double-row bearing spaced 8.9 cm (3.5 in.) apart in the bearing box. At the design condition of 4,000 rpm, the rotor rim was expected to increase in diameter by 2 mm (0.08 in.) due to thermal expansion of the rotor and rotational stresses in the rim.

The liquid pickup probe assembly consists of three tube sections. The inlet section was formed from 12 mm (0.5 in.) diameter stainless steel tubing that was formed into a rectangular cross-section by collapsing the tube onto an aluminum mandrel. The mandrel had been machined to the contour for a two-phase flow diffuser: convergent at 5 deg. half angle to a throat height of 1.02 mm (0.04 in.) which is one-half the height of 2.04 (0.08 in.) of the probe inlet. Downstream of the throat, the probe diverges at a 4 degree half angle up to the circular tube cross-section. At this point, the tube is curved with 4.44 cm (1.75 in.) radius of curvature through 135 degrees.

An adapter fitting is attached at this point to provide a gradual transition up to a 19 mm (0.75 in.) diameter tube which has a 90 degree bend and 5.1 cm (2 in.) bend radius with 20.3 cm (8 in.) straight that extends through the separator side plate seal fitting. The probe can either be fixed rigidly for flow collection tests or mounted free to rotate for measurement of static probe torque with the addition of lever arm and strain gauge type load cell.

To provide improved durability of the liquid probe inlet section, a boronizing process was applied to the probe tip. This process produces a diffusion layer 0.025 mm (0.01 in.) deep by the thermo/chemical application of boron. This had the effect of increasing surface hardness to 3,000 Vickers from 750 for Type 304 stainless.

Test Facility. In order to provide the capability of system tests at the Biphasic Engines test facility as well as geothermal field wells, the test equipment was installed in a semi-trailer van to provide a mobile test capability. To make the system as self-contained as possible, an instrumentation and control system was provided. In order to measure inputs to the system, a separate 5.08 cm (2 in.) pipe line with orifice metering section and throttle valve and line valve was installed for both a steam input and separate liquid brine lines. These two lines are connected at the inlet to the mixer. The brine flow picked up by the liquid flow probe is metered by a third orifice metering section and piped back to the point on the trailer where the steam and brine inputs are attached. The pressure at 14 points is measured with bourdon tube pressure gauges installed on the control panel with three diaphragm gauges used to measure orifice flowmeter differential pressures. Twelve iron-constantan thermocouples are installed with control console readout on a digital temperature indicator. An optical sensor and light-interrupting device are installed on the rotary separator shaft with a frequency display meter installed on the control console that serves to measure separator rotational speed.

To monitor rotor dynamics, three proximity probes were installed. Two of the probes were mounted 90 degrees apart on stationary holders to measure the distance to the rotor circumference. The third probe was installed to

measure the distance to the rotor disc in the direction parallel to the rotor shaft centerline. A readout instrument at the control console combines the signals from the two circumferential sensors to output a measure of the radial movement of the rotor. A third probe measures the translation or bending of the rotor parallel to the rotor shaft. Displacements up to 1.27 mm (0.05 in.) can be measured with a sensitivity of 2% maximum.

The mobile test system schematic is shown in Fig. A-3.

The test system can interface to either a steam boiler with steam and liquid flow capability or to a geothermal well with separate steam and liquid flow capabilities. For either type flow system that does not have separate flows but an unknown quality two-phase flow, the system calibration curves of the type shown in Figs. A-1 and B-2 can be used to assess the conditions at the wellhead. The measurement of nozzle inlet and exit pressure and separator speed thus determines the nozzle inlet quality from Fig. B-2 and the nozzle inlet flowrate from Fig. A-1. Measurement of temperature and pressure at upstream locations such as the wellhead, permits calculation of quality and flowrates at these upstream locations within the accuracy permitted by calculation of pure steam and liquid properties by methods described in following sections.

Two additional diagnostic capabilities are included in the test system as indicated by the throttling calorimeter and stand pipe systems in Fig. A-2. The calorimeter permits the measurement of water content in the separator steam discharge as a means of further evaluating separator performance. This is of particular importance with respect to the capability of the rotary separator system with geothermal input to output steam of sufficient purity for subsequent condensation to meet agricultural or potable needs.

The standpipe and related liquid trap and metering system provide a means of separating collected brine flows from non-condensable gases also collected by the liquid pickup. The rate of level change in the standpipe with valve V5 closed provides a measure of the rate of non-condensibles

entering the standpipe with the valve V6 adjusted to maintain constant liquid flowrate and a pressure level to ensure a subcooled liquid.

ROTARY SEPARATOR SYSTEM TESTS

The performance evaluations of the rotary separator were conducted with steam and water over a range of system inlet pressures from 0.15 to 0.52 MPa (20 to 75 psia) and steam qualities of 0.03 to 0.15. The steam and hot water were provided by an 175 hp oilfired boiler at the Biphasic Engines test facility. The liquid flow from the separator output was returned to the feed water tank on the boiler to provide continuous system capability. Test tests were conducted in two parts in order to calibrate the nozzle separately from the separator. The nozzle tests are described in Appendix B.

The separator performance tests were conducted with the nozzle and separator assembly installed in the test van as shown in Fig. A-4. The steam and water enter the mixer at the top left and enter the separator on the center at the left causing the separator to rotate counterclockwise. The separator shaft extends through the bearing housing on the rear of the assembly to which is attached the rotor braking system and tachometer. The liquid pickup outlet tube is shown extending through the front separator housing at the top center from which point it enters the liquid standpipe shown on the left. The test conditions were varied over the range of nozzle inlet pressures to 0.069 to 0.52 MPa with several qualities at each pressure. At each test condition the system pressures, temperatures, separator speed, liquid standpipe level were recorded by photographing the control console display panel. On the majority of tests the liquid separator pickup performance was evaluated on the basis of recovered flow measurements which was supplemented on selected conditions by (1) measurement of liquid recovery pressures as controlled by operation of the flow-trotting valve, V6, or (2) measurement of the torque applied to the probe as a result of flow momentum loss upon entering the liquid pickup probe.

LABORATORY TEST RESULTS

The combined tests of the nozzle and rotary separator permitted measurement of the energy transfer of the nozzle liquid jet power to the separated liquid film rotating inside the separator. Evaluation of the nozzle exit jet power was based on the separate measurements of the nozzle exit velocity. The nozzle calibration test results are described in Appendix B. For the flow geometry described in Fig. 1, the maximum possible energy transfer is determined primarily by the velocity losses defined by the (1) inclination of the nozzle to the plane of separator rotation (angle θ), (2) impingement of the nozzle exit vector and the separator surface tangent (angle α), and (3) rotor windage and bearing torque losses, T_3 . The later combined loss is evaluated as the product of the square of the separator velocity, separator surface area, steam density and skin friction coefficient. The separator rotational speed is then estimated by equating the input flow momentum torque less the above losses to the momentum of the flow inside the separator or: $V_s = (m_t V \cos\theta \cos\alpha R_n - T_2) / (m_t R_s)$. The results of this calculation for three nozzle inlet qualities using the theoretical nozzle exit velocities of Fig. B-1 are shown in Fig. A-5 where separator rotational speed, rpm , is shown as a function of nozzle inlet pressure. The measured data represent the results of the nozzle separator test with the corresponding nozzle inlet qualities indicated by the numbers in the symbols. The transfer of nozzle liquid flow kinetic energy to separator liquid film energy was 70% at the design point of 0.55 MPa (80 psia) nozzle inlet pressure. This was calculated as the square of the ratio of the separator flow velocity to nozzle exit velocity which is proportional to the $\cos\theta \cos\alpha$ product as shown in Fig 1. The separator efficiency is thus proportional to the square of this term. For an angle change from 17.5 to 10 and 18 to 14 degrees for angles θ and α , respectively, the calculated efficiency is increased from 70% to 77.7%, which is a reasonable geometry for a full admission nozzle.

The flow separator evaluation was based on measurement of the liquid collection rate by the flow pickup probe to be discussed in the following section and by measurements of liquid content in the steam discharge. This steam

quality was evaluated by both a throttling calorimeter installed in the steam discharge and by the addition of a liquid drop-out tank on the exit of the steam line. Both measurements indicated steam qualities of greater than 98%. A calculator of the condensation rate on the uninsulated separator side wall and steam discharge piping indicated all the liquid content in the steam was from condensation.

Liquid Recovery. Several forms of stationary liquid flow recovery systems were tested to maximize separator and recovery performance. Three pickup designs were tested, the basic difference being (1) the cross-sectional shape of the tube at the liquid entry being circular or rectangular, (2) the tube bend radius shown as R_p in Fig. 1, and (3) the location and shape of the diffuser included in the tube. For either the round or rectangular inlet probes with a tight bend radius $R < 2.5$ cm, the liquid recovery was inadequate, apparently caused by instability of the interface between condensing steam and the pure liquid that follows. This condition was manifested in loud hydraulic noises and severe vibrations of the probe tip and flow spillage from the inlet. This condition was eliminated when the probe discharge flow was throttled, resulting in probe pressures in excess of 1.72 MPa (250 psig). This is analogous to a choked flow supersonic diffuser with a free-standing normal shock off the tip of the probe. The steam condensation occurring in the detached shock wave eliminates the probe vibrations; however, the flow spillage precludes this as an effective recovery design. A successful probe design utilized a more gradual bend radius $R_p = 4.45$ cm (1.75 in.) with a supersonic diffuser contour machined into the probe between the start of the tube bend curvature and the inlet. The probe contour had a rectangular cross-sectional inlet with 2.0 mm height and 16.8 mm width (0.08 in. x 0.66 in.). The diffuser contour consisted of a convergent flow section to a full round cross-section of 10.7 mm (0.5 in.) diameter. This probe design resulted in significantly reduced flow noise and vibration, and liquid recovery rates which agreed with the predicted flow calculated as $(1 - X_6) m_t$ where X_6 is the nozzle exit quality and m_t is the nozzle total flowrate. The ratio of measured liquid pickup flowrate to nozzle total flowrate is shown in Fig. A-7. The measured liquid recovery

fractions shown were within $\pm 2\%$ of the calculated values. A series of tests were conducted at low nozzle inlet pressures up to 0.21 MPa (30 psia) to compare probe discharge recovery pressure with the pressure P_g calculated from:

$$p_g = 0.5 r_a (\rho_g + \rho_l) (v_7^2 - v_8^2) / (1 + r_a)$$

where $\rho_g + \rho_l$ are steam densities and r_a is the ratio of steam to liquid flow areas at the probe inlet and v_7 and v_8 are the separator and probe liquid discharge velocities. These data are compared to the predicted values in Fig. A-8. At the design condition, the maximum probe discharge pressure is 7.58 MPa (1100 psia). For the probe flowrate of 1.84 kg/s (4.1 lbm/s) this represents a hydraulic outpower $p_h = 13$ kW evaluated as:

$$p_h = (1 - S_6) m_t (p_8 - 1.1 p_5) / \rho_l$$

which allows for a reinjection pumping pressure of 110% of nozzle inlet pressure. This output represents a 32% conversion of wellhead available energy for atmospheric steam discharge.

FIGURE A-1 Comparison of Measured and Calculated Nozzle Flowrate for Steam-Water Mixtures

FIGURE A-2. Trailer Mounted Separator System

FIGURE A-3
Geothermal Rotary Separator Test Van Installation
at Biphase Engines, Inc. for Clean Steam/Water Tests
A-12

STEAM

LIQUID

SEPARATOR HOUSING

STEAM OUT

LIQUID STANDPIPE

FIGURE A-5. Comparison of Measured and Calculated Separator Speed

PRESSURE, psia

FIGURE A-6. Comparison of Measured and Predicted Liquid Recovery Fraction

FIGURE A-7. Discharge Pressure from Static Liquid Pickup Probe

FIGURE A-8. Rotary Separator Windage and Bearing Losses

APPENDIX B

TWO-PHASE NOZZLE

DESCRIPTION

The operation of the Biphase Engines, Inc. rotary separator is based on the centrifugal separation of vapor and liquid phases from a geothermal or other source of high enthalpy two-phase flow. A prime requisite for efficient operation is that the energy to drive the separator be derived from the well-flow enthalpy. This implies an energy conversion of flow enthalpy to kinetic energy. This is conventionally done with a De Laval type nozzle in steam flow applications. In the application of geothermal well flow which typically exists as a two-phase flow of liquid and small amounts of steam (steam quantities of 0 to 20%) a specially designed nozzle must be used. Since a two-phase nozzle operates by expansion of the flow from high to low pressure which results in the acceleration of the steam phase from low to high velocity. The liquid is accelerated by hydrodynamic drag of the steam or the liquid due to the relative velocity between the steam and liquid. The design of an efficient nozzle therefore requires the analysis of a complicated flow problem which requires consideration of interphase mass and momentum transfer with droplet breakup and frictional losses. An analysis of this type, Ref. 1, was used to select a nozzle contour optimized for maximum energy transfer expanding from 0.55 to 0.1 MPa (80 to 14.8 psia) with 5% inlet steam quality. The analysis provides a prediction of flow conditions at specified positions along the nozzle. Table I contains the output summaries for the inlet throat and exit stations for the 0.69 MPa (80 psi) inlet case. These calculation output results were used to prepare the performance characteristics shown in Figs. B-1 and B-2 for nozzle flowrate and nozzle exit velocity as a function of inlet pressure and inlet steam quality. Figs. B-3 and B-4 show the corresponding nozzle exit to throat area ratio and nozzle throat area required respectively for the three steam inlet qualities, 0.01, 0.05, 0.10.

Two nozzles were fabricated to the required size for the 0.69 MPa/0.05 steam quality inlet conditions. Table B- II contains dimensions from which a metal mandrel was machined. The mandrel was used as a form placed inside a 0.5 m

long leg of 6.3 cm diameter pipe. The annular space between the mandrel and pipe wall was filled with an epoxy, after having first wrapped the mandrel with three layers of fiberglass cloth. A vacuum was applied to the epoxy volume while it was cured at 150°C for 24 hours. The completed nozzle was 63.5 cm long with inlet, throat and exit diameters of 5.65, 2.88, 5.08 cm respectively. Prior to tests of the nozzle with the rotary separator, the nozzle was tested to permit the direct measurement of nozzle exit thrust from which the nozzle exit velocity could be calculated.

Nozzle Calibration. The nozzle calibration consisted of operation over the range of flow conditions with the nozzle mounted to permit measurement of the thrust produced by the flow discharge into the atmosphere. The thrust was measured by mounting the nozzle on a parallelogram test configuration as shown in Fig. A-3, where the nozzle is free to move in the direction parallel to the flow. The motion is restrained by the attachment of a transducer which measures the nozzle thrust. The steam and water flows are metered and controlled by the test-van systems and connected to the nozzle mixer by means of flexible hoses attached at right angles to the thrust measurement plane. A dead-weight calibration system permits application of static forces parallel to the thrust line to verify load cell calibration. Application of the calibration weights during flow tests permit verification of no thrust errors from extraneous forces such as table hinge pin friction or misalignment of the thrust table or flexible hoses. The discharge from the nozzle was ducted into a free-standing barrel where the steam was vented to the atmosphere as shown. A separate pump returns the liquid collected to the boiler feed tank. The test measurements included liquid and steam flowrates, temperature and pressure upstream of the mixer and mixture temperature and pressure at the nozzle inlet.

TEST RESULTS

The nozzle and separator performance tests were conducted to measure component

efficiencies in order to permit an evaluation of system performance for the existing and optimized designs.

The nozzle calibration tests permitted calculation of the nozzle exit velocity by the relationship:

$V_6 = Tg/m_t$ where T = measured thrust, N , m_t = total flowrate (liquid plus steam to the nozzle) kgm/s , g = gravitational constant. The results of this calculation for the conditions tested are compared in Fig. B-2 with the theoretical nozzle exit velocity based on the procedure of Reference 6. For the design condition of 0.55 MPa (80 psia) nozzle inlet pressure, the predicted exit velocity is 198 m/s (650 ft/s) at 5% nozzle inlet quality. For the data of Fig. B-2, the variation of measured velocity is $\pm 4\%$ about a mean represented by the predicted values shown as the solid lines. The close agreement was obtained for an assumed droplet size of 1270 micron liquid droplet diameter at the nozzle inlet. For this condition, the fraction of nozzle inlet enthalpy converted to nozzle exit jet power kinetic energy is 0.67. This is calculated as the ratio of the exit kinetic energy $\frac{1}{2}m_t V_6^2$, to the isentropic enthalpy drop corresponding to expansion from nozzle inlet to atmospheric exit conditions. For the low inlet pressure and large initial droplet size, the low efficiency is considered a result of poor flow uniformity. The importance of liquid atomization is indicated by the calculated nozzle efficiency increase from 67% to 88% as a result of droplet diameter reduction from 1270 to 50 microns. Operation at elevated pressure levels also shows a predicted efficiency increase from 67% to 78% at constant drop size for a nozzle inlet pressure increase from 0.55 to 1.55 MPa. For low temperature reservoirs, the most likely means of achieving high nozzle efficiency is with maximum droplet atomization which will require an optimization of wellhead pressure drop distribution between the nozzle and preatomizer placed at the nozzle inlet. The variation of calculated nozzle efficiency as a function of inlet pressure and quality is shown in Fig. B-5 for atmospheric and 0.1 MPa (2 psia) exit pressures. The efficiency is shown to increase from 70% to 85% for a nozzle inlet pressure increase from 0.55 to 2.0 MPa. This compares closely with measurements with steam and water for high inlet pressures,

Ref. 4.

The enthalpy drop was calculated as $H_5 - H_{6i}$ where:

$$H_{6i} = X_6 H_{g6} + (1 - X_6) H_{f6} \text{ and}$$

$$H_5 = X_5 H_{g5} + (1 - X_5) H_{f5}.$$

The subscripts 5 and 6 refer to nozzle inlet and exit conditions and subscripts g and f refer to pure steam and liquid properties (enthalpies).

The nozzle exit quality calculated from the relation for constant entropy.

$S_5 = S_6$ which yields:

$$X_6 = X_5 S_{g5} + (1 - X_5) S_{f5} / (S_{g6} - S_{f6}) = 0.21$$

For this condition, $((1 - X_5) \times 100) = 79\%$ of the nozzle jet power is contained in the liquid flow component to be transferred to the separator and liquid pickup system for conversion to hydraulic output power with a stationary flow probe. The utilization of a radial inflow gas turbine would make the remaining $21\% = (X_6 \times 100)$ of the nozzle jet power available for shaft output power conversion. The addition of a condenser for 0.014 MPa (2 psia) subatmospheric steam-condensing pressure would provide additional energy for ideal steam turbine conversion in the amount of 18% of the nozzle jet power.

FIGURE B-1. Nozzle Flowrate Variation with Inlet Pressure

FIGURE B-2. Comparison of Measured and Calculated Nozzle Exit Velocity for Steam Mixtures B-6

FIGURE B-3a. Steam-Water Nozzle Exit to Throat Area Ratio

FIGURE B-3b. Steam-Water Nozzle Exit Velocity

FIGURE B-4. Steam-Water Nozzle Throat Area

FIGURE B-5. Steam-Water Nozzle Efficiency
B-10

TABLE B-I
COMPUTER PERFORMANCE PREDUCTIONS FOR STEAM-WATER NOZZLE

TIME = 101521 MINUTES

REAL-TIME PHASE I ONE-COMPONENT NOZZLE FLOW

APRIL 23, 1973

CASE NUMBER 5

50 PSIA, 0.05 QUALITY

NS 10	VS 31	MSJ 1	MDP 0	MGED 0	VDS 30	VSD 30	NA 30	MXG 30
DO -2.5000-01	PFI 0.0000	RAXD 0.0000	EAT 1.0000+02	ALAN 0.0000	DO 5.0000-02	PO 8.0000+01	VGO 5.0000+01	VLO 5.0000+01
TADJ 0.0000-03	TADJ 5.0000-03	XGJ 5.0000-02	EDS 1.0000-03	ESJ 1.0000-03	EDAS 1.0000-03			

X(2) TABLE FOLLOWS

TABLE 10 - 25 INCH NOZZLE

X(2) TABLE INPUT

PRESS Y	8.0000+01 0.0000	7.9877+01 7.9071+01	7.9577+01 7.8517+01	7.9084+01 7.8221+01	7.8029+01 7.6224+00	7.6297+01 7.4135+00	7.3499+01 7.0843+00	6.9319+01 6.6149+00
PRESS X	6.4757+01 6.6457+00	5.9900+01 7.4764+00	5.7745+01 5.3771+00	5.3415+01 9.1378+00	4.9997+01 9.9645+00	4.6741+01 1.0799+01	4.3760+01 1.1630+01	4.0923+01 1.2461+01
PRESS X	3.8251+01 1.3291+01	3.5757+01 1.2122+01	3.3431+01 1.0757+01	3.1245+01 1.5783+01	2.9201+01 1.6614+01	2.7293+01 1.7445+01	2.5513+01 1.8276+01	2.3854+01 1.9106+01
PRESS X	2.2317+01 1.9937+01	2.0571+01 2.7759+01	1.8535+01 2.1595+01	1.8294+01 2.2429+01	1.7142+01 2.3260+01	1.6075+01 2.4091+01	1.5087+01 2.4921+01	1.4172+01 2.5752+01
PRESS X	1.3325+01 2.6583+01	1.2544+01 2.7413+01	1.1322+01 2.8244+01	1.1156+01 2.9075+01	1.0541+01 2.9905+01	9.9732+00 3.0736+01	9.4499+00 3.1567+01	8.9671+00 3.2396+01
PRESS X	8.5215+00 3.3227+01	8.1105+00 3.4057+01	7.7433+00 3.4447+01	7.3744+00 3.5720+01	7.0533+00 3.6551+01	6.7513+00 3.7382+01	6.4705+00 3.8213+01	6.2089+00 3.9043+01
PRESS X	5.4547+00 3.9937+01	5.7352+00 4.0705+01	5.5175+00 4.1533+01	5.3150+00 4.2360+01	5.1230+00 4.3197+01	4.9391+00 4.4028+01	4.7633+00 4.4858+01	4.5944+00 4.5689+01
PRESS X	4.4314+00 4.6527+01	4.2734+00 4.7350+01	4.1175+00 4.8101+01	3.9693+00 4.9012+01	3.8219+00 4.9843+01	3.6767+00 5.0673+01	3.5333+00 5.1504+01	3.3914+00 5.2335+01

TABLE B-I (CONT'D)

APRIL 29, 1976

A
80 PSIA, 0.05 QUALITY

X	P	XG	VB	A	T	VG	VL
2.3002+01	1.7500+01	1.3009+01	5.0995+02	4.9393+01	6.7945+02	9.1001+02	5.6257+02
VS	D	RV	RA	AG	WL	ROG	ROL
3.4744+02	3.4321+03	2.1373+02	1.3212+02	1.3609+01	8.6391+01	4.3929+02	5.9600+01
LM	CLM	VIGM	VILM	RJGM	ROLM	HG	REM
9.5438+02	1.0079+00	2.9433+02	5.5065+01	4.4224+02	5.9590+01	1.1536+03	5.3549+02
STGMA	COM	XL	S	V3D			
5.8759+01	5.4350+01	2.4454+02	5.6972+01	5.9590+02			
XGB	AB	KVB	KGB	M-B	Q	REF	
1.3638+01	7.3650+01	2.1425+02	1.3638+01	3.5362+01	2.1526+02	1.3303+05	
YJ	COM	THD	DELJ	DELJS	REDOM	CFOM	TWOM
5.9651+00	5.7553+00	4.5355+02	4.5653+01	5.9310+02	7.1556+06	1.2313+03	2.6667+02
X	P	XG	VB	A	T	VG	VL
2.5000+01	1.5000+01	1.4145+01	5.5195+02	5.5519+01	6.7270+02	9.7092+02	5.9940+02
VS	D	RV	RA	AG	WL	ROG	ROL
3.7153+02	3.4321+03	2.5912+02	1.5975+02	1.4145+01	8.5855+01	3.4022+02	5.9800+01
LM	CLM	VIGM	VILM	RJGM	ROLM	HG	REM
7.6957+02	1.0067+00	2.9092+02	5.9071+01	3.5318+02	5.9790+01	1.1507+03	5.0202+02
STGMA	COM	XL	S	V3D			
5.9441+01	5.5388+01	2.5933+02	5.5997+01	5.3642+02			
XGB	AB	KVB	KGB	M-B	Q	REF	
1.4179+01	3.2637+01	2.5485+02	1.4179+01	5.5821+01	2.6085+02	1.1738+05	
YJ	COM	THD	DELJ	DELJS	REDOM	CFOM	TWOM
4.2038+00	5.7941+00	5.0073+02	5.1505+01	5.1379+02	7.9490+06	1.2116+03	2.4731+02

TABLE B-I (CONT'D)

APRIL 29, 1976

90 PSIA, 0.05 QUALITY

2.7454+01	1.2500+01	1.4909+01	7.0020+02	5.4970+01	5.5290+02	1.0360+03	6.4136+02
5.7459+02	3.4321+03	3.2815+02	2.0314+02	1.4908+01	5.5092+01	3.2049+02	6.0030+01
9.7531+02	1.0654+00	2.8750+02	7.3414+01	3.2350+02	6.0018+01	1.1472+03	4.5529+02
3.1344	3.04	VL	S	V30			
5.0123+01	5.7051+01	9.7573+02	5.5569+01	5.5277+02			
1.4948+01	2.5214+01	3.2920+02	1.4948+01	3.5052+01	3.3020+02	1.0148+03	
5.5475+03	7.8311+00	5.5577+02	5.8195+01	DEL7S	RED7M	CFDM	TWDM
				7.2741+02	9.0647+06	1.1875+03	2.2011+02
5.7537+01	1.0000+01	1.5657+01	7.5741+02	7.9028+01	5.5290+02	1.1142+03	6.9117+02
4.2337+02	3.4321+03	2.2990+02	2.5667+02	1.5657+01	5.4343+01	2.5030+02	6.0280+01
7.3173+02	1.0001+00	2.9135+02	7.7737+01	2.5331+02	6.0268+01	1.1433+03	4.0511+02
3.1344	3.04	VL	S	V30			
5.0943+01	5.9173+01	9.8210+02	5.5953+01	7.3763+02			
1.5735+01	1.1473+02	2.3147+02	1.5705+01	5.4295+01	4.3247+02	8.6878+04	
2.9537+03	7.4858+00	5.5070+02	5.6730+01	DEL0S	RED0M	CFDM	TWDM
				5.3660+02	1.0637+07	1.1591+03	1.9128+02

TABLE B-1 (CONT'D)

APRIL 29, 1976

9
90 PSIA, 0.05 QUALITY

2.7525+01	2.5000+00	1.7523+01	1.0774+03	2.5200+02	5.9340+07	1.5784+03	9.5538+02
VS	D	RV	RA	RG	RL	RNG	RDL
2.2324+02	1.2321+03	2.1123+03	1.2773+03	1.7533+01	8.9417+01	7.0950+03	6.1488+01
1.0173+03	2.7296+01	2.4927+02	1.1632+00	7.4262+03	6.1457+01	1.1190+03	1.8590+02
3124	274	XL	S	V30			
5.3130+01	7.2921+01	1.7192+03	5.7822+01	1.3422+03			
X33	AB	AVB	438	1.3	Q	REF	
1.7732+01	3.7133+02	2.1254+03	1.7702+01	3.7228+01	2.1274+03	3.0222+04	
YJ	434	140	DELO	DEL33	RED7M	CFDM	THDM
2.2552+00	1.2342+01	1.4615+01	1.5033+00	1.7721+01	2.2035+07	1.0411+03	7.1900+03

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VS	D	XG	V3	A	T	VG	VL
5.6432+00	5.5000+01	5.5215+02	1.7023+02	2.1325+01	7.5722+02	3.1473+02	1.6015+02
VS	D	RV	RA	MG	UL	RDG	RDL
1.5457+02	4.4005+03	2.5642+01	1.3557+01	5.5215+00	9.3478+01	1.5024+01	5.7375+01
V	SLM	VISM	VILM	RJG4	RJLM	MG	REM
4.1151+02	1.3239+00	3.3373+02	4.4115+01	1.5052+01	5.7371+01	1.1791+03	9.2233+02
SLG4	SL	XL	S	V3D			
4.9772+01	3.0000	9.1170+02	9.0801+01	1.5350+02			
VJ	ADM	TAD	DELJ	DELJ3	REDDM	CFDM	TDDM
2.5054+00	-1.7457+00	1.3257+02	1.3645+01	1.7057+02	9.4607+05	1.7509+03	2.6842+02

TABLE B-II

NOZZLE DESIGN CONTOUR FOR 100 psia, 0.05 QUALITY, $\dot{m}_t = 5.66$ pps

X	A	d	X	A	d
in.	in. ²	in.	in.	in. ²	in.
0	68.661	2.224	10.934	21.593	1.247
3.0135	26.01	1.369	11.473	22.303	1.267
3.831	21.63	1.248	12.036	23.017	1.288
4.365	20.37	1.213	12.635	23.859	1.311
4.770	18.94	1.168	13.266	24.792	1.336
5.128	18.465	1.153	13.932	25.801	1.363
5.478	18.091	1.142	14.640	27.193	1.400
5.823	17.884	1.135	15.396	28.694	1.468
5.995	17.814	1.133	16.208	30.362	1.478
6.177	17.865	1.134	17.086	32.205	1.523
6.543	17.894	1.135	18.039	34.277	1.571
6.9177	18.091	1.141	19.102	36.767	1.627
7.306	18.214	1.145	20.273	40.184	1.701
7.705	18.488	1.154	21.612	44.292	1.786
8.116	18.689	1.160	23.157	49.229	1.883
8.545	19.038	1.171	25.000	55.537	2.000
8.982	19.305	1.179			
9.445	19.723	1.192			
9.919	20.317	1.210			
10.414	20.891	1.227			

APPENDIX C

PERFORMANCE PREDICTIONS

CALCULATION DESCRIPTION

In order to extend the predictions of operating characteristics for a rotary separator over a range of sizes and wellhead conditions, an analytical performance model of the system was computer programmed. This permitted the parametric analyses of the rotary system performance as a function of the independent variables listed in Table C-I. Assignment of values to these eight variables allows the calculation of 48 performance variables listed in Table C-II.

The calculation procedure evaluates the thermodynamic properties of enthalpy and entropy at the inlet and nozzle exit by a table look-up for the nozzle inlet pressure and quality. The steam table values are modified according to the procedure on page C - 4, for those cases where a nonzero value of brine concentration was specified. The nozzle throat and exit areas are evaluated from the relationships given on Figs. B-3 and B-4 for one nozzle. The size of the annular nozzle was calculated based on the rotor size, nozzle angle and nozzle exit height. The total number of nozzles and total flowrate is calculated for a full admission nozzle. The separator speed was evaluated on the basis of equating the momentum of the liquid flow on the separator rim to the nozzle exit momentum parallel to the rim velocity less the momentum losses due to rotor windage, bearing losses, and probe losses (step 1780 of program listing).

The output pressure and hydraulic power from a stationary liquid pickup were calculated by the two-phase diffuser subroutine (steps 4000 to 5000). It calculates the stagnation pressure for the flow at the probe inlet conditions less the reinjection well pressure requirement assumed to be 110% of the nozzle inlet pressure. The diffuser pressure recovery is calculated with a diffuser efficiency which is in turn a function of the gas to liquid volume ratio at the probe inlet, step 1850. The rotary pitot subroutine (steps 6000 to 6700) performs the separator calculation with the stationary liquid pickup replaced by a rotating pickup that outputs

pressurized brine for reinjection and output shaft power. The rotational speed of the rotary pickup is one of the calculation results. It is based on first calculating the relative velocity of the flow with respect to the probe inlet, V_r Fig. C - 1. It is this velocity head that must overcome (1) the centrifugal force as the liquid flows radially inward in the liquid pickup and (2) the pressure head required for reinjection. The output torque from the probe was evaluated as the difference of the separator flow momentum torque at the probe inlet and the torque losses due to bearing, seal, external probe form drag and flow loss due to flow interception with probe wall.

The separator performance with rotary pitot turbine was also calculated with a separator design improvement consisting of several concentric separator rims (steps 6800 to 2995). This design reduces the momentum losses by reducing the effective angle at which the flow impinges on the separator rim. For this case, an additional power output was also calculated based on the addition of steam turbine reaction blading in the separator steam discharge. The additional power comes from converting the kinetic energy of the separated steam to torque which is added to the liquid turbine output. This steam turbine was assumed to have a 70% kinetic energy to torque conversion efficiency.

The power output and efficiency was calculated also for an alternate liquid turbine design referred and shown in Fig. C-1 as a U-tube turbine. This turbine design has a higher power output compared to the rotary pitot. The U-tube does not have a diffuser and related diffuser losses since there is no pressure recovery required to meet centrifugal or reinjection head requirements. The output torque comes from the reaction to flow reversal inside the U-tube. By rotation of the U-tube at a velocity, $V/2$, being one-half the liquid film velocity, V , at the tube inlet, the flow leaves the tube at zero absolute velocity. Thus the entire inlet flow momentum must be transferred to the probe and finally as output torque less the bearing, windage and frictional losses.

CALCULATION RESULTS

The calculation results are summarized in Figs. C-2 through C-6. The variation of total flow capacity as a function of wellhead pressure at inlet qualities of 0.01, 0.05 and 0.10 are shown for nozzle exit heights which are $K = 0.1$ & 0.2 , of the rotor diameter. The nozzle exits are square in cross-section for both $K = 0.1$ and 0.2 with $20 - 7.6 \times 7.6$ cm (3×3 in) and $9 - 15.2 \times 15.2$ cm (6×6 in) nozzles respectively. This results in approximately twice the flowrate for the $K = 0.2$ case as shown in Fig. C-2. Curves 2, 4 and 6 thus represent the $K = 0.2$ case for inlet qualities 0.01, 0.05, 0.10 and curves 1, 3 and 5 are the corresponding $K = 0.1$ flowrate characteristics. There are two sets of these six curves: the upper set for an atmospheric nozzle exit pressure. The lower set of six curves for a 0.014 MPa (2 psia) nozzle exit pressure shows a considerably reduced flow capacity corresponding to the larger nozzle exit areas required for lower exit pressures. This has the effect of reducing the output power also since the flowrate decreases more rapidly than the nozzle jet power per unit flowrate increases. The variation of output for the same cases is shown in Fig. C-3 for a rotary separator with 0.1 (2 psi) separator pressure. As an example, for an inlet pressure of 0.55 MPa (80 psia) and inlet quality 0.5, the flowrate is 269 kgm/s (592 lbm/s) and the output power is 2.4 mWe for $K = 0.2$, curve 4, for the atmospheric exit pressure nozzle. For the subatmospheric case and the same rotor size, $D_1 = 0.76$ m (30 in), the flowrate is reduced to 60 kgm/s (132 lbm/s) a factor of 4.5 whereas the power is reduced to 1.05 mWe, curve 4, Fig. C-4, a factor of 2.3. Although there are large power (P_o) and flowrate ($\dot{m}$) differences, the ratio of $P_o/\dot{m}\Delta h_f$ are nearly the same for the two cases. The efficiency for conversion of isentropic enthalpy change to liquid turbine output power is shown in Fig. C-5 for the atmospheric nozzle exit to be 0.287 and from Fig. C-4 for the subatmospheric nozzle exit the efficiency is 0.268. Fig. C-5 shows also the increase in efficiency by three hardware changes to the configuration represented by curve 4 as an example. The addition of the radial inflow steam reaction blading increase the power output and efficiency as shown by the increase from curve 4 to 4S. The addition of the multi-rim separator with four

concentric separator surfaces reduces the impingement losses by reducing the impingement angle from 29.9° to four impingements at 18.2° . The ratio of the square of cosine of these angles is the factor by which the efficiency increased with the four rim separator, i.e. from Fig. curve 4S to 4M or 0.35 to 0.417. The final system modification is the addition of a steam turbine which converts 70% of the enthalpy change for steam expansion from atmospheric to 0.014 MPa (2 psia) condensing pressure to output power. The maximum efficiency with all three modifications is seen to be 0.65.

BRINE PROPERTIES

The high concentration of total dissolved solids, TDS, typical of most geothermal resources requires some means of adjusting the thermodynamic properties for water to be more representative of brine properties. A subroutine was added to the program which modifies the values of enthalpy, entropy, density and saturation pressure for contained? in the program for both liquid and vapor phases of water. Based on the input value of TDS concentration the pure water properties are modified by a calculation procedure modified after the procedure of Ref. 2. All the brine properties are calculated from a one step procedure as a function of brine concentration, temperature and pressure. The calculation of brine-liquid entropy however requires a stepwise calculation of the entropy at 2°C steps from 20°C to the temperature desired. Steps 8210 to 8270 contain the one step property calculations and steps 9010 to 9800 is the procedure required for the stepwise calculation of liquid entropy.

In order to estimate the thermodynamic properties for the brine and steam for the field test data without having to execute the calculation procedure described above for each test point, the series of curves shown in Figs. C-6, C-7 and C-8 were prepared. Fig. C-6 contains the variation of enthalpy change for the isentropic expansion for pure steam and water from the nozzle inlet pressure to either the atmospheric or subatmospheric nozzle exit pressure 0.014 MPa (2 psia) for three values of nozzle inlet quality, 0.01, 0.05, and 0.10. Shown at the top of Fig C-6 are the corresponding six curves of nozzle exit quality. Figs. C-7 and C-8 contain the factors which when

multiplied by the pure fluid enthalpy change values from Fig. C-6 give the isentropic enthalpy change for the brine of concentration -C.

ROTARY PITOT LIQUID TURBINE

FIGURE C-1. U-TUBE LIQUID TURBINE

P_2 = NOZZLE EXIT PRESSURE

D_1 = ROTOR DIAMETER = 0.76 m (30 in)

T_2 = NOZZLE INCLINATION ANGLE = 24°

FIGURE C-2. Steam-Water Nozzle Flowrate

FIGURE C-3. Output Power For a Rotary Separator With Atmospheric Discharge

FIGURE C-4. Output Power For a Rotary Separator With a Discharge Pressure of 0.1 MPa

FIGURE C-5. Variation of Rotary Separator Efficiency
C-10

FIGURE C-6. Enthalpy Variation for Clean Water-Steam Mixtures

FIGURE C-7. Nozzle Enthalpy and Exit Quality Variation With Total Dissolved Solids, C , For an Atmospheric Separator Pressure

FIGURE C-8. Nozzle Enthalpy and Exit Quality Variation With Total Dissolved Solids, C , for Subatmospheric Separator Pressure C-13

TABLE C-I
ROTARY SEPARATOR COMPUTER PERFORMANCE ANALYSIS PROGRAM

Independent Variables

Nozzle inlet pressure	P_1
Nozzle exit pressure	P_2
Nozzle inlet steam quality	X_1
Rotary separator diameter	D_1
Nozzle exit height	D_2
Nozzle inclination angle	T_2
Liquid pickup wall thickness	W_7
Brine total dissolved solids concentration	C

TABLE C-II

ROTARY SEPARATOR COMPUTER PERFORMANCE ANALYSIS PROGRAM TESTING

```

LYS
100 I 2-5-77 /T12/
200 VAR = ZERO
400 REAL A$(2,6),V(2,6),H(6,17)
500 STRING X5,X6
600 DEF FNA(A,B,C)= 10-(A*LOG10(B) + C)
650 DEF FND(A,C,B) = A + B*LOG(C)
660 DEF FNE(A,C,B) = A*C-B
700 READ H(J,K) FOR J = 1 TO 6 FOR K = 1 TO 17
705 DATA 94, 1116.2, .175, 1.92, 2, 126.1,
120.9, 1127.3, .2199, 1.8626, 4, 153,
150.8, 1139.3, .2676, 1.806, 8, 182.9,
180, 1150.4, .3121, 1.7568, 14.7, 212,
196.2, 1156.5, .3358, 1.732, 20, 228
710 DATA 236.1, 1169.8, .3921, 1.6765, 40, 267.2,
262.2, 1177.7, .4273, 1.644, 60, 292.7,
282.1, 1183.1, .4534, 1.6208, 80, 312.04,
298.5, 1187.2, .4743, 1.6027, 100, 327.8
720 DATA 312.6, 1190.4, .4919, 1.5879, 120, 341.3,
330.6, 1194.1, .5141, 1.5695, 150, 358.4,
366.2, 1199.9, .5564, 1.5354, 225, 391.9,
394.0, 1202.9, .5882, 1.5105, 300, 417.4,
424.2, 1204.6, .6217, 1.4847, 400, 444.6
730 DATA 449.5, 1204.7, .649, 1.4639, 500, 467,
467.5, 1203.9, .6679, 1.4495, 580, 482.6,
491.6, 1201.8, .6928, 1.4304, 700, 503.1
800 READ V(1,J) FOR J = 1 TO 6
810 DATA 35.018695, .59868737, 69.712642, .50252618, 116.06353, .43631295
820 READ V(2,J) FOR J = 1 TO 6
830 DATA 184.511, .369651, 275.517, .318439, 356.656, .287799
850 READ A$(1,J) FOR J = 1 TO 6
860 DATA .629334, -.56330, .8196, -1.147, .7921, -1.17
870 READ A$(2,J) FOR J = 1 TO 6
880 DATA .50533, .3481, .704, -.2705, .76955, -.4796
900 READ A(J) FOR J = 1 TO 6
910 DATA -.5665, 2.15, -.805695, 2.86527, -.8397, 3.056
920 READ M(J) FOR J = 1 TO 6
930 DATA .556, -.123, .79824, -.843664, .8496, -1.0657
1000 G1, G2, G3, I8 = 0
1010 FOR I2 = 1
1015 P2 = H(5, I2) ! NOZZ EXIT PRESS
1020 IF I2 = 4 THEN G1 = 1 ELSE G1 = 2
1030 FOR I = 8, 11, 12, 14
1032 Z(J) = H(J, I) FOR J = 1 TO 6
1035 P1 = H(5, I) ! NOZZ INLET PRESS
1040 G2 = G2 + 1
1050 FOR X1 = .01, .05, .1
1060 IF X1 < .09 THEN IF X1 < .05 THEN G3 = 1 ELSE G3 = 2 ELSE G3 = 3
1070 FOR N4 = .85
1200 M$ = FNA(M(2*G3-1), P1, M(2*G3)) ! M DOT FOR 1 NOZZ OF 1.0082 SQ IN
1240 V2 = FNE(V(G1, 2*G3-1), P1, V(G1, 2*G3))
1300 A$ = FNA(A$(G1, 2*G3-1), P1, A$(G1, 2*G3)) ! EXIT TO THROAT AREA RATIO
1400 A2 = FNA(A(2*G3-1), P1, A(2*G3)) ! THROAT AREA, SQ IN FOR 1.00 LBM/SEC
1480 A3 = A$*A2*M$/100 ! NOZZ EXIT AREA FOR 1 NOZZ OF 1.0082 THROAT DIA.
1490 M1 = M$
1520 FOR R1 = 15, 24, 30
1540 W4 = .033*R1 ! DIST FROM OUTER NOZZ EDGE TO ROTOR RIM ID
1560 W1 = .02*R1 ! NOZZ WALL THICK
1580 T2 = 24, A7 = 10°, W7 = .045, C = .10 ! NOZZ INCLINATION, PROBE CHAMFER ANGLE, PROBE WALL THICKNESS, SALT CONC.
1600 FOR K = .1, .2

```


4600 RETURN
5000 ! NOZZLE EXIT CONDITIONS
5050 DEF FNB(A,B,C) = (A*B+(1-A)*C)
5100 H1 = FNB(X1,H(2,I),H(1,I))
5120 S1 = FNB(X1,H(4,I),H(3,I)) ! INLET ENTROPY
5130 X2 = (S1-H(3,I2))/(H(4,I2)-H(3,I2))
5140 H5 = X2*H(2,I2)+(1-X2)*H(1,I2) ! ISENTROPIC NOZZ EXIT ENTHALPY
5150 D4 = N4*(H1-H5) ! ENTHALPY DROP ACROSS NOZZ
5160 H2 = H1 - D4 ! NOZZLE EXIT ENTHALPY
5170 E8 = .948*M2*V2^2/(64.4*738)/(M2*(H1-H5)) ! NOZZLE EFFICIENCY
5400 RETURN
6000 ! ROTARY PITOT PROBE
6050 L6 = SQRT(M2*288/(60*V3)) ! SQUARE PROBE INLET WIDTH
6060 IF L6>=W4 THEN L7= L6^2/W4 ELSE L7 = L6 !L6=PROBE INLET RADIAL DIM,L7=INLET WIDTH
6070 U3 = W7*(2*L6+L7)/(L6*L7), U4 = R3*W4/(L6*L7) ! RATIO OF PRB WALL FRONTAL AREA TO INLET AREA, TOTAL ROTOR AREA TO PRB
6100 P4 = P1,V4 = 50 ! PROBE EXIT PRESS AND VEL
6150 S7 = (V3*R2/R1-(W2+B1)*32.2*12/(M2*R1))/(1+U3*(1-.5*COS(A7*PI/180))) ! SEPARATOR VELOCITY WITH PROBE DEFLECTION LOSSES
6200 V5 = (S7^2-2500)/(2*S7)-144*32.2*((P4-(1-N7)*P2)/N7-P2)/(S7*D7) ! ROTARY PROBE VEL
6350 T4 = S7*M2*(1-X2)*R1/(32.2*12)-B1 ! TURBINE TORQUE
6400 N5 = V5*360/(PI*R1) ! PROBE RPM
6425 L9 = S7*360/(PI*R1) ! SEP RPM WITH ROTARY PITOT
6450 P7 = T4*N5/7045 ! TURBINE OUTPUT POWER
6500 E2 = P7*.948/(M2*(H1-H5)) ! SYSTEM EFFICIENCY
6525 E3 = (P7+.7*X2*M2*V2^2/(64.4*738))*948/(M2*(H1-H5)) ! SYSTEM EFFICIENCY TPPT WITH GAS TURBINE
6550 S2 = 3.3*S7^2/(2*PI^2)/1000 !! ROTOR STRESS
6600 A5 = 144 *M2 /((S7-V5)*60) ! PROBE INLET AREA, SQ IN
6700 O2 = P7/(M2*V2^2/(64.4*738)) ! RATIO OF PROBE OUTPUT TO NOZZLE JET POWER
6800 ! MULTI DECK SEPARATOR WITH GAS TURBINE
6850 S8 = (V3*(1.25-R1/(R2*4))-(W2+B1)*32.2*12/(M2*R1))/(1+U3*(1-.5*COS(A7*PI/180))) !SEPARATOR VELOCITY WITH PROBE DEFLECTION LOSS
6870 S5 = (S8^2-2500)/(2*S8)-144*32.2*((P4-(1-N7)*P2)/N7-P2)/(S8*D7) ! ROTARY PROBE VEL
6880 S9 = S8*M2*(1-X2)*R1/(32.2*12)-B1 ! TURBINE TORQUE
6885 N8 = S5*360/(PI*R1) ! PROBE RPM
6890 K7 = S9*N8/7045 ! TURBINE OUTPUT POWER
6990 E7 = (K7+X2*M2*V2^2/(64.4*738))*948/(M2*(H1-H5)) ! SYSTEM EFFICIENCY*
6995 E9 = E7*N4/E8 ! TPPT EFFICIENCY WITH NOZZLE EFFICIENCY =N4(STEP 1070)*
7250 ! ROTARY TURBINE PERFORMANCE WITH ONE NOZZLE
7350 O4 = S7*M1*(1-X2)*R1/(32.2*12)-B1 ! TURBINE TORQUE*
7450 O7 = O4*N5/7045 ! TURBINE OUTPUT POWER
7500 O3 = O7*.948/(M1*(H1-H5)) ! SYSTEM EFFICIENCY
7800 ! U TUBE TURBINE
7810 A6 = 144*(1-X2)*M2/(60*V7/2) ! U TUBE PROBE CROSS SECTIONAL AREA AQ IN
7820 C1 = .008 ! SKIN FRICTION COEFF.
7830 V8 = V7/2
7840 N6 = V8*360/(PI*R1) ! U TUBE RPM
7850 P6 = M2*(1-X2)*(V7-V8)*(2-C1/210.85)*N6*R1/(12*32.2*7047.4) ! MAX POWER
7860 E6 = P6*.948/(M2*(H1-H5)) ! EFFICIENCY
7890 RETURN
8000 !
8210 T6 = ((Z(6)-32)/1.8-11.48*C^1.32)/(1+.255*C^1.32) ! BOIL PT FOR #20 WITH SAME SAT PRESS AS BRINE , DEG C
8215 B6 = 1055.04/.45359 ! CONVERSION CONST BTU/LBM J/KGM
8220 B3 = 4.86652295E-10*(T6*1.8+32)^4.498217 ! SAT PRESS OF BRINE FOR TEMP T1, PSIA
8225 H(1,I) = Z(1)*(1-(.238A+4.275E-4*T6)*(1-(1-C/.35)^3))^1 ! LIQ BRINE ENTHALPY , BTU/LBM
8230 B4 = (P1-B3)/B3*T6^4*2.18E-11 ! ENTHALPY CHANGE
8240 H(2,I) = Z(2)*(1+B4) ! BRINE VAPOR ENTHALPY , BTU/LBM
8250 B5 = 201.0583*(T6*1.8+32)^-.2216986*16.01846 ! KGM/M3 LIQ DEN
8260 H(3,I) = 1/(R6*1.8)*(H(1,I)*B6-B3*6894.757/R5)/(273.18+ T6/2) ! LIQ BRINE ENTROPY, BTU/LBM DEG F
8270 H(4,I) = H(3,I)+(H(2,I)-H(1,I))/(.273+T6)*1.8 ! VAPOR ENTROPY ,RTU/LBM DEG F
8400 RETURN
8450 PRINT H(J,I) FOR J = 1 TO 6
A500 PRINT P2,P1,X1,A5,A2,A3,M1,R1,D2,W3,R2,N1,M2,V2,N2,N3,W2,T3,B1,V7,M5,B7,U3,P8,P9,E1

TABLE C-II (CONT'D)

TABLE C-II
SAMPLE ANALYSIS PROGRAM OUTPUT

EXIT P2	INLET P1	QUAL X1	ARTO A5	TRT-A A2	EXT A3	FLOW M1	RD M1	HT U2	WDT W3	NRD R2	NO N	TOT M2	MVN VEL V2	SEP N2	RPM N3	TORQ W2	LOSS T3	SVEL B1	FLOW V7	RV M5	NO R7	PRBP Z3	PWR P8	EFF P9	EFF E1
RINW R3	QUAL X2	H1 M5	M2	DEL D4	H P V5	VEL V5	TORQUE T4	PRPM N5	PWR P7	EFF E2	STRS S2	PRB A5	PRY O2	PWR 07	LOSS 03	AUTB A6	RPM N6	PWR P6	EFF E6	EFF E3	EFF E7				
14.7	80.0	.01	4.3	11.8	4	8.6	15	1.5	2.9	13.8	8.	170.3	483	3517	3100	9	34	2	406	8.6	.0	28	949	171	.276
5.94	.106	291.1	282.8	284.0		7.1	180	0	965	1377	189	.304	26	.784	.547	23	.304	.744	1550	218	.351	.346	.409		
.556	.625	.850	395	3021		.495	.856	1.481																	
14.7	80.0	.01	4.3	11.8	4	8.6	15	3.0	1.5	13.0	14.	120.8	483	3517	2932	11	58	2	384	8.6	.0	26	849	260	.244
10.21	.106	291.1	282.8	284.0		7.1	173	0	1600	1319	300	.281	24	1.388	.506	21	.281	1.351	1466	335	.314	.323	.412		
.556	.630	.850	382	2915		.495	1.122	2.544																	
14.7	80.0	.01	4.3	11.8	4	8.6	30	3.0	1.5	27.5	30.	258.0	483	1758	1547	95	249	1	405	8.6	.0	29	945	626	.275
21.80	.106	291.1	282.8	284.0		7.1	185	0	1253	708	729	.320	27	2.821	.576	24	.320	2.735	773	796	.350	.362	.428		
.556	.654	.850	405	1547		.990	1.640	2.718																	
14.7	80.0	.01	4.3	11.8	4	8.6	30	6.0	.7	26.0	48.	411.3	483	1758	1462	129	397	1	383	8.6	.0	25	844	881	.243
34.75	.106	291.1	282.8	284.0		7.1	175	0	11042	670	1051	.290	25	4.673	.521	22	.290	4.613	731	1133	.312	.331	.423		
.556	.647	.850	387	1477		.990	2.071	4.332																	
14.7	80.0	.05	2.6	21.5	3	4.7	15	1.5	1.8	13.8	13.	63.1	630	4594	4033	14	40	3	528	4.7	.0	10	1606	260	.300
5.33	.138	327.2	314.1	316.1		11.1	239	0	1067	1825	276	.319	43	.569	.524	21	.319	.494	2017	319	.368	.377	.449		
.608	.627	.850	505	3860		.495	.710	1.018																	
14.7	80.0	.05	2.6	21.5	3	4.7	15	3.0	.9	13.0	22.	106.4	630	4594	3818	18	67	3	500	4.7	.0	28	1439	391	.267
8.99	.138	327.2	314.1	316.1		11.1	231	0	1748	1767	438	.300	40	.984	.493	20	.300	.881	1909	482	.330	.359	.458		
.608	.640	.850	491	3750		.495	.922	1.716																	
14.7	80.0	.05	2.6	21.5	3	4.7	30	3.0	.9	27.5	46.	220.1	630	2297	2011	148	278	1	526	4.7	.0	18	1596	902	.298
18.60	.138	327.2	314.1	316.1		11.1	247	0	7662	943	1026	.339	45	1.932	.557	22	.339	1.730	1005	1106	.366	.398	.472		
.608	.660	.850	520	1987		.990	1.326	1.775																	
14.7	80.0	.05	2.6	21.5	3	4.7	30	6.0	.4	26.0	70.	332.7	630	2297	1901	190	420	1	498	4.7	.0	21	1427	1211	.265
28.11	.138	327.2	314.1	316.1		11.1	235	0	11095	899	1416	.310	42	3.035	.509	20	.310	2.765	951	1494	.327	.368	.470		
.608	.657	.850	498	1904		.990	1.630	2.682																	
14.7	80.0	.10	2.2	28.7	2	3.6	15	1.5	1.5	13.8	16.	55.6	785	5722	4999	21	44	5	654	3.6	.0	22	2467	342	.309
4.70	.179	372.2	353.3	356.2		16.0	295	0	1087	2253	347	.314	63	.421	.481	22	.314	.335	2500	412	.372	.395	.475		
.652	.620	.850	612	4678		.495	.597	.720																	
14.7	80.0	.10	2.2	28.7	2	3.6	15	3.0	.7	13.0	26.	93.1	785	5722	4739	26	73	5	620	3.6	.0	17	2217	513	.276
7.87	.179	372.2	353.3	356.2		16.0	289	0	1782	2204	557	.301	60	.717	.461	21	.301	.592	2370	619	.334	.382	.490		
.652	.639	.850	600	4584		.495	.772	1.205																	
14.7	80.0	.10	2.2	28.7	2	3.6	30	3.0	.7	27.5	53.	189.9	785	2861	2491	212	298	1	652	3.6	.0	31	2449	1159	.307
16.04	.179	372.2	353.3	356.2		16.0	307	0	7695	1172	1280	.339	68	1.387	.520	24	.339	1.148	1245	1395	.369	.420	.504		
.652	.657	.850	636	2428		.990	1.103	1.229																	
14.7	80.0	.10	2.2	28.7	2	3.6	30	6.0	.4	26.0	79.	280.9	785	2861	2356	265	441	1	617	3.6	.0	21	2192	1528	.273
23.73	.179	372.2	353.3	356.2		16.0	294	0	10952	1124	1748	.313	62	2.127	.480	22	.313	1.795	1178	1848	.330	.394	.505		
.652	.659	.850	611	2335		.990	1.342	1.818																	
14.7	150.0	.01	6.4	8.3	6	12.2	15	1.5	4.3	13.8	6.	68.2	703	5124	4495	18	48	4	588	12.2	.0	31	1994	337	.284
5.7	.147	339.2	322.8	325.2		14.0	258	0	1270	1971	355	.300	53	.539	.500	64	.300	.475	2248	424	.358	.362	.4		
.614	.850	562	4294		.495	.699	.987																		

TABLE C-II (CONT'D)

EXIT	INLET	GUAL	ARTO	TRT-A	EXT	FLOW	RD	HT	WDT	NRD	NO	N	TOT	M	N	VEL	SEP-RPM	TORG	-LOSS	SVEL	FLOW	RV	NO	PRBP	PWR	FFF
P2	P1	X1	A3	A2	A3	M1	R1	D2	W3	R2	N	N	M2	V2	N2	N3	W2	B1	V7	M*	R7	23	PA	P9	ET	
RIMW	GUAL	ENTHALPY				DEL	H	P	VEL	TORQUE	PRPM	PWR	EFF	STRS	PRB	AR	PRIO	PWR	EFF	AUTB	RPM	PWR	FFF	FFF	FFF	
R3	X2	H1	H3	H2		D4		V5	P5	T4	N5		P7	E2	S2	A5		O7	O3	A6	N6	P6	E6	E3	E7	
NEFF	REFF	NEFF	VEL	RSKPM	RTRD	PRBHT	PRBWDTH																			
E8	E9	N4	S7	L9	W4	L6	L7																			
14.7	150.0	.01	6.4	8.3	6	12.2	15	3.0	2.2	13.0	10.	118.3	703	5124	4257	24	83	4	557	12.2	.0	24	1788	519	.252	
10.00	.147	339.2	322.8	325.2	14.0	250	0	2143	1909	581	.283	50	.955	.472	60	.283	.869	2128	659	.321	.344	.443				
.599	.628	.850	547	4180	.495	.920	1.711																			
14.7	150.0	.01	6.4	8.3	6	12.2	30	3.0	2.2	27.5	21.	257.2	703	2562	2241	201	362	1	587	12.2	.0	26	1983	1261	.282	
21.73	.147	339.2	322.8	325.2	14.0	268	0	9891	1024	1438	.322	56	1.973	.537	68	.322	1.794	1121	1589	.356	.384	.458				
.599	.650	.850	581	2219	.990	1.357	1.859																			
14.7	150.0	.01	6.4	8.3	6	12.2	30	6.0	1.1	26.0	35.	422.4	703	2562	2119	278	594	1	555	12.2	.0	21	1773	1834	.250	
35.69	.147	339.2	322.8	325.2	14.0	255	0	15587	975	2158	.294	52	3.357	.491	62	.294	3.116	1060	2333	.318	.356	.457				
.599	.649	.850	557	2129	.990	1.739	3.053																			
14.7	150.0	.05	4.3	12.9	4	7.8	15	1.5	2.9	13.8	8.	63.8	865	6300	5504	27	55	7	721	7.8	.0	33	2990	468	.307	
8.39	.176	373.8	351.1	354.5	19.2	319	0	1379	2437	477	.313	76	.428	.475	59	.313	.350	2752	574	.377	.394	.475				
.659	.613	.850	676	5166	.495	.609	.750																			
14.7	150.0	.05	4.3	12.9	4	7.8	15	3.0	1.5	13.0	14.	109.6	865	6300	5218	35	95	7	683	7.8	.0	21	2687	718	.274	
9.26	.176	373.8	351.1	354.5	19.2	312	0	2323	2382	785	.300	73	.749	.456	56	.300	.634	2609	886	.338	.381	.491				
.659	.633	.850	663	5064	.495	.799	1.289																			
14.7	150.0	.05	4.3	12.9	4	7.8	30	3.0	1.5	27.5	30.	234.2	865	3150	2742	289	405	2	718	7.8	.0	27	2969	1705	.305	
19.78	.176	373.8	351.1	354.5	19.2	333	0	10529	1272	1902	.340	83	1.519	.516	64	.340	1.289	1371	2092	.374	.421	.506				
.659	.653	.850	703	2686	.990	1.167	1.377																			
14.7	150.0	.05	4.3	12.9	4	7.8	30	6.0	.7	26.0	48.	373.3	865	3150	2595	386	646	2	679	7.8	.0	17	2658	2418	.271	
31.64	.176	373.8	351.1	354.5	19.2	319	0	16160	1220	2798	.314	77	2.505	.477	59	.314	2.172	1297	2985	.335	.395	.508				
.659	.656	.850	677	2586	.990	1.474	2.195																			
14.7	150.0	.10	3.6	16.9	4	6.1	15	1.5	2.5	13.8	10.	58.7	1033	7527	6544	37	61	9	857	6.1	.0	21	4226	590	.314	
4.96	.213	417.0	386.6	391.1	25.8	377	0	1411	2881	577	.307	104	.343	.438	60	.307	.259	3272	713	.379	.412	.501				
.701	.607	.850	787	6015	.495	.535	.577																			
14.7	150.0	.10	3.6	16.9	4	6.1	15	3.0	1.2	13.0	17.	100.2	1033	7527	6212	47	104	9	813	6.1	.0	21	3807	905	.282	
8.47	.213	417.0	386.6	391.1	25.8	372	0	2384	2845	963	.300	101	.893	.428	58	.300	.466	3106	1098	.342	.404	.522				
.701	.633	.850	778	5945	.495	.699	.987																			
14.7	150.0	.10	3.6	16.9	4	6.1	30	3.0	1.2	27.5	35.	212.1	1033	3764	3259	390	438	2	853	6.1	.0	23	4191	2116	.311	
17.92	.213	417.0	386.6	391.1	25.8	397	0	10694	1516	2301	.339	114	1.189	.483	66	.339	.939	1629	2556	.376	.443	.537				
.701	.651	.850	825	3151	.990	1.016	1.044																			
14.7	150.0	.10	3.6	16.9	4	6.1	30	6.0	.6	26.0	55.	332.6	1033	3764	3086	510	687	2	808	6.1	.0	21	3758	2963	.278	
28.10	.213	417.0	386.6	391.1	25.8	383	0	16216	1462	3364	.316	106	1.923	.450	61	.316	1.556	1543	3594	.337	.420	.543				
.701	.658	.850	798	3047	.990	1.273	1.637																			
14.7	225.0	.01	8.3	6.6	8	15.3	15	1.5	5.5	13.8	4.	66.8	896	6531	5706	29	60	7	747	15.3	.0	20	3213	516	.307	
8.65	.176	374.5	350.7	354.3	20.3	323	0	1500	2468	525	.312	82	.424	.465	120	.312	.354	2853	647	.384	.395	.477				
.672	.604	.850	702	5360	.495	.613	.758																			
14.7	225.0	.01	8.3	6.6	8	15.3	15	3.0	2.8	13.0	8.	116.4	896	6531	5409	39	104	7	708	15.3	.0	26	2887	801	.273	
9.83	.176	374.5	350.7	354.3	20.3	316	0	2561	2411	877	.299	79	.750	.445	115	.299	.650	2705	1012	.345	.382	.493				
.672	.625	.850	688	5255	.495	.808	1.320																			

TABLE C-II (CONT'D)

EXIT	INLET	QUAL	ARTO	TRT-A	EXT	FLOW	RD	MT	WDT	NRD	NO	M	TOT	M	N	VEL	SEP-RPM	TORQ	-LOSS	SVEL	FLOW	RV	NO	PRBP	PWR	EFF	
P2	P1	X1	A5	A2	A3	M1	R1	U2	W3	R2	N	M2	M2	N	V2	N2	N3	W2	T3	B1	V7	M5	R7	Z3	P8	P9	E1
RIMW	QUAL	-----	ENTHALPY	-----	DEL	H	P	VEL	TORQUE	PRPM	PWR	EFF	STRS	PRB	AR	PRTIO	PWR	EFF	AUTB	RPM	PWR	EFF	EFF	EFF	EFF	EFF	
R3	X2	M1	M2	M2	U4	V5	P5	T4	N5	P7	E2	S2	A5	O2	O7	O3	A6	N6	P6	E6	E3	E7					
NEFF	REFF	NEFF	VEL	RSRPM	RTD	PRBHT	PRWDTH																				
E8	E9	N4	S7	L9	W4	L6	L7																				
14.7	225.0	.01	8.3	6.6	8	15.3	30	3.0	2.8	27.5	17.	255.1	896	3266	2843	326	457	2	744	15.3	.0	21	3191	1956	.305		
21.55	.176	374.5	350.7	354.3	20.3	339	0	11921	1293	2188	.341	89	1.562	.507	131	.341	1.356	1422	2451	.382	.423	.510					
.672	.645	.850	730	2790	.990	1.197	1.447																				
14.7	225.0	.01	8.3	6.6	8	15.3	30	6.0	1.4	26.0	28.	425.3	896	3266	2690	454	763	2	704	15.3	.0	20	2857	2894	.270		
35.93	.176	374.5	350.7	354.3	20.3	324	0	19149	1238	3366	.314	85	2.689	.468	121	.314	2.388	1345	3658	.342	.397	.513					
.672	.649	.850	704	2688	.990	1.545	2.412																				
14.7	225.0	.05	6.0	9.3	6	10.8	15	1.5	4.1	13.8	6.	64.0	1060	7724	6720	40	68	10	880	10.8	.0	23	4456	677	.324		
5.41	.203	407.9	377.0	381.6	26.3	382	0	1609	2921	667	.320	110	.357	.441	113	.320	.278	3360	830	.398	.423	.514					
.725	.603	.850	813	6209	.495	.551	.614																				
14.7	225.0	.05	6.0	9.3	6	10.8	15	3.0	2.0	13.0	10.	110.8	1060	7724	6377	53	117	10	835	10.8	.0	18	4013	1049	.290		
9.36	.203	407.9	377.0	381.6	26.3	377	0	2750	2879	1124	.311	108	.625	.429	110	.311	.508	3189	1295	.358	.414	.536					
.725	.628	.850	802	6129	.495	.725	1.063																				
14.7	225.0	.05	6.0	9.3	6	10.8	30	3.0	2.0	27.5	22.	240.2	1060	3862	3346	440	509	2	876	10.8	.0	31	4420	2520	.322		
20.30	.203	407.9	377.0	381.6	26.3	403	0	12657	1539	2765	.353	121	1.285	.487	124	.353	1.049	1673	3092	.395	.456	.552					
.725	.647	.850	851	3252	.990	1.068	1.152																				
14.7	225.0	.05	6.0	9.3	6	10.8	30	6.0	1.0	26.0	36.	393.0	1060	3862	3168	601	833	2	829	10.8	.0	25	3962	3672	.287		
33.21	.203	407.9	377.0	381.6	26.3	388	0	20017	1482	4209	.329	113	2.167	.453	116	.329	1.813	1584	4534	.354	.432	.558					
.725	.654	.850	823	3144	.990	1.366	1.885																				
14.7	225.0	.10	4.9	12.0	5	8.6	15	1.5	3.4	13.8	7.	60.4	1233	8984	7779	53	74	13	1018	8.6	.0	28	5971	831	.328		
5.10	.237	449.6	409.9	415.8	33.8	440	0	1652	3364	789	.312	142	.300	.408	112	.312	.217	3890	1006	.397	.438	.538					
.764	.599	.850	923	7051	.495	.497	.498																				
14.7	225.0	.10	4.9	12.0	5	8.6	15	3.0	1.7	13.0	12.	104.2	1233	8984	7391	69	128	13	968	8.6	.0	29	5391	1288	.295		
8.80	.237	449.6	409.9	415.8	33.8	438	0	2834	3346	1346	.308	141	.521	.404	111	.308	.394	3696	1566	.359	.435	.565					
.764	.629	.850	918	7015	.495	.652	.859																				
14.7	225.0	.10	4.9	12.0	5	8.6	30	3.0	1.7	27.5	26.	224.2	1233	4492	3871	573	553	3	1014	8.6	.0	34	5915	7054	.325		
18.94	.237	449.6	409.9	415.8	33.8	468	0	12958	1787	3286	.350	159	1.060	.458	126	.350	.810	1936	3698	.394	.476	.582					
.764	.648	.850	975	3726	.990	.957	.957																				
14.7	225.0	.10	4.9	12.0	5	8.6	30	6.0	.8	26.0	42.	361.9	1233	4492	3668	769	893	3	960	8.6	.0	22	5311	4408	.291		
30.58	.237	449.6	409.9	415.8	33.8	452	0	20273	1727	4969	.328	149	1.761	.429	118	.328	1.380	1834	5359	.343	.454	.590					
.764	.657	.850	945	3611	.990	1.215	1.492																				
14.7	400.0	.01	11.9	4.7	12	21.1	15	1.5	7.9	13.8	3.	64.8	1265	9217	7986	58	82	14	1045	21.1	.0	32	6292	974	.357		
5.47	.220	432.0	393.8	399.5	32.5	439	0	1866	3357	889	.340	151	.303	.408	289	.340	.232	3993	1161	.444	.469	.575					
.835	.586	.850	952	7270	.495	.508	.521																				
14.7	400.0	.01	11.9	4.7	12	21.1	15	3.0	4.0	13.0	5.	113.3	1265	9217	7587	76	143	14	993	21.1	.0	29	5679	1443	.320		
9.58	.220	432.0	393.8	399.5	32.5	436	0	3245	3334	1536	.336	150	.534	.402	286	.336	.427	3793	1834	.401	.465	.605					
.835	.616	.850	946	7227	.495	.671	.911																				
14.7	0.0	.01	11.9	4.7	12	21.1	30	3.0	4.0	27.5	12.	250.6	1265	4609	3975	642	634	3	1041	21.1	.0	35	6235	3577	.357		
21.5	.220	432.0	393.8	399.5	32.5	468	0	15242	1787	3866	.382	169	1.120	.458	325	.382	.901	1987	4452	.440	.511	.611					
.635	.635	.850	1005	3837	.990	.999	1.007																				

TABLE C-II (CONT'D)

EXIT	INLET	QUAL	ARTO	TRT-A	EXT	FLOW	RD	HT	WDT	NRD	NO	N	TOT	M	N	VEL	SEP-RPM	TORG	-LOSS	SVEL	FLOW	RV	NO	PRRP	PWR	EFF		
P2	P1	X1	A5	A2	A3	M1	R1	D2	W3	R2	N	N	M2	M	N	V2	N2	N3	W2	T3	B1	Y7	M5	R7	Z3	PA	P9	E1
RIMW	QUAL	-----ENTHALPY-----				DEL	H	P	VEL	TORQUE	PRPM	PWR	EFF	STRS	PRB	AR	PRTIO	PWR	EFF	AUTB	RPM	PWR	EFF	EFF	EFF			
R3	X2	H1	H5	H2	D4	V5	P5	T4	N5	P7	E2	S2	A5	O2	O7	O3	A6	N6	P6	E6	F3	E7						
NEFF	REFF	NEFF	VEL	RSRPM	RTRO	PRBHT	PRRWTH																					
E8	E9	N4	S7	L9	W4	L6	L7																					
14.7	400.0	.01	11.9	4.7	12	21.1	30	6.0	2.0	26.0	20.	424.9	1265	4609	3766	904	1075	3	986	21.1	.0	29	559A	5401	.315			
35.90	.220	432.0	393.8	399.5	32.5	452	0	25086	1727	6150	.359	159	1.949	.430	305	.359	1.613	1883	6776	.395	.487	.634						
.835	.646	.850	975	3725	.990	1.300	1.707																					
14.7	400.0	.05	9.7	5.9	10	17.1	15	1.5	6.5	13.8	4.	64.0	1416	10313	8904	72	81	17	1166	17.1	.0	33	7823	1126	.361			
5.41	.244	463.2	416.9	423.9	39.3	493	0	1973	3763	1054	.337	185	.275	.391	282	.337	.199	4452	1382	.443	.485	.599						
.864	.589	.850	1051	8029	.495	.477	.477																					
14.7	400.0	.05	9.7	5.9	10	17.1	15	3.0	3.2	13.0	7.	111.7	1416	10313	8466	94	158	17	1108	17.1	.0	26	7072	1767	.324			
9.44	.244	463.2	416.9	423.9	39.3	490	0	3431	5747	1825	.335	183	.481	.388	280	.335	.366	4233	2181	.400	.482	.630						
.864	.620	.850	1047	7999	.495	.630	.802																					
14.7	400.0	.05	9.7	5.9	10	17.1	30	3.0	3.2	27.5	14.	245.8	1416	5157	4429	795	696	4	1160	17.1	.0	35	7743	4279	.357			
20.77	.244	463.2	416.9	423.9	39.3	526	0	16075	2010	4585	.382	208	1.003	.442	319	.382	.769	2215	5256	.438	.530	.650						
.864	.640	.850	1114	4257	.990	.935	.935																					
14.7	400.0	.05	9.7	5.9	10	17.1	30	6.0	1.6	26.0	24.	413.1	1416	5157	4199	1109	1170	4	1099	17.1	.0	32	6958	6424	.319			
34.91	.244	463.2	416.9	423.9	39.3	509	0	26233	1944	7239	.359	196	1.730	.416	300	.359	1.364	2099	7939	.394	.507	.660						
.864	.650	.850	1082	4133	.990	1.212	1.484																					
14.7	400.0	.10	7.8	7.4	8	14.0	15	1.5	5.4	13.8	4.	62.7	1585	11548	9929	89	99	22	1300	14.0	.0	32	9726	1331	.358			
5.30	.274	502.2	445.9	454.4	47.9	551	0	2053	4211	1227	.329	226	.246	.370	273	.329	.168	4964	1617	.434	.500	.622						
.890	.594	.850	1162	8877	.495	.446	.446																					
14.7	400.0	.10	7.8	7.4	8	14.0	15	3.0	2.7	13.0	8.	109.1	1585	11548	9450	116	173	22	1237	14.0	.0	30	8811	2090	.322			
9.22	.274	502.2	445.9	454.4	47.9	549	0	3559	4192	2118	.327	224	.430	.367	271	.327	.307	4725	2550	.393	.497	.652						
.890	.623	.850	1157	8842	.495	.589	.700																					
14.7	400.0	.10	7.8	7.4	8	14.0	30	3.0	2.7	27.5	17.	238.9	1585	5774	4936	981	757	5	1292	14.0	.0	32	9613	5013	.353			
20.19	.274	502.2	445.9	454.4	47.9	590	0	16641	2253	5322	.375	255	.888	.421	311	.375	.644	2468	6093	.429	.546	.674						
.890	.644	.850	1236	4720	.990	.871	.871																					
14.7	400.0	.10	7.8	7.4	8	14.0	30	6.0	1.3	26.0	28.	397.8	1585	5774	4682	1355	1261	5	1226	14.0	.0	40	8650	7473	.316			
33.61	.274	502.2	445.9	454.4	47.9	571	0	26884	2180	8319	.352	240	1.519	.396	292	.352	1.131	2341	9127	.386	.523	.684						
.890	.654	.850	1199	4581	.990	1.124	1.276																					

EDIT 1580

1580 T2 = 17.5 , A7 = 10 , W7 = .045 ; NOZZ INCLINATION, PROBE CHAMFER ANGLE, PROBE WALL THICKNESS

1580 T2 = 24

1580 T2 = 24 , A7 = 10 , W7 = .045 ; NOZZ INCLINATION, PROBE CHAMFER ANGLE, PROBE WALL THICKNESS

RUN

APPENDIX D
DRAWING INDEX

<u>Drawing No.</u>	<u>Title</u>	<u>Sheet No.</u>
76-102-1A	Separator Assembly	1
76-102-1B	Geothermal Separator - Mod II	
76-102-2	Separator End Plate Details	
76-102-3	Separator End Plates Welding Assembly and Machining Details	
76-102-4	Separator Support Welding Details	
76-102-5	Round Nozzle	2
76-102-6	Bearing Assembly and Details	
76-102-7	Rotor Assembly	3
76-102-8	Geothermal Mobile Test Vehicle	
76-102-9	Nozzle Adapter Assembly and Details (Round)	
76-102-10	Nozzle Assembly	4
76-102-11	Nozzle Adapter Assembly and Details (Rectangular)	
76-102-12	Nozzle Adapter Assembly and Details (Rectangular)	
76-102-13	Geothermal Mobile Test Assembly	
76-102-14	Flange Adapter	
76-102-15	Nozzle Calibration Assembly	
76-102-16	Steam Outlet Flange	
76-102-18	Rotary Separator Test Schematic	5
76-102-19	Drum Rework	
76-102-20	Probe Assembly and Details	6
76-102-21	Lever, Torque Test, Probe	
76-102-22	Flange Assembly	
76-102-23	Seal Modification	
76-102-24	Seal Modification	
76-102-25	Probe Inlet No. 1	
76-102-26	Probe Inlet No. 2	
76-102-27	Assembly, Pickup Probe	7
76-102-28	Seal Arrangement, Housing Bearing	
76-102-29	End Plate Rework	
76-102-30	Steam Exhaust Flange Rework

NOTE:
 1) SEE DWG 86-103 FOR SEPARATOR END PLATE DETAILS.
 DWG 86-104 FOR SEPARATOR END PLATE WELDING ASSY
 & MACHINING DETAILS
 DWG 86-105 FOR SEPARATOR SUPPORT WELDING DETAILS.
 DWG 86-107 FOR SEPARATED BEARING ASSY.
 (18) ITEMS (2) (4) (6) ARE USED W/ (2) DRAIN OF 7.68" WIDTH.
 (13) ITEMS (2) (4) (6) ARE USED W/ (2) DRAIN OF 6.18" WIDTH.

TOLERANCES EXCEPT AS NOTED
 .0005 .010 .001 60 ANGLES
 SURFACE ROUGHNESS 125/ EXCEPT
 MEAT TREAT

SPECIAL

Scale $\frac{1}{8}$ " = 1"
 (EXCEPT WHERE NOTED)

DATE	6-3-76	APPROV	DWG NO	BE-101
BY	Harvey Kemp		PROJECT	102
DESCRIPTION	SEPARATOR ASSY.			

BIPHASE ENGINES, INC.
307 OCEAN PARK BLVD
SANTA MONICA, CALIF.
PHONE 450-8892
DWG NO BE-101
PROJECT 102

TELEPHONE BOOKS AND NOTES 4-24-78 - 410 - 474-12-02 (HOLLYWOOD) SURFACE DISCHARGE 410-474-12-02 JULY TRAIL		DRAWING TITLE ROUND NOZZLE		BIPHASE ENGINES INC. 2707 OCEAN PARK BLVD. SANTA MONICA, CALIF. PHONE 432-2224	
REVISION		DATE 6-5-78	APPROVED	DWG. NO. 76-102-5	
SCALE : FULL		DESIGN Kenny A. J.	BY JH	PRICE 102

NOTE:
(1) DEBUR SHARP CORNERS & EDGES

[illegible]

view A-A

VIEW 13 - 13

DATE RECEIVED BY FBI MONTH DAY YEAR FBI OFFICE FBI DIVISION		FBI NO. 102 SEARCHED ☐ INDEXED ☐ SERIALIZED ☐ FILED ☐ APR 12 1968 FBI - NEW YORK		DIAPHASE ENGINEERS, INC. R353, PICK UP PROBE EXPERIMENTAL NEW YORK BE-127	
DATE SENT BY FBI MONTH DAY YEAR FBI OFFICE FBI DIVISION		FBI NO. 102 SEARCHED ☐ INDEXED ☐ SERIALIZED ☐ FILED ☐ APR 12 1968 FBI - NEW YORK		DIAPHASE ENGINEERS, INC. R353, PICK UP PROBE EXPERIMENTAL NEW YORK BE-127	