**Transamerica
Delaval**

Transamerica Delaval Inc.
Biphase Energy Systems
2800 Airport Ave.
Santa Monica, CA. 90405-6188
213-391-0691
Telex: 472-0221 (Biphase SNM)

DOE/ET/27124--T3

DE85 008337

TEST OF A ROTARY SEPARATOR TURBINE WITH SEPARATED FEED
CONIGURATION

Contract No. DE-AC03-79ET27124

Prepared for the
UNITED STATES DEPARTMENT OF ENERGY

March 14, 1985

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MASTER

Jeff Record
Project Engineer

Donald J. Cerini
Program Manager

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INTRODUCTION

The Rotary Separator Turbine (RST) has been developed for power generation from two-phase (liquid/gas) streams. One application of the RST is for power production from a liquid-dominated geothermal resource. The theory of operation of the RST is presented in previous papers.^{1,2} This report assumes that the reader is familiar with the theory of operation of the RST.

In September 1981, a power skid, consisting of a full scale RST and generators, was delivered to Phillips Petroleum geothermal production well 54-3 at Roosevelt Hot Springs, Utah. In the 6 weeks that followed, auxiliary systems were installed and checked out, to complete installation of the RST test system. These auxiliary systems included process supply and disposal piping, process controls, instrumentation, seal and lube oil supply, and electrical transformer and switchgear required for generator interface to the utility grid. This was the first installation of a full scale RST on a geothermal well, with full scale meaning the RST is sized to process the entire output of the well.

Performance tests were run from October 1981, to October 1982. The results were run by Biphase Energy Systems/Transamerica Delaval, Utah Power and Light, and Phillips Petroleum, with the sponsorship of the Electric Power Research Institute. Results of these tests are summarized in previous reports.^{2,3} From October 1982 to April 1983, the RST was run continuously on a non-interference basis with other site operations. This test period is called the Endurance Test. The goals of the test were to demonstrate ease of operation, durability in the geothermal environment, and availability, at levels high enough to allow use of the RST in a geothermal power plant. Test results are summarized in previous papers.³

Following the Endurance Test, the RST test system was reconfigured to maximize the power output (and minimize cost/kW) of an RST power plant. Further performance test were run from March 1984 to June 1984. These test are called the Upgrade Tests. The results of this series of tests are presented in this report.

Section II is a description of the test system used during the Endurance Tests. Results of the first series of performance tests (10/81 - 10/82), and of the Endurance Test, are briefly discussed in this section.

Section III presents the rationale for the reconfiguration of the RST power system. Test system changes, and a summary of the test system shakedown are presented in Section IV.

The Upgrade Test series consists of three short (2-3 day) test periods. Test system operations, as well as immediate conclusions and system modifications, are presented in Sections V through VII. Results and conclusions are presented in Section VIII.

II. FIELD TESTING OF 54-INCH RST (8/81 to 4/83)

Test System Description

The Control and Instrumentation Diagram for the test system used from 8/81 to 4/83 is shown as Figure II-1. The well supplied two-phase flow directly to the RST test system. Flow to the RST was split into four equal portions by a flow splitter. The flow then went through a pneumatic nozzle-throttle valve (fully open during normal operation) and a mixer (Kenics) before entering the two-phase nozzle.

Flow could be bypassed around the RST in either of two ways. The main bypass through valve WFBV (see Figure II-1) was used during well warm-up and RST start-up. The emergency bypass, valves DV1-DV4, was used in event of an RST trip to maintain well flow. During an emergency shutdown, the nozzle-throttle valves NT1-NT4 would close while the emergency bypass valves DV1-DV4 would open. This would happen during a period of about one second, isolating the RST while maintaining constant well flow rate.

All bypass flow was directed through the accumulator. The accumulator separated the liquid from the steam. The steam was combined with the RST outlet steam, the flow rate was measured, and the steam was vented to atmosphere through SVV1

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TITLE	DESCRIPTION	OPERATION
ALCV	ACCUMULATOR LIQUID-LEVEL CONTROL VALVE	P
APCV	ACCUMULATOR PRESSURE-CONTROL VALVE	P
BBV	BYPASS BLOCK VALVE	M
BRV	BRINE RECIRCULATION VALVE	M
BTU	BRINE THROTTLING VALVE	P
CCOV	CASE DRAIN DISPOSAL VALVE 1	M
CCOV2	CASE DRAIN DISPOSAL VALVE 2	M
CCOV3	CASE DRAIN DISPOSAL VALVE 3	M
CCOV4	CASE DRAIN DISPOSAL VALVE 4	M
CCOV5	CASE DRAIN DISPOSAL VALVE 5	M
CCOV6	CASE DRAIN DISPOSAL VALVE 6	M
CCOV7	CASE DRAIN DISPOSAL VALVE 7	M
CCOV8	CASE DRAIN DISPOSAL VALVE 8	M
CCOV9	CASE DRAIN DISPOSAL VALVE 9	M
CCOV10	CASE DRAIN DISPOSAL VALVE 10	M
CCOV11	CASE DRAIN DISPOSAL VALVE 11	M
CCOV12	CASE DRAIN DISPOSAL VALVE 12	M
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CCOV99	CASE DRAIN DISPOSAL VALVE 99	M
CCOV100	CASE DRAIN DISPOSAL VALVE 100	M

NOMENCLATURE-VALVES

⌵	GATE VALVE
⌵	GLOBE VALVE
⌵	BUTTERFLY VALVE
⌵	BALL VALVE
⌵	MANUALLY OPERATED GATE VALVE
⌵	PNEUMATIC BALL VALVE
⌵	SLUICE

NOMENCLATURE-INSTRUMENTATION

○	DISPLACEMENT (ANAL)
○	DIFFERENTIAL PRESSURE
○	CURRENT
○	PRESSURE TRANSDUCER
○	POWER
○	POWER FACTOR
○	REACTIVE POWER
○	SPEED PICK-UP
○	RESISTANCE TEMPERATURE DETECTOR
○	THERMOCOUPLE
○	VIBRATION (RADIAL)
○	VALVE POSITION
○	ORIFICE PLATE
○	SLUICE (W/COOLED) BEARING
○	REINJECTION PUMP
○	PRODUCTION WELL

DATA SYSTEM SUMMARY

CHG	NAME	DESCRIPTION
0	TC-0	THERMOCOUPLE JUNCTION REFERENCE TEMPERATURE
1	TC-1	SYSTEM INLET TEMPERATURE
2	TC-2	NOZZLE THROTTLE VALVE #1 INLET TEMPERATURE
3	TC-3	NOZZLE THROTTLE VALVE #2 INLET TEMPERATURE
4	TC-4	NOZZLE THROTTLE VALVE #3 INLET TEMPERATURE
5	TC-5	NOZZLE THROTTLE VALVE #4 INLET TEMPERATURE
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99	TC-99	NOZZLE THROTTLE VALVE #98 INLET TEMPERATURE
100	TC-100	NOZZLE THROTTLE VALVE #99 INLET TEMPERATURE

LET	CHANGE	DATE	BY
A	REVISED	1-20-66	DS
B	REVISED	2-21-66	DS

Figure II-1. Performance Test Control and Instrumentation Diagram

RST TEST
CONTROL AND INSTRUMENTATION
DIAGRAM
A 55-1233 B

and SVV2. The steam bypass to the Phillips system, SVV3, was not used. The brine flow rate out of the accumulator was also measured. The reinjection pump was never used for more than a few hours, and all liquid was directed to injection well 82-33.

The rotary separator was coupled, through a gear box, to a 600 kW drag generator. RST performance was measured as a function of drag loading. Although hardware was in place to synchronize this generator to the utility grid, the generator was never synchronized. The power from this generator was dissipated by a 400 kW load bank.

The liquid turbine drove the 2.2 MW liquid turbine generator. The coupling was direct, so that when synchronized to the grid, the liquid turbine was constrained to rotate at 1800 rpm. During start-up on one nozzle, the electrical load was supplied by a 400 kW load bank, until the generator could be synchronized to the grid. RST performance was measured as a function of RST casing pressure, flow rate (1-4 nozzles), inlet pressure and drag loading.

The RST steam output left the casing through the 24-inch steam exhaust. The RST steam flow rate was measured upstream of the throttle valve. RST casing pressure was controlled by the outlet steam throttle valve, STV. Samples were taken periodically to evaluate RST outlet steam quality.

The liquid left the RST casing through the diffuser. Flow rate was measured using an orifice plate. The downstream throttle valve, BTV, was used to simulate various reinjection pressure requirements, and to subcool the liquid for the flow measurement.

Any spilled liquid left the RST casing through the case drain. A throttle valve was used to maintain a liquid seal in the line.

Instrumentation is summarized on Figure II-1. An HP 9825 desktop computer was used as the controller for the data acquisition system, which consisted of an HP scanner, printer and voltmeter. 80 data channels could be scanned over a period of several seconds. The data was reduced in the field, and was saved on magnetic tape for post-test data analysis.

Performance Test Results

Performance data was taken periodically from August 1981 to October 1982. During this time, tests were run at several different well characteristics, dependent on the degree of downhole scaling. Maximum RST flowrate of 515 klbm/hr was achieved with a wellhead pressure of 410 psia.

A primary machine performance parameter evaluated was machine efficiency, η_{RST} . RST machine efficiency is defined in Equation 1.

$$\eta_{RST} = \frac{P}{\dot{m} (\Delta H)_s} \quad (1)$$

where

P = RST shaft power (both shafts)

$\dot{m}$ = total flow rate through RST

and $(\Delta H)_s$ = isentropic enthalpy change across the two-phase nozzles.

RST efficiency, as a function of casing pressure, is shown on Figure II-2, for 1, 2, 3, and 4 nozzle operation, at the maximum well flow characteristic. Maximum efficiency measured was roughly 36% at a casing pressure of 43 psia. For this condition, RST inlet pressure was 406 psia, with an inlet steam fraction of 0.061. Maximum electrical power of 1491 kW was measured at this point. The solid lines represent predicted efficiency, for 1, 2, 3 and 4-nozzle operation.

If this RST configuration were used in a power plant, a steam turbine would operate on the steam exhausted from the RST. By

RST EFFICIENCY

Figure II-2. RST Efficiency (at peak flow rate) vs. RST Casing Pressure

measuring the RST outlet steam flow rate and pressure, and using an estimated steam turbine efficiency, the power output that would be expected from an RST power plant was calculated. Similarly, the power output expected from an optimized single-stage flash system was calculated.

Steam turbine efficiency, η_{ST} , is predicted using the Baumann's rule:

$$\eta_{ST} = \frac{(1 + X)}{2} \cdot 0.85 \quad (2)$$

where X = calculated steam quality at the steam turbine exit. This is an iterative calculation, as steam turbine exit quality is dependent on η_{ST} .

The intermediate temperature for an optimized single-flash system is the average of the downhole and condensing temperatures. Condensing pressure is assumed to be 2 psia for both systems. The ratio of the calculated power output of an RST system to the calculated power output of an optimized single-stage flash system is defined as the Resource Utilization Ratio, RUR.

RUR as a function of casing pressure, for 1, 2, 3 and 4 nozzle operation, is shown in Figure II-3. RUR reaches a maximum

Figure II-3. RUR vs. Casing Pressure, with Maximum Wellflow Condition

value of 1.28 at the minimum casing pressure of 43 psia. The solid lines represent predicted values from the performance model.

More detailed discussions of the test results are presented in previous papers. 2,3

Endurance Test (10/28/82 to 4/3/83)

Following the performance testing, relatively long term, steady state operation of the RST was demonstrated. The endurance test was run to accumulate a total of 4011 hours of running time on the unit. The goals of the test were to determine availability of the RST during steady state operation, to evaluate the operability of the RST (ease of operation during start-ups and shutdowns, as well as steady state operation), and to get a preliminary evaluation of the durability of the RST components in the geothermal environment.

During the endurance testing, there were a total of 16 shutdowns. Of these, five were due to power outages and seven were due to miscellaneous site operations unrelated to the RST. Two system trips were caused by shorts in the load banks, due to blowing snow. One trip was the result of an unexplained generator field loss, and one outage due to a loss of line power was extended due to generator damage (attributed

to a lightning strike). The only outage related to the RST was an inspection lasting 57 hours after 2000 hours of running time. This inspection would not be part of an RST maintenance schedule. Thus, for the endurance test, the RST machine availability was 100%.

Operability of the RST proved to be excellent. Start-ups and shutdowns were easily accomplished, and steady state running of the RST was essentially "hands off".

After the 4000 hours, the RST was disassembled and shipped to Trenton for an inspection. The inspection report from the Trenton facility is included as Appendix A. There was no significant damage to any fluid-handling component due to the 4000 hours of operation. Bearings and seals were in good condition. The only significant problem noted was a build-up of silica scale on the outside of the rotary separator. As this scale flaked off irregularly (due to thermal stresses from the frequent start-ups), the vibration characteristics of the rotary separator changed. This scale build-up was attributed to liquid splash, caused by divergent nozzle-exit jets during off-design operation. Deflectors were designed to fit on the nozzle during subsequent runs. These deflectors would direct any divergent liquid droplets onto the rotary separator. In addition, a hydraulic cleaning system was implemented for periodic use in the future. The cleaning system will use high-pressure water to strip off any silica

build-up on the outside of the rotary separator, without need
of an RST teardown.

III. SEPARATED FEED CONFIGURATION

Optimization Study

Concurrent with the running of the endurance test was an optimization study, funded by Utah Power and Light. One goal of the study was to identify the RST power plant configuration that would lead to the minimum levelized busbar cost. The result of the study was a plant configuration referred to as the separated feed configuration. Figure III-1 shows the P&ID for the separated feed test system at Roosevelt Hot Springs. This test system is similar to a power plant, except that all steam is vented to atmosphere rather than expanded through a steam turbine.

The principle change is the addition of a flash tank upstream of the RST. Following separation of the wellflow, the high-pressure steam would be taken to the high-pressure port of a dual-admission steam turbine. The high-pressure liquid is taken to the RST. A flash is taken across the nozzle-throttle valves, to produce 1% steam at the nozzle inlets. During normal (clean well) operation, this set of conditions will result in RST operation at the design speeds.

As the well characteristic decays due to downhole scaling, it may become necessary to add a small fraction of the high-pressure steam to the nozzle inlet streams, to maintain

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Figure III-1. Upgrade Test (Separated Feed Configuration) P&ID

VALVE SUMMARY		
TITLE	DESCRIPTION	OPERATION
1	INLET VALVE	MANUAL
2	INLET VALVE	MANUAL
3	INLET VALVE	MANUAL
4	INLET VALVE	MANUAL
5	INLET VALVE	MANUAL
6	INLET VALVE	MANUAL
7	INLET VALVE	MANUAL
8	INLET VALVE	MANUAL
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95	INLET VALVE	MANUAL
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98	INLET VALVE	MANUAL
99	INLET VALVE	MANUAL
100	INLET VALVE	MANUAL

LEGEND - VALVES
 BUTTERFLY VALVE
 BALL VALVE
 HAND OPERATED GATE VALVE
 PNEUMATIC BALL VALVE
 GATE VALVE

LEGEND - INSTRUMENTATION
 DIFFERENTIAL PRESSURE
 CURRENT
 PRESSURE TRANSDUCER
 POWER
 POWER FACTOR
 REACTIVE POWER
 SPEED PICK-UP
 RESISTANCE TEMP DETECTOR
 THERMOCOUPLE
 VIBRATION (RADIAL)
 VALVE POSITION
 OIL COOLED BEARING
 REINJECTION PUMP
 PRODUCTION WELL
 VOLTAGE
 SHAFT POSITION

DATA SYSTEM SUMMARY			
STATION	DESCRIPTION	INPUT	ANAL
1	SYSTEM INLET TEMPERATURE	TYPE E	
2	SEPARATOR TANK TEMPERATURE	TYPE E	
3	LIQUID HEADER TEMPERATURE	TYPE E	
4			
5			
6	MIXER 1 INLET TEMPERATURE	TYPE E	
7	MIXER 2 INLET TEMPERATURE	TYPE E	
8	MIXER 3 INLET TEMPERATURE	TYPE E	
9	MIXER 4 INLET TEMPERATURE	TYPE E	
10	EST. STEAM OUTLET TEMPERATURE	TYPE E	
11	EST. STEAM OUTLET ANNULAR TEMPERATURE	TYPE E	
12			
13	EST. CASING TEMPERATURE	TYPE E	
14	CASE DRAIN TEMPERATURE	TYPE E	
15	EST. OILFIELD TEMPERATURE	TYPE E	
16	SECONDARY NOZZLE INLET TEMPERATURE	TYPE E	
17	ATMOSPHERIC TEMPERATURE	TYPE E	
18			
19			
20	SYSTEM INLET PRESSURE	0-500 PSIA 0-5VDC	
21	SEPARATOR TANK PRESSURE	0-500 PSIA 0-5VDC	
22			
23	LIQUID HEADER PRESSURE	0-500 PSIA 0-5VDC	
24	HIGH PRESSURE STEAM ANNULAR INLET	0-500 PSIA 0-5VDC	
25	MIXER 1 INLET PRESSURE	0-500 PSIA 0-5VDC	
26	MIXER 2 INLET PRESSURE	0-500 PSIA 0-5VDC	
27	MIXER 3 INLET PRESSURE	0-500 PSIA 0-5VDC	
28	MIXER 4 INLET PRESSURE	0-500 PSIA 0-5VDC	
29	MIXER 5 INLET PRESSURE	0-500 PSIA 0-5VDC	
30	EST. CASING PRESSURE	0-250 PSIA 0-5VDC	
31	EST. STEAM OUTLET ANNULAR PRESSURE	0-50 PSIA 0-5VDC	
32	MIXER 1 INLET PRESSURE	0-500 PSIA 0-5VDC	
33	EST. OILFIELD PRESSURE	0-500 PSIA 0-5VDC	
34	SECONDARY NOZZLE INLET PRESSURE	0-500 PSIA 0-5VDC	
35	PRIMARY OUTBOARD RADIAL VIBRATION - X	0-500 PSIA 0-5VDC	
36	PRIMARY OUTBOARD RADIAL VIBRATION - Y	0-500 PSIA 0-5VDC	
37	PRIMARY INBOARD RADIAL VIBRATION - X	0-500 PSIA 0-5VDC	
38	PRIMARY INBOARD RADIAL VIBRATION - Y	0-500 PSIA 0-5VDC	
39	U-TUBE OUTBOARD RADIAL VIBRATION - X	0-500 PSIA 0-5VDC	
40	U-TUBE OUTBOARD RADIAL VIBRATION - Y	0-500 PSIA 0-5VDC	
41	U-TUBE OUTBOARD RADIAL VIBRATION - X	0-500 PSIA 0-5VDC	
42	U-TUBE OUTBOARD RADIAL VIBRATION - Y	0-500 PSIA 0-5VDC	
43	SECONDARY OUTBOARD RADIAL VIBRATION - X	0-500 PSIA 0-5VDC	
44	SECONDARY OUTBOARD RADIAL VIBRATION - Y	0-500 PSIA 0-5VDC	
45	SECONDARY INBOARD RADIAL VIBRATION - X	0-500 PSIA 0-5VDC	
46	SECONDARY INBOARD RADIAL VIBRATION - Y	0-500 PSIA 0-5VDC	
47	PRIMARY ROTOR POSITION	0-500 PSIA 0-5VDC	
48	U-TUBE ROTOR SHAFT POSITION	0-500 PSIA 0-5VDC	
49	SEPARATOR LIQUID OUTLET AP	0-500 PSIA 0-5VDC	
50	EST. LIQUID INLET AP	0-500 PSIA 0-5VDC	
51	EST. LIQUID OUTLET AP	0-500 PSIA 0-5VDC	
52	EST. STEAM OUTLET AP	0-500 PSIA 0-5VDC	
53	STEAM TURBINE INLET FLOW AP	0-500 PSIA 0-5VDC	
54	EST. STEAM ENHANCEMENT ANNULAR AP	0-500 PSIA 0-5VDC	
55	CASE DRAIN LIQUID LEVEL	0-500 PSIA 0-5VDC	
56	PRIMARY ROTOR RPM	0-500 PSIA 0-5VDC	
57	U-TUBE ROTOR RPM	0-500 PSIA 0-5VDC	
58	SECONDARY ROTOR RPM	0-500 PSIA 0-5VDC	
59	PRIMARY INBOARD BOTTOM PAD TEMPERATURE	0-500 PSIA 0-5VDC	
60	PRIMARY INBOARD ADJACENT PAD TEMPERATURE	0-500 PSIA 0-5VDC	
61	PRIMARY OUTBOARD ADJACENT PAD TEMPERATURE	0-500 PSIA 0-5VDC	
62	1-1 66 GENERATOR - PHASE A CURRENT	0-500 PSIA 0-5VDC	
63	1-2 66 GENERATOR - PHASE B CURRENT	0-500 PSIA 0-5VDC	
64	1-3 66 GENERATOR - PHASE C CURRENT	0-500 PSIA 0-5VDC	
65	1-4 66 GENERATOR - PHASE A VOLTAGE	0-500 PSIA 0-5VDC	
66	1-5 66 GENERATOR - PHASE B VOLTAGE	0-500 PSIA 0-5VDC	
67	1-6 66 GENERATOR - PHASE C VOLTAGE	0-500 PSIA 0-5VDC	
68	1-7 66 GENERATOR - POWER	0-500 PSIA 0-5VDC	
69	1-8 66 GENERATOR - REACTIVE POWER	0-500 PSIA 0-5VDC	
70	1-9 66 GENERATOR - POWER FACTOR	0-500 PSIA 0-5VDC	
71	1-10 66 GENERATOR - PHASE A CURRENT	0-500 PSIA 0-5VDC	
72	1-11 66 GENERATOR - PHASE B CURRENT	0-500 PSIA 0-5VDC	
73	1-12 66 GENERATOR - PHASE C CURRENT	0-500 PSIA 0-5VDC	
74	1-13 66 GENERATOR - PHASE A VOLTAGE	0-500 PSIA 0-5VDC	
75	1-14 66 GENERATOR - PHASE B VOLTAGE	0-500 PSIA 0-5VDC	
76	1-15 66 GENERATOR - PHASE C VOLTAGE	0-500 PSIA 0-5VDC	
77	1-16 66 GENERATOR - POWER	0-500 PSIA 0-5VDC	
78	1-17 66 GENERATOR - REACTIVE POWER	0-500 PSIA 0-5VDC	
79	1-18 66 GENERATOR - POWER FACTOR	0-500 PSIA 0-5VDC	
80			
81	SYSTEM INLET VALVE POSITION	0-100% 4-20 mA	
82	SEPARATOR LIQUID OUTLET VALVE POSITION	0-100% 4-20 mA	
83	SEPARATOR STEAM OUTLET VALVE POSITION	0-100% 4-20 mA	
84	EST. LIQUID OUTLET VALVE POSITION	0-100% 4-20 mA	
85	EST. STEAM OUTLET VALVE POSITION	0-100% 4-20 mA	
86			
87	SEPARATOR LIQUID LEVEL	0-50 INCH 4-20 mA	

1	REVISED PER FIELD	3/9/88	MLC	
ZONE	CHG	ALTERATION	DATE	BY
CERTIFIED FOR CONSTRUCTION				
SEE DO-2001N FOR TOL'S ETC NOT SPECIFIED				
SEE DO-2001N FOR IDENTIFICATION MARKINGS				
Transamerica Delaval Refining Energy Systems				
TITLE: RST UPGRADE P & ID				
SCALE	1/4" = 1'	1ST ORDER		
DRAWN	1-87-84	CHECKED		
ENG. APP.	JR	SYMBOL		
ISSUED				
FEDERAL CODE IDENT. NO.	16712			

the desired operating speeds. This steam enhancement capability is shown on Figure III-1, although there will be no steam enhancement flow at the design point.

This configuration does not include a drag generator. The drag generator has been eliminated because it is desirable to have the entire system power output on a single shaft.

The steam output of the RST would be taken to the low-pressure port of the dual-admission steam turbine. The power-output shaft of the RST would be coupled to the steam turbine shaft through a gear box. The steam turbine shaft will directly drive the generator.

The liquid flow out of the RST through the diffuser would go to a liquid accumulator, where it mixes with any excess liquid produced by the well that cannot be used by the RST. Steam produced by flashing any high-pressure liquid would join the RST outlet steam, through a pressure-equalizing line. Present plans call for using an injection pump to inject the liquid output of the RST. Depending on the pressure requirement of the injection well, RST operating speeds could be adjusted to provide for injection of the RST liquid output without use of a pump.

Performance Comparison

In this section, a performance comparison for an RST system, an optimized two-stage flash system, and an optimized single-stage flash system is presented. The Roosevelt Hot Springs inlet conditions are used, but several simplifying assumptions are made and the results derived here are for comparison only. The conditions are summarized by Table III-1.

Assumptions made in the calculations are as follows:

1. For both the RST system and the two-stage flash system, the steam turbine efficiency, $\eta_{st} = 0.79$. For the single-stage flash system, $\eta_{st} = 0.80$. These numbers are inconsistent with Baumann's rule which is used in the data reduction. For specific power plants, interstage moisture removal can be employed to reduce the efficiency penalty resulting from increasing amounts of moisture in the latter stages. The dual admission turbine efficiency is penalized one percentage point to account for losses resulting from the admission of the lower pressure steam.
2. Non-condensable content of the well flow is zero. Condensing pressure of 2 psia is assumed.

Table III-1. Process Conditions for Power
System Performance Comparison

Inlet Conditions:	Flowrate = 1,000,000 lbm/hr Enthalpy = 472 Btu/lbm Pressure = 390 psia at wellhead
Single-stage flash:	Intermediate pressure = 53 psia
Two-stage flash:	High-pressure inlet = 113 psia Low-pressure inlet = 21 psia
RST system:	High-pressure steam turbine Inlet = 280 psia RST inlet = 255 psia Low-pressure steam turbine Inlet = 42 psia
Condensing Conditions:	Condensing pressure = 2 psia zero non-condensable gas content

3. Major parasitics included were reinjection pumping power and cooling water pumping power.
4. RST efficiency is assumed to be 36%. This is consistent with measured results from the performance test.

The calculations are summarized in Table III-2. With the assumptions made, an RST power plant would produce a net shaft power output of 14.66 MW, versus 11.20 MW for an optimized single-stage flash plant and 13.89 MW for an optimized two-stage flash plant. This results in a performance advantage of 30.9% for an RST system over a single-stage flash, and 5.5% over a two-stage flash plant.

The economic advantage of the separated-feed RST configuration is comprised of more than just the performance advantage.

Other contributing factors include:

1. Short lead time until plant operation.
2. Shop-fabricated skids, instead of expensive field erection.
3. Less heat rejection to atmosphere than the two-stage flash plant because of lower steam turbine flow rate (lower cooling water requirements).

IV. SYSTEM MODIFICATIONS AND SHAKEDOWN TEST (2/29/84)

RST Modification

As a result of the design study, it was necessary to test the RST in a separated feed configuration. System modifications are discussed below. The nozzle design points, for the original RST nozzles (tested to 4/83) and the Upgrade Test nozzles are summarized below in Table IV-1. Due to the lower inlet energy for the upgrade test, the nozzle-exit velocity, V_b , for the Upgrade Test nozzles was much lower, 720 vs. 1134 ft/s. This eliminated any significant risk of RST overspeed, thus eliminating the need for a drag generator. In addition, the lower inlet pressure and steam fraction made it possible to fit higher flow-capacity nozzles in the RST. This was needed to keep the design RST outlet power at a reasonable level (1.2 MW), and allow one RST to handle the full flow from one typical Roosevelt Hot Springs well.

As indicated by Table IV-1, the Upgrade Test nozzles could pass more flow, raising the design flow rate through the RST from 540 klbm/hr to 894 klbm/hr. To accommodate this, new U-tube scoops were designed, stress analyzed using a three-dimensional finite element model, cast and finish machined. Also, a new, larger diffuser was designed and fabricated.

During the performance testing, the tube matrix which had been installed on the rotary separator to prevent hydrodynamic instabilities proved to be susceptible to scale build-up inside the tubes. Although this had not been a problem during the 4000 hours of operation, long term running (>1 year) would require elimination of such stagnant cavities by conversion to a non-symmetric configuration similar to the configuration that had been tested successfully at the Biphasic laboratory. The tube matrix was removed, and a solid ring-shape section was added to the nozzle side of the rotary separator. If successful, this rotor configuration would make possible one-piece fabrication of the rotary separator, as well as eliminate the potential scale hazard presented by the tube matrix. In addition, the tube matrix was removed from the U-tube side of the rotary separator, to allow for the new, larger scoops.

Piping Changes

After reviewing existing technology for geothermal wellhead flash tanks, we selected a vessel built by WKM. The tank was a vertical separator, with an overall height of 27 feet (from the ground to the top of the steam exhaust flange), and a 48-inch diameter.

Due to the vertical height of the liquid in the flash tank, and to the partial pressure of non-condensable gases in the flash tank, single phase liquid between the flash tank and the nozzle-throttle valves was predicted. This simplified the problem of spitting the flow evenly between the four nozzles. The flow splitter that had been used during the performance tests was removed and replaced with a simple distribution manifold.

Additional piping changes were:

1. Rework of the supply piping from the wellhead to accommodate the flash tank.
2. Rework of the nozzle-inlet piping to reflect the replacement of the flow splitter with the manifold.
3. Addition of steam enhancement piping. This consisted of a 4-inch line from the flash tank to a steam manifold located above the RST, and individual 2-inch lines to each nozzle.
4. Replacement of existing 6-inch diffuser-outlet line with 8-inch line. The new higher flow rates had made the pressure drop in the 6-inch line prohibitive.
5. Replacement of existing RST steam-outlet piping with larger line. The existing piping had placed a lower limit on RST casing pressure that was undesirable.
6. Elimination of the existing bypass piping, dump valve piping, accumulator tank and pump.

7. Piping of the flash tank outlet steam to the Phillips holding pond. This flow was not combined with the RST outlet steam.

The piping arrangement for the Upgrade Test is shown schematically by Figure III-1, the RST Upgrade P&ID.

The new piping arrangement, with the steam enhancement provision, generated the question of how to best mix the separate liquid and steam streams. Plans were made to try three different methods. First, a set of four tube-bundle mixers were designed and built. These mixers consisted of a group of 8-inch long 1/4-inch stainless steel tubes, in a hexagonal pattern as viewed from the end. The brine would flow inside the tubes. The steam would be admitted radially inward to the tubes. The mixer geometry was such that the steam would accelerate as it flowed along the outside of the tubes. At the design condition, the velocity of the steam leaving the mixer would be greater than the liquid velocity. The resulting shear was expected to break up the liquid jets into small droplets, which would enhance nozzle performance. Also, since each liquid jet was surrounded by steam, liquid would not be allowed to build up on the walls of the nozzle.

Plans were also made to run with flow mixers manufactured by Cleveland (hollow tubes with two twisted metal inserts), and with just a hollow sleeve.

Control and Instrumentation Modifications

Instrumentation is shown on Figure III-1. The main changes made were:

1. Replacement of all orifice plates with Annubar flow-averaging Pitot tubes for six flow metering locations. The Annubars, if reliable, would reduce permanent pressure losses due to flow metering, and would also reduce straight-leg piping requirements.
2. The flash tank was fully instrumented, for pressure, temperature, and liquid level.

In addition, five new control loops were added. The PID calculations were done on the EPTAK 240 controller, by Eagle Signal. Excess capacity was added to the controller to do mass flow calculations for the digital displays. The control loops are summarized below.

1. Flash tank pressure would be controlled by an upstream throttle valve (FTITV on Figure III-1). If tank pressure dropped below set point, the valve would open to admit more flow. Control on this valve was intended to be sluggish, to avoid having FTITV chasing fluctuations resulting from small changes downstream.
2. Liquid level in the flash tank would be controlled by the bypass valve, FTLOTV.
3. An alternate method of controlling liquid level would be with the steam enhancement throttle valve, RSITV. If level were to raise above the set point, the controller would respond by closing the valve. This would result in less steam to the nozzles, which would increase the liquid flow rate and decrease the tank level. By setting set point and gain such that the proportional bands for FTLOTV and RSITV did not overlap, we would be able to operate with both loops active simultaneously. If RST capacity were greater than the well production rate, FTLOTV would be fully closed and level would be controlled by throttling steam enhancement. If RST capacity were less than the production rate, RSITV would remain at some predetermined setting and FTLOTV would control the tank level.

4. Rotary separator speed would be continuously monitored. If the speed exceeded a pre-determined value, the RST steam-outlet throttle valve (RSOTV) would close, increasing casing pressure and reducing speed.
5. FTSOTV would control steam flow rate in event of an RST trip.

The Upgrade Test arrangement placed a limit on the testing. The nozzles used were designed for a power plant that had the liquid turbine rotating at 1400 rpm. With the test system, the only way to operate at full flow power levels was to synchronize to the grid, which constrains the liquid turbine to rotate at 1800 rpm. Thus, to operate at the design speeds and design liquid flow rate, we would have to reduce casing pressure significantly below the design value of 42 psia.

Shakedown Test (2/29/84)

The test system was run for an 8-hour period. Steady flow through the flash tank was demonstrated, with automatic pressure and level control. During start-up of the RST, injection pressure was much higher than expected (120 psig vs. 30 psig), resulting in no liquid flow through the diffuser. All liquid left the casing through the case drain, after spilling. Because of the additional windage losses associated

with the spillage, rotor speeds required for synchronization were never reached. A maximum rotary separator speed of $N_s = 2100$ rpm was obtained.

Conclusions and Modifications

Because of unexpectedly high injection pressure requirements, we were unable to reach acceptable operating speeds with the RST. To solve this problem, a bypass line with a manual block valve (RL0BV2 on Figure III-1) was installed to direct the diffuser flow into the flash tank steam outlet leg downstream of the steam outlet throttle valve (pressure equals atmospheric plus line losses of approximately 250 feet of piping). We intended to use this bypass leg during start-ups, then switch over to the injection well after reaching acceptable operating speeds. Also, a bypass leg with a manual valve (FTS0TV2) was installed to use in place of the pneumatic valve FTS0TV1 at low-flow conditions. The pneumatic valve operation had been erratic at low flow conditions.

V. UPGRADE TEST SERIES #1 (3/22 - 3-23/84)

Test Account System operation for 3/22 and 3/23/84 is summarized below:

3/22/84   1430: Started to flow well at low flow rate (200 klbm/hr) to warm up injection line.

             1630: Injection line warm-started to increase flow.

             1900: Started rotation of secondary rotor. We saw excessive low frequency vibration on both secondary rotor and rotary separator channels. Shut down to investigate. Decided signals were probably real, but not hazardous.

             2150: Re-started secondary rotor.

             2200: Re-started rotary separator. Liquid out of diffuser was bypassed through RL0BV2, into the low-pressure portion of the flash tank steam outlet line.

             2248: Reached a condition with $N_S = 2080$ and $N_U = 1700$ rpm, and prepared to load liquid turbine. At this point, most liquid was still leaving the RST casing through the case drain.

2310: While loading liquid turbine, at a condition with $N_s = 2020$ and $N_u = 1430$ ($U/C = 0.71$), we encountered high vibrations on the rotary separator. Vibration levels were approximately 8 mils ptp, and vibrations were actually heard and felt in the control trailer. We were able to exit the vibration mode immediately by unloading the liquid turbine which raised the value of U/C .

2350: Following several unsuccessful attempts to pass through the vibrations, we shut down for the night, still bypassing well flow through the flash tank.

3/23/84 0820: Returned to site. Well had been shut in at 0700. Re-started the well.

0930: Flow initiated through the flash tank. Loud banging noises were coming from the injection line, due to non-condensable gases trapped in a dead headed portion of the injection line. Banging stopped when Phillips bled the gas through a vent valve.

1425: Started rotation of the rotary separator. We encountered the same vibrations as we had the previous night. The vibrations occur at the transition point when flow begins to transfer out of the RST through the

diffuser. This corresponds to going from the spill to the no-spill regime in the RST. The vibrations were on the rotary separator at the liquid turbine synchronous speed.

1715: We brought on a second nozzle partially and were able to synchronize to the grid, achieving a condition of $N_s = 2340$ and $N_u = 1800$ rpm. We tried to achieve complete liquid transfer by increasing the flow to the second nozzle and encountered the excessive vibrations at $N_s = 2530$ rpm ($U/C = 0.71$).

1900: Shut down system.

Conclusions and Modifications

We felt that the vibration problems were the result of trying to pass from the spill to the no-spill regime at fairly high energy levels ($N_s > 2000$ rpm). This was necessary because the generator exciter worked in an automatic mode and would not begin to operate until a condition of $N_u > 1400$ rpm was reached. The corrective action chosen was to modify the excitation system to allow excitation of the generator in a manual mode. This would allow operation at the desired speed ratio of $U/C \approx 0.60$ as the RST came up to operating speed, passing from the spill to the no-spill regime at much lower

energy levels. A second 400 kW load bank (previously used with the drag generator) was connected to give twice the load capacity.

VI. UPGRADE TEST SERIES #2 (4/2 - 4/3/84)

Test Account System operation for 4/2 and 4/3/84 is summarized below:

4/2/84

Prior to starting the test, we ran vibration surveys with the main lube-oil pump, the auxiliary lube-oil pump, and the cooling fan. No low-frequency noise was seen.

1345: Injection well warm-up was completed.

1512: Rotation of rotary separator.

1600: Demonstrated ability to come up in speed, by maintaining $U/C < 0.60$ using manual generator excitation. With $N_s = 2300$ rpm, we lost control momentarily, allowing N_u to rise, and vibrations began. We recovered by closing the nozzle throttle valve, stopping the liquid flow to the RST. The U/C display was erratic, making it difficult to maintain the proper speed ratio. Also, as we approach $N_u = 1800$ rpm, the required load is nearly 300 kW. Due to the discrete nature of the load banks, this forces us to control with 50 or 100 kW increments. To synchronize easily, we must reduce casing pressure and optimize steam enhancement to raise the required load to 310 kW. This

will allow us to control with 5 or 10 kW increments.

1940: On the third start-up, we synchronized to the grid.

2020: Brought on a second nozzle.

2110: Brought on the third nozzle.

2200: Brought on the fourth nozzle. The observed increase in power level was from 0.95 to 1.12 MWe at equal steam enhancement fractions, indicating partial plugging of the fourth nozzle.

2355: Completed diffuser pressure recovery survey. With secondary rotor speed of 900 rpm, we saw a pressure increase of 130 psi, up to 140 psig at the diffuser outlet.

4/3/84 0044: Shut down test system.

Conclusions and Modifications

Following the test series, the tube mixers were removed and inspected. The mixer for nozzle #3 (the fourth nozzle brought on) was almost completely plugged at the inlet, with pipe scale. The other three mixers were still mostly open with perhaps 10% of the tubes plugged. The major problem with the tube mixers was excessive pressure drop across the tubes. The nozzles were designed for 255 psia inlet flow (25 psi drop

across the throttle valve and mixer). We were experiencing pressure drops up to 75 psi across the tube mixers. This restricted flow capacity to a maximum of 560 klbm/hr through the RST. Data is presented graphically in the analysis section (Section VIII). To increase the flow capacity, we replaced the tube mixers with the Cleveland mixers.

Throughout the test run, low-frequency noise on the vibration monitors was observed. The expansion joints on the individual nozzle lines were not "breaking away" to provide thermal expansion compensation, and the resultant force was being transmitted to the RST casing. To remedy this, an expansion loop was installed in the main RST liquid feed line.

VII. UPGRADE TEST SERIES #3 (6/4 - 6/6/84)

Test Account System Operation for 6/4 to 6/6/84 is summarized below:

6/4/84   1440: Ran a trial start-up of the rotary separator to measure breakaway torque. Rotation began with flash tank pressure = 125 psig, 60 psig at nozzle inlet, and 40% open on valve RSITV.

1515: With RST isolated, increased flash tank pressure to 280 psia.

1541: Rotation of rotary separator.

1700: Synchronized the liquid turbine generator to the grid. The generator was being excited in the manual mode in order to maintain U/C < 0.60 during the start-up.

1825: Completed a survey on the effect of steam enhancement with one nozzle. With the nozzle liquid throttle valve wide open (272 psig at mixer inlet), steam enhancement flow was 0.0 lbm/s, $N_s = 2970$ rpm and the power output was 330 kWe. We closed the liquid throttle valve (N2LTV), achieving a 14 psi drop across the valve. Steam enhancement flow rate increased to 4000 lbm/hr (about 2%

of the inlet flow). This had the effect of increasing nozzle spouting velocity, evidenced by an increase to $N_s = 3170$ rpm, but also reducing the power level to 285 kWe due to a decrease in liquid flow into the nozzle.

At this condition, with casing pressure $P_{10} = 2.0$ psig, case drain flow rate was 15 gpm ($= 6500$ lbm/hr). This high flow rate was due to the severe off-design operating condition, as the design nozzle exit pressure is 30 psig.

1900: Raised casing pressure to $P_{10} = 10$ psig, to observe the effect on case drain flow rate. We measured a flow rate of 3.75 gpm (1700 lbm/hr, less than 1% of the total flow rate). This indicates strongly that case drain flow rate will be acceptable when the design conditions are reached.

1942: Completed a survey of re-injection capability with one nozzle. The maximum pressure attained at the diffuser outlet was 36 psig (30 psi recovered by the diffuser). When the diffuser throttle valve (RL0TV) was closed further to increase the required

pressure recovery, we saw significant liquid spillage off of the secondary rotor, resulting in a slowdown of both the rotary separator and the secondary rotor. We recovered by opening RL0TV.

At this point, the RST liquid-outlet flow was going into the flash tank steam outlet line rather than the injection well. The liquid bypass from the flash tank was going to the injection well 82-33. The liquid bypass flow of ~200 klbm/hr required 100 psig to push it through the long (2 mile) injection line. We felt that this pressure requirement was largely due to the two-phase flow in the line resulting from the 170 psi drop across FTL0TV.

1945: Brought on the second nozzle (#4). At this point, the well flow and the RST capacity were pretty well matched and liquid bypass dropped to a low rate. Injection pressure dropped to 35 psig.

2100: Completed a diffuser pressure-recovery survey, achieving a maximum outlet pressure of 61 psig (a 52 psi increase across the diffuser with a secondary rotor speed of 540 rpm).

2120: Increased the well flow rate by opening FTS0TV2. This reduced the resistance to the steam flow out of the flash tank, which reduced tank pressure. Sensing this, the flash tank pressure controller opened FTITV, which increased the flow rate from the well.

2130: Brought on the third nozzle (#1). With three nozzles, we opened the nozzle throttle valves from 50% to 80%. This reduced the steam enhancement flow rate from 2.5% to 1.0% of the total flow, which increased power output from 1000 to 1170 kWe.

2210: Completed a pressure recovery survey, obtaining a maximum pressure of 81 psig (a 72 psi increase) with $N_{SS} = 560$ rpm.

2330: Brought on the fourth nozzle (#3).

6/5/84

0030: Completed a survey of the effect of steam enhancement. With minimum steam enhancement of 1.3% of the total flow, we reached a

maximum power level of 1430 kWe. At this point, $N_s = 2975$ rpm, for $U/C = 0.605$. At this late hour, we were reluctant to reduce speed any more (by further reducing steam enhancement), so we ended the survey and began to prepare for night shift operation at steady state.

0040: Returned to three nozzle operation. After adjusting the well flow for minimum bypass, we switched the diffuser outlet flow from the steam line to the injection well by opening RL0BV1 and closing RL0BV2. Injection pressure steadied out at 26 psig. Operation was steady state for the next 9 hours.

1106: The pneumatic flash tank steam-outlet throttle valve, FTS0TV1, opened unexpectedly resulting in a drop in tank pressure. We tripped the system to avoid raising U/C unacceptably high and encountering vibrations (excessive flow spillage would have led to a shutdown anyway). Upon inspection, we found an air leak in the line to the outside valves. Since FTS0TV1 was

the last valve on the air line, it was the one to fail. Repaired the leak and prepared for a re-start.

1446: Rotation of rotary separator.

1505: With one-nozzle operation, we synchronized to the grid. At this time, diffuser flow was into the steam line. During previous operation, we had noted that time transients of the injection line were slow, i.e. a reduction in bypass liquid would cause a drop in injection pressure, but it may take 30 minutes for the injection pressure to approach its steady state value.

1602: Opened RL0BV1 to allow the built-up pressure in the injection line to vent into the steam line. Over the next 15 minutes, injection pressure dropped from 38 to 7 psig.

1620: Closed RL0BV2 directing all liquid flow into the injection line. Our intention was to bring on additional nozzles while putting all liquid into the injection well. For the next four hours we monitored injection pressure, which initially climbed to 40 psig, and finally steadied out at 38 psig,

even though the flash tank liquid bypass was valve leakage only (FTLQTV was closed).

2010: Decided that bringing on a second nozzle while the RST liquid outlet was going to the injection well was impractical. To bring on a second nozzle, we would need to increase the well flow rate, which would have resulted in significant two-phase flow to 82-33 for a short time. This would have driven injection pressure to a prohibitive level.

2030: Switched the diffuser outlet flow back into the steam line. Tried to bring on a second nozzle while simultaneously increasing the well flow to minimize two-phase flow to the injection well. While doing so, flash tank pressure dropped temporarily while FTITV responded to the increased flow demand. This resulted in the slowdown of the rotary separator which led to spillage, and eventually to high vibrations. At this time, we shut off liquid flow to the RST.

2200: Shut-in of the well was completed. This would allow the injection well to cool down and empty for the next day's run.

6/6/84

1020: Began to flow the well.

1220: Began to turn the rotary separator with diffuser flow into the steam line.

1308: Synchronized to the grid. With minimum liquid bypass from the flash tank, injection pressure was 35 psig.

1310: To prepare for bringing on the second nozzle, we increased the well flow rate, bypassing ~200 klbm/hr for a short time. Injection pressure rose to 108 psig.

1325: Brought on the second nozzle (#4). Within 10 minutes, injection pressure had dropped to 50 psig with minimum liquid bypass.

1340: Boosted the well flow rate to accommodate three-nozzle operation resulting in two-phase flow to the injection well for a short period of time. Injection pressure rose to 108 psig. Brought on the third nozzle (#1).

1420: Transferred the diffuser-outlet flow from the steam line into the injection line. Our

intention was to bring on the fourth nozzle while injecting to 82-33. Determined an acceptable lower limit on the nozzle throttle valve to be about 45% open. Next, opened the nozzle throttle valves completely to maximize the liquid flow per nozzle, and increased the well flow accordingly.

Brought on the fourth nozzle while simultaneously closing the other three nozzle throttle valves to match the four nozzle flow rate with high steam enhancement to the three nozzle flow rate with low steam enhancement. Using this procedure, were able to bring on the fourth nozzle without bypassing significant liquid.

1940: Completed a survey of the effect of steam enhancement during four-nozzle operation. Maximum power observed was 1492 kWe, seen with a steam enhancement rate of 0.7% of the total flow, which was 840 klbm/hr. A maximum flow rate of 860 klbm/hr was measured with minimum steam enhancement of 0.5%. At this point, $N_s = 2956$ rpm ($U/C = 0.609$) and we decided not to reduce steam

enhancement further. With casing pressure of 27 psia, a case drain flow rate of 6 gpm (=2700 lbm/hr) was measured.

2150: Completed a survey of casing pressure by increasing to 30 psia. This reduced N_s to 2902 rpm ($U/C = 0.620$), and reduced the case drain flow rate to 2.1 gpm (=950 lbm/hr). When attempting to increase casing pressure further, encountered severe vibrations at $N_s = 2890$ rpm ($U/C = 0.623$). The vibration signature indicated the presence of a traveling wave on the rotary separator, with the dominant spike on the spectrum analyzer at 90% of synchronous speed. We exited the vibration mode by reducing casing pressure, allowing the rotary separator to accelerate. Because of the constraint on the liquid turbine speed to turn at 1800 rpm, we were unable to achieve the design back pressure of 42 psia on the nozzles.

2215: Completed a pressure recovery survey. Maximum diffuser outlet pressure of 116 psig (a 100 psi increase) was seen with $N_{ss} = 640$ rpm.

2245: Cut back to three nozzle flow, still feeding the diffuser outlet flow to the injection

line. While trying to match the well flow rate to the RST capacity, the flash tank ran dry. This resulted in a slow down of the rotary separator. In addition, the secondary rotor accelerated up to a speed faster than the velocity of the U-tube exit jet. This indicated a coupling to the secondary rotor by either the U-tubes themselves or by flow spilling off of the rotary separator. This unusual operating mode was ended by increasing flash tank pressure and reducing the RST casing pressure, to accelerate the rotary separator.

6/7/84   0030: De-synchronized and completed an orderly shut down of the RST, followed by shutting in the well.

VIII. ANALYSIS

Test Series #2 (4/2 - 4/3/84)

Much of this test series was devoted to determining the impact on RST performance of varying steam enhancement. Post-test data analysis proved inconclusive because of the excessive pressure drop across the tube mixer. Figure VIII-1 presents the results of a steam enhancement survey done at the four-nozzle condition. The top graph, rotary separator speed vs. steam enhancement fraction, shows speed to drop as steam enhancement fraction is reduced. This is as expected because the nozzle spouting velocity will decrease as inlet steam fraction is reduced.

The middle section of Figure VIII-1 shows power output vs. steam enhancement fraction. As steam fraction is reduced from 4.4% to 2.3%, power output drops from 1120 kWe to 980 kWe. This effect was not expected. As nozzle inlet steam fraction increases with constant pressure, nozzle spouting velocity increases but liquid flow rate decreases. These two effects offset each other and we would expect a flat curve for power vs. steam fraction.

The lower section of Figure VIII-1 shows the reason for this behavior. The pressure drop across the mixers was so high (50

Figure VIII-1. Effect of Steam Enhancement (4/3/84).

to 100 psi) that when steam enhancement flow was reduced, the new operating point for the nozzle was at a dramatically lower pressure but at roughly the same flow rate. Thus, nozzle spouting velocity decreased, but there was no increase in mass flow to compensate. This will not be the case when the flow is controlled only by the nozzle (low mixer ΔP).

As a result of the excessive mixer ΔP , we were unable to approach the design flow rate of 894 klbm/hr. RST flow rate and power output as a function of number of nozzles are shown on Figure VIII-2. Flowrate is expected to vary linearly with the number of nozzles, and power should increase almost linearly (windage and bearing losses are independent of flow rate). This is different from the performance test data taken prior to 4/83 because operation of the flash tank at constant pressure eliminates the effect of well drawdown.

This linear behavior of both flow rate and power level is shown clearly on Figure VIII-2 for 1, 2, and 3 nozzle operation. The fourth nozzle did not have full effect because the upstream mixer was plugged with debris. Because of the large pressure drop across the mixer, and the susceptibility of the tube mixers to plugging, we replaced them with the Cleveland mixers for the third test series (6/4 - 6/6/84).

Figure VIII-2. RST Power, Flowrate vs Number of Nozzles

Analysis of Typical 3-Nozzle Data Point (A-80 on 4/3/83)

Theoretical: Assuming an isenthalpic expansion from 293 to 187 psia, we can calculate:

Nozzle-inlet steam quality, $X_{N1} = 0.050$

Predicted nozzle efficiency, $\eta_N = 0.82$

(for an optimized nozzle)

Bulk velocity, $V_b = 897 \text{ ft/s}$

Liquid velocity, $V_L = 885 \text{ ft/s}$

Tangential component = $V_{L-\tan} = 766 \text{ ft/s}$

Figure VIII-3. RST Flow Capacity vs Steam Enhancement Percent

Conditions at nozzle inlet:

P = 260	P = 258	P = 253	P = 245
X = 1.5%	X = 2.1%	X = 2.8%	X = 3.5%

Figure VIII-4. RST Power Output vs Steam Enhancement Fraction

Solve for predicted rotary separator speed, V_s (Equation 1):

$$\frac{1}{2} \dot{M}_L V_{L-\tan}^2 = \frac{1}{2} \dot{m}_L V_s^2 + w + b \quad (1)$$

where

b = bearing losses

w = windage losses

Using predicted curves for losses, and solving equation 1 for separator velocity:

$$V_s = 743 \text{ ft/s, which would yield } N_s = 3154 \text{ rpm}$$

The measured valve was 2895 rpm. There are two reasons for the low measured value of N_s . First, the nozzle was operating at conditions significantly off design, which should reduce nozzle efficiency. Second, because the casing pressure was so far off design (25 vs 42 psia), the nozzle-exit jet was under-expanded. This results in a divergent nozzle-exit jet,, which causes splash resulting in additional windage.

If all of the speed reduction were due to reduced nozzle efficiency, we can imply a nozzle efficiency of $\eta_N = 0.69$.

If all of the speed reduction is the result of additional windage losses, this would indicate total windage and bearing

losses of 302 hp, versus the predicted value of 78 hp. We can conclude that RST performance was reduced due to operating conditions very far off design.

Now to examine liquid turbine performance:

Speed of liquid on rotary separator = $V_S = 682.1 \text{ ft/s}$

Speed of liquid-turbine rotor = $V_U = 424.1 \text{ ft/s}$

If liquid turbine operation were 100% efficient, the secondary rotor would be driven at a speed of $V_{SS} = 166.1 \text{ ft/s}$ (= 705 rpm), and the shaft power output would be 811 kW.

Electrical power measured at 610 kWe implies a generator efficiency of $\eta_{GE} = 93.1\%$, which yields a shaft power output of 655 kW. Thus, liquid-turbine efficiency was measured at 81% which is as predicted.

Test Series #3 (6/4 to 6/6/84)

By replacing the tube bundle mixers with the Cleveland twisted-insert type mixers, we reduced the pressure drop measured across the mixers from 50-100 psi to 10-20 psi. This enabled us to reach lower nozzle-inlet steam fractions, and obtain higher flow rates through the RST. This effect is shown by Figure VIII-3 which is a plot of RST flow rate and well flow rate versus steam enhancement fraction. For these

data points, well flow rate was matched to RST flow rate with minimal liquid bypass (leakage across the valve which, due to the flow ring construction, was approximately 50 klbm/hr). Maximum flow rate through the RST was 860 klbm/hr. At this point, the steam enhancement rate was 0.5% of the total flow, and nozzle inlet quality was 1.2%. This flow, added to the high pressure steam that was vented to atmosphere and the liquid bypass, totaled to a well flow rate of 1 Mlbm/hr.

The effect of steam enhancement is shown by Figure VIII-4. These data points were taken with the RST steam outlet throttle valve (RSOTV) fully open. As steam enhancement was reduced, total flow rate increased (Figure VIII-4), which resulted in an increase in the RST steam outlet flow rate. As steam flow rate increased, RST casing pressure increased slightly. Casing pressure varied between 28 and 29 psia over the survey. RST power output was multiplied by a factor which accounted for the difference in ideal energy (isentropic enthalpy change) resulting from the slight variation in casing pressure.

The lower horizontal scale is steam enhancement percent which is defined as the flow rate of the steam added through the steam enhancement line divided by the total flow rate through the RST, times 100%. The upper horizontal scale shows the conditions at the nozzle inlet. This accounts for steam

generation as the liquid is flashed across the throttle valve and as the two-phase mixture goes through the mixer. The minimum steam enhancement point (0.5%) corresponds to a nozzle-inlet steam fraction of 1.5% at 260 psia. At this point, RST power output reaches its maximum value of 1490 kWe.

The slope of the defined curve is quite steep at higher steam enhancement rates, and gets more shallow at the lower steam enhancement rates. As steam enhancement is decreased, flow rate increases which increases power level and nozzle spouting velocity decreases, which decreases power level. At the higher rates of steam enhancement, the higher mass flow effect dominates. As lower values of steam enhancement, the effect of reduced nozzle spouting velocity becomes more evident, to the point where the power output would be expected to reach a maximum at approximately 1% nozzle inlet steam.

The reduction in nozzle spouting velocity with reduced steam enhancement is caused by two effects. First, as steam enhancement is reduced, the ideal energy available (BTU/lbm) decreases, resulting in a lower spouting velocity if we assume constant nozzle efficiency. Second, as the nozzle inlet steam fraction is reduced, we expect nozzle efficiency to decrease also. Due to the off-design operation of the nozzle, this effect is impossible to quantify.

Table VIII-1. Survey of Maximum Diffuser
Outlet Pressure

NO. of NOZZLES	P ₁₀ psig	P ₁₅ psig	N _{ss} rpm	$\eta_{diff} (= \frac{P_{15} - P_{10}}{\frac{1}{2} \rho v^2_{ss}})$
1	6.7	36.4	581	0.25
2	8.9	61.0	531	0.53
3	9.0	81.0	560	0.66
4	15.6	115.9	643	0.70

Diffuser Pressure Recovery

A survey was run to determine maximum re-injection pressure available as a function of flow rate. This test was conducted under the sponsorship of the Department of Energy. The results are summarized below, in Table VIII-1. To determine the maximum pressure attainable, the RST liquid-outlet throttle valve (RLQTV) was slowly closed, until liquid spill from the secondary rotor was observed (indicated by cycling of the case drain throttle valve). At this point, the valve was opened slightly to regain stable operation (no spill) at the maximum repressurization point. Due to the discrete nature of valve operation, the maximum pressure attainable may actually be slightly higher than measured.

Maximum pressure recovery was 100 psi during 4-nozzle operation. Diffuser efficiency at this point was 70%. As expected, diffuser performance improved as liquid flow increased (inlet void fraction decreased). This effect is shown on Figure VIII-5.

Steam samples were taken from the RST outlet line during 3- and 4-nozzle operation. The results are summarized below, in Table VIII-2. The steam qualities measured are poor, but the strong upward trend with increasing casing pressure is encouraging. During 4-nozzle operation, steam quality

Table. VIII-2. Summary of RST Outlet
Quality Measurements

NO. of NOZZLES	P ₁₀ psia	RST OUTLET STEAM QUALITY
3	23.5	0.65
3	28.9	0.78
4	27.2	0.58
4	29.6	0.66
4	30.1	0.73

Figure VIII-5. Diffuser Efficiency vs Inlet Void Fraction

increased from 58% to 73% as casing pressure was increased from 27.2 to 30.1 psia. The design casing pressure is 42 psia. This indicates that nozzle under-expansion is resulting in spray either deflecting off of the nozzle separator, or missing the rotary separator altogether. The nozzle extensions added to direct spray onto the rotary separator were not sufficient.

During the previous Performance Test (10/82), RST outlet steam quality of 95% was measured during 4-nozzle operation with a total RST flow rate of 540 klbm/hr. This was with RST casing pressure of 43 psia, versus the design casing pressure of 50 psia. This indicates that the steam quality problems will be largely eliminated as the design casing pressure is approached, but further effort is required to improve the steam quality during off-design operation.

Analysis of Maximum Power Point (Scan MM-34)

The critical measured parameters needed for performance analysis are summarized below:

Flash tank pressure = 283 psia

Nozzle inlet pressure = 258 psia

Steam enhancement flow rate = 1.4 lbm/s (5040 lbm/hr)

RST total flow rate = 236 lbm/s (850 klbm/hr)

RST casing pressure = P_{10} = 28.6 psia

Rotary separator speed = N_s = 2964 rpm

Generator power output = 1491 kWe

Secondary rotor speed = $N_{ss} = 642$ rpm

Assuming an isenthalpic expansion from 283 to 258 psia, we can calculate the nozzle-inlet steam quality, $X_{NI} = 0.017$.

Now, assuming an isentropic expansion to 28.6 psia, and a nozzle efficiency $\eta_N = 0.80$, we can calculate predicted nozzle exit velocities:

Bulk Velocity $V_b = 873$ ft/s

Liquid Velocity $V_L = 848$ ft/s

Tangential Component = $V_{L-tan} = 737$ ft/s

We can solve for predicted rotary separator speed, V_s , using predicted values for windage and bearing losses. We get $V_s = 721$ ft/s, which yields $N_s = 3059$ rpm.

The measured value was $N_s = 2964$ rpm. As discussed earlier, the decreased operating speed results from a combination of additional windage and decreased nozzle efficiency, both resulting from nozzle under-expansion. If we assume $\eta_N = 0.80$, the additional windage required to slow the rotary separator to 2964 rpm would be 179 hp, for total windage and bearing losses of 261 hp. If we assume the predicted value of 82 hp for windage and bearing losses (not the case based on

the steam quality measurements, we can place a lower bound on nozzle efficiency of $\eta_N = 0.75$.

If the liquid turbine were 100% efficient, the available shaft power would be 1921 kWe. The measured power level indicates a generator efficiency of 0.953, which implies a liquid turbine efficiency, $\eta_{LT} = 0.81$, which is as predicted. RST machine efficiency at this point was 33.1%.

Conclusions (6/4 - 6/6/83 Testing)

Milestones and Conclusions from the third test series are listed below:

1. High-flow (850 klbm/hr) operation of the RST was demonstrated. The higher flow rate was made possible by the installation of the lower pressure drop Cleveland mixers. The acceptable speed range of the rotary separator appears to be limited on the low side by hydraulically-induced vibrations (at $N_S = 2890$ rpm during 4-nozzle operation). This limit is well below the normal operation speed of the rotary separator.
2. Nozzle performance is near predicted ($\eta_N > 0.75$), even for the severe off-design operating condition.

3. Acceptable liquid turbine performance ($\eta_N > 0.80$) was demonstrated.
4. Steam quality was poor due to the severe off-design casing pressure. We expect the quality to improve considerably with operation closer to the design point, but some further work is needed to improve the steam quality at off-design conditions.
5. Pressure recovery of 100 psi was demonstrated at a diffuser efficiency $\eta_D = 0.70$.

Reference

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2. Cerini, Donald J. and Jeff Record. "Wellhead Power Production with a Rotary Separator Turbine. "Proceedings of the 17th Intersociety Energy Conversion Engineering Conference. August 1982.
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APPENDIX A

Inspection Report

S.O. 680000

Biphase Rotary Separator Turbine

The 54RST Biphase Rotary Separator Turbine is a first generation geothermal machine which was operated at Roosevelt Hot Springs near Milford, Utah, during 1982 and 1983. The first phase of the testing developed performance data. Phase Two was a steady-state, endurance test which brought the total operating time of the RST to over 4,000 hours.

At the end of the endurance test, the RST was partially disassembled in the field to conduct an evaluation. Both the primary and secondary rotor assemblies were returned to Trenton for disassembly and inspection. Certain parts were then modified to upgrade the flow capability of the machine to 10^6 pounds per hour.

This report summarizes the results of the inspection.

Inspection Report

S.O. 680000

54-RST

- Prior to disassembly of the barrels, the following visual observations were made:

Primary Rotor Assembly - Most surfaces exposed to the geothermal environment had a thin (0.015"), uniform coating of dark scale, apparently metallic sulfides. The inside of the tube matrix showed an accumulation of dirt, scale, and other debris. A thicker (0.10") white scale, apparently silica, covered only the outside diameter of the primary rotor. In a random pattern, various size sections of scale had separated from the wheel, giving a marbly checkered appearance. It appeared that the outer diameter of the wheel had contacted the "squeal-rings" in the case. After removal of the scale, it was observed that metal-to-metal contact had not occurred. "Dirt" had accumulated between the transfer holes on the "U"-Tube side of the rotor. No evidence of erosion or corrosion.

Secondary Rotor Assembly - Most surfaces exposed to the geothermal environment had a thin (0.015") uniform coating of dark scale, apparently metallic sulfides. No other scale. No evidence of erosion or corrosion.

- Disassembly of the Primary Barrel Assembly:

Inboard Journal Bearing (PMT-2000 F) - Good condition, dark spot on one pad. Mean radial clearance 0.006". Reinstalled.

Outboard Journal Bearing (PMT-2000 E) - Very good condition. Mean radial clearance 0.0045". Reinstalled.

Thrust Bearing, Inboard Pad (PMT-2100D) - Very good. Reinstalled.

Thrust Bearing, Outboard Pad (PMT-2100D) - Good, evidence of previous exposure to water. Cleaned and reinstalled.

Koppers Face Seal (PMT-2200B) - Seal face showed even wear pattern (no evidence of abnormal leakage) with several circular, concentric grooves. Returned to Koppers for evaluation and correction. "O"-rings from Koppers seal apparently did not leak. The Kalrez "O"-ring became flattened and should not be reinstalled. The other "O"-rings (EPDM, ethylene-propylene) swelled too much to be reinstalled. Replacement "O"-rings were ordered for all seals. EPDM used for all seal replacements.

Oil Seal Rings, Inboard Journal Bearing (TRP-588G) - The ring located at the inboard side of the bearing was originally machined with a rectangular cross-section. During operation the ring rotated with the

Outboard "U-Tube" Rotor Thrust Bearing (TRP-1189C) - Very good condition. Reinstalled.

Koppers Face Seal (PMT-2200A) - Seal showed an excellent wear pattern with no evidence of abnormal leakage. "O"-ring condition same as PMT-2200B. EPDM used for all replacement "O"-rings. Cleaned parts and reinstalled.

Koppers Piston Ring Intershaft Seal (PMT-2200C) - Rings showed excellent wear pattern on sleeve, with no evidence of abnormal leakage. EPDM used for all replacement "O"-rings. Cleaned parts and reinstalled.

Secondary Rotor (TRP-415D) - The rotor was thoroughly inspected and no evidence of erosion, corrosion, or stress-corrosion cracking attributable to operation in the geothermal environment was found. The thin, dark scale was removed using a "Sand/Hydroblast" method, and the rotor reinstalled.

"U-Tube" Rotor (TRP-415J) - The rotor was thoroughly inspected, and no evidence of erosion, corrosion, or stress-corrosion cracking attributable to operation in the geothermal environment was found. The "U-Tube" mount was machined to the "Upgrade" configuration, the rotor cleaned, rebalanced, and reinstalled.

"U-Tube" Elements (TRP-8DX1) - Visual inspection of the "U-Tube" showed light scaling and no evidence of erosion, corrosion, or stress-corrosion cracking attributable to operation in the geothermal environment. Additional testing for hydrogen embrittlement proved the "U-Tube" to be in excellent condition. New high-flow "U-Tubes" were manufactured and installed as part of the upgrade.

Instrumentation and Miscellaneous Parts - All probes and sensors were tested and replaced where required. Other parts of the secondary barrel assembly were cleaned, inspected, and reinstalled.

• **General Comments:**

Both the primary and secondary barrel assemblies were reworked to eliminate a troublesome oil leak between the barrel and the case. Cross-drilling and repiping of the barrel assemblies easily corrected this condition.

A new seal assembly procedure was developed to provide an easy method of checking the seal preload.

G. W. Romanyshyn
Design Engineer - R&PD

GWR:dob

shaft and wore to an "L" shaped cross-section, losing 1/3 of its width. This did not affect the seal leakage adversely. A new ring has been designed with balance holes to reduce the pressure loading on the seal face. Tighter tolerances have been specified.

Primary Rotor (TRP-415 GX1) - The rotor was thoroughly inspected and no evidence of erosion, corrosion, or stress-corrosion cracking attributable to operation in the geothermal environment was found. This inspection did, however, locate a flaw in the HAZ of the attachment weld which after extensive examination was determined to have occurred during the welding process.

The defective area was ground down to sound material; a stress-analysis performed; and the rotor was returned to service. (Note: Modifications were made on the rim of the rotor as part of the "Upgrade" to handle 10⁶ pounds per hour of geothermal fluid and to eliminate the metal tube matrix.) The inspection also included spin tests in the balance machine to determine the effect that the scale build-up had on rotor balance. Prior to scale removal, the rotor unbalance was 253.OZ-IN. As much scale as possible was removed using a "Sand/Hydroblast" method, and the imbalance was reduced about 40%. While this is still higher than the specified balance tolerance of 4.5 OZ-IN., it is known that by removing the metal matrix tubes an area of considerable scale/debris collection is eliminated. After completion of the "Upgrade", the rotor was balanced per specification.

Instrumentation and Miscellaneous Parts. - All probes and sensors were tested and replaced where required. Other parts of the primary barrel assembly were cleaned, inspected, and reinstalled.

. Disassembly of the Secondary Barrel Assembly:

Inboard Secondary Journal Bearing (TRP-395A) - Very good condition, some axial marks due to assembly and disassembly. Mean radial clearance 0.003". Reinstalled.

Outboard Secondary Journal Bearing (TRP-395) - Very good condition. Mean radial clearance 0.003". Reinstalled.

Inboard Secondary Thrust Bearing (TRP-395) - Very good condition. Reinstalled.

Outboard Secondary Thrust Bearing (TRP-1189E) - Very good condition. Reinstalled.

Inboard "U-Tube" Rotor Journal Bearing (TRP-395B) - Good, but scratched during disassembly. Mean radial clearance 0.003". Rebabbitted and installed.

Outboard "U-Tube" Rotor Journal Bearing (TRP-395C) - Very good condition. Mean radial clearance 0.003". Reinstalled.

Inboard "U-Tube" Rotor Thrust Bearing (TRP-1189D) - Very good condition. Reinstalled.