Demonstration of a Variable Phase Turbine Power System for Low Temperature Geothermal Resources

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Executive Summary

A new power cycle, the Variable Phase Cycle ("VPC") was demonstrated at the Coso Geothermal Resource of Terra-Gen in California. The VPC cycle used a liquid heat exchanger with a flashing, two-phase turbine to produce power from geothermal brine. The turbine rotated at low speed, directly driving a generator without a gearbox. The liquid heat exchanger maximized the heat extraction from the brine, demonstrating the potential of the VPC cycle to maximize power production from low temperature geothermal resources. Increases in power production from 20-40% can be achieved, depending upon the allowable brine return temperature.

The heat source for the VPC power plant was geothermal brine at 235F from the separator of a flash steam plant. The brine is normally re-injected without using the thermal energy. The VPC power plant generated 794 kWe from the previously wasted heat. The power produced agreed with the predicted power for the amount of brine available.

The power plant design is very simple compared to power plants based upon the organic Rankine cycle or Kalina Cycle. A simple liquid-liquid heat exchanger was used instead of a boiler and separator. There were no recuperative heat exchangers or gearboxes and the turbine was a simple design with no close tolerances. As a result, the cost of the major equipment for the demonstration plant was only $900/kWe. Turnkey cost for replicas of the 1 MWe system are estimated to be in the range $1400-1700/kWe installed.

Operation of the power plant and VPC cycle were very controllable and stable when the brine flow was well behaved. Startup and grid connection were routinely performed in a few minutes. However, bursts of flow from the flash steam power plant separator of up to 50% occurred in a short duration (~15 seconds). These excursions produced violent inputs in energy to the system. Initially, problems were encountered with operation of the power plant and equipment when these events happened. However, modifications to the control system and equipment were made which enabled stable operation when these bursts of energy input occurred.

During the period of the project, a severe decline in the brine flow from the flash steam separator occurred which limited the power production to about 200 kWe at the specified brine return temperature of 175 F. The power production was increased to 500 kWe by simply lowering the brine return temperature to 150 F, providing more heat input to the power plant. A companion Department of Energy program is being conducted by Energent to enable long term operation at these low return temperatures without heat exchanger scale deposits. The VPC power plant has been placed in standby condition with periodic operation until the brine flow is restored to a usable value or until the results of the companion program can be applied to enable operation of the heat exchanger at 125 F return temperature.

1.0 Introduction

1.1 Geothermal Power Availability

The selection and sizing of any power plant, geothermal or otherwise, depends on two main factors—heat source temperature and total heat input. The former determines the power generation cycle chosen and the latter defines the total power output. Higher temperature resources typically have higher efficiencies as determined by Carnot but they are unfortunately less common. There are generally far more resources at a lower temperature in a given area than there are high temperature resources [MIT Future of Geothermal Energy].

It is common for geothermal wells to have a depth of 3km (10,000 ft.) or less although it is within current drilling technology to reach as far as 10km (30,000ft). For example, Figure **1** and Figure **2** show ground temperatures at the depths of 3.5km and 4.5km respectively for the continental United States. From the maps it is immediately obvious that the vast majority of land is below 200C in temperature. Figure **3** estimates the total amount of heat at various depths and temperatures for the continental United States.

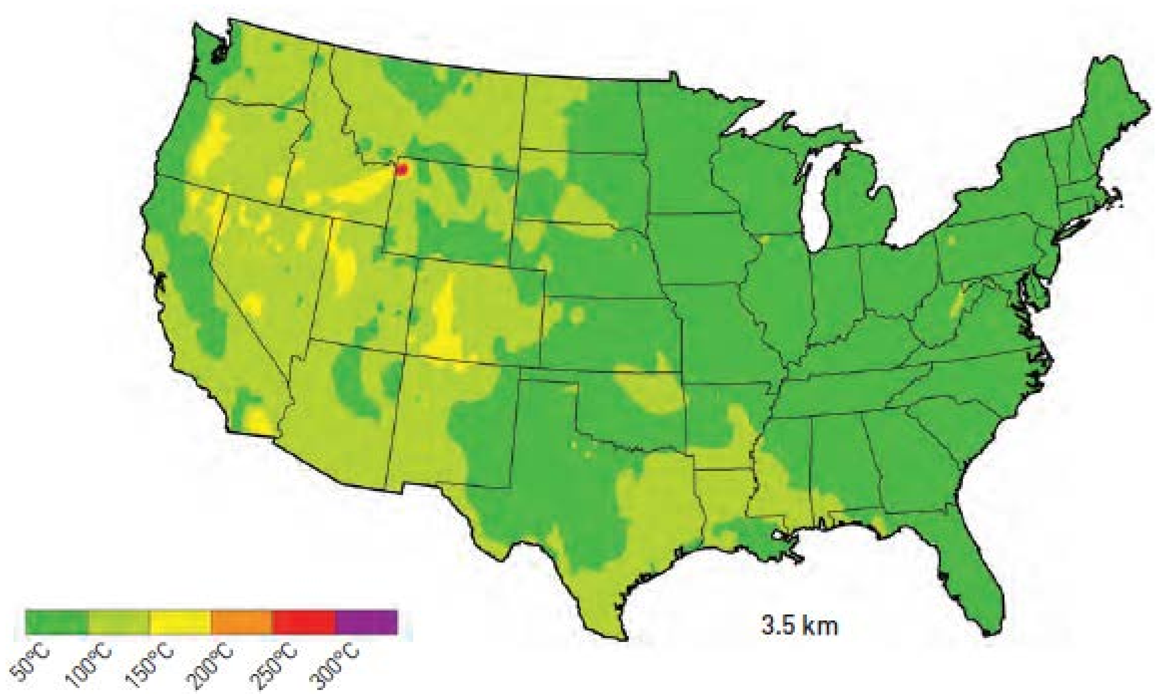


Figure 1 Ground Temperature for Continental United States at 3.5km Depth [MIT Future of Geothermal Energy]

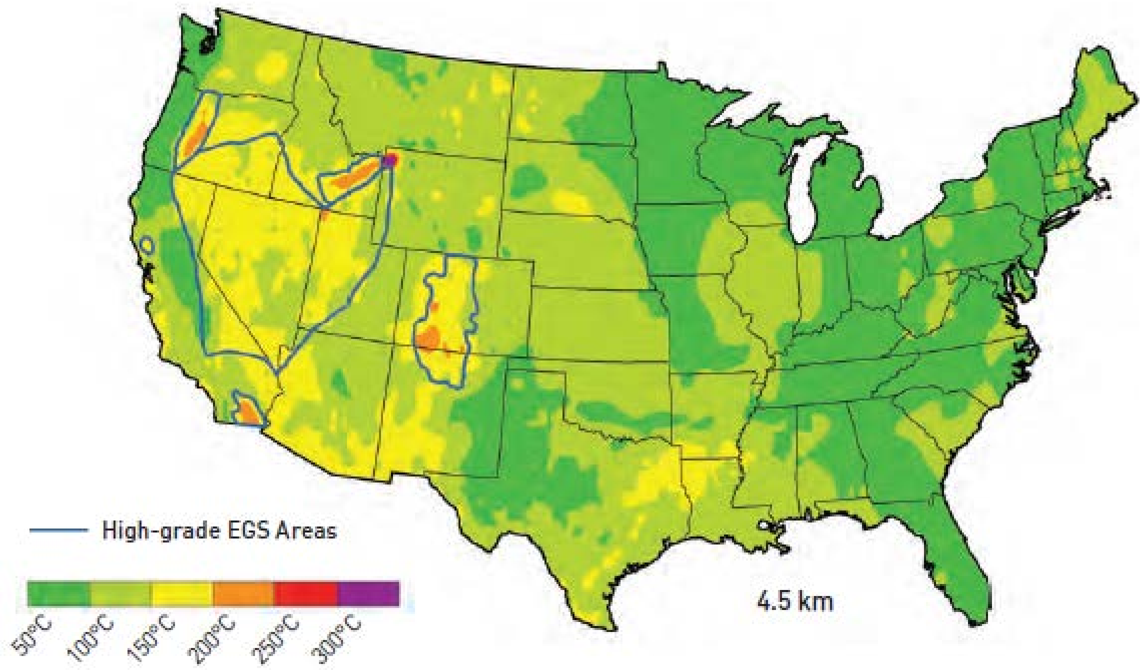


Figure 2 Ground Temperature for Continental United States at 4.5km Depth [MIT Future of Geothermal Energy]

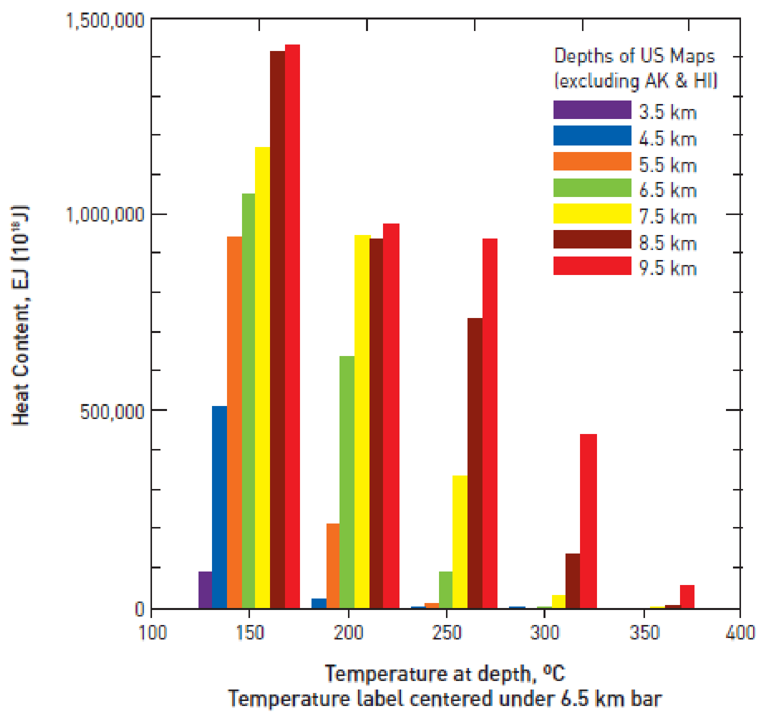


Figure 3 Estimated Available Heat at Various Depths for Continental United States [MIT Future of Geothermal Energy]

Notice that for shallower and more accessible depths the total heat available drops by more than an order of magnitude for every 50C temperature rise. It is only at depths greater than 5.5km that this effect disappears but there is greater technical and economic challenge to reaching deeper depths. Therefore, any technology that can efficiently generate power at lower temperatures will have a significant impact on the total electricity produced by the industry.

Many flashing steam geothermal power plants separate their liquid from the wells before it reaches the steam turbine but then make no thermodynamic use of the hot liquid. The separated brine may be re-injected into the ground or used for makeup water for the cooling towers or sent for some other use. This water typically has a temperature of 100-120C which has previously been considered as too low of a temperature to use. A system that could use this heat to make electricity could be potentially economically lucrative because the separated brine is an existing heat source that requires no additional qualification (it is generally already very well measured) and minimal additional infrastructure as no new wells are required and only minimal piping, cooling and electrical upgrades are necessary.

It is common for some areas of hydrocarbon development to reach relatively high temperatures [MIT Future of Geothermal Energy]. These wells may produce oil with a high temperature and/or may produce significant amounts of water in conjunction with the oil. This water is frequently separated and could then be used in any power generation cycle in an identical manner to geothermal brine.

1.2 Thermodynamic Limitations to Power Production

Limitations to power production from low temperature geothermal resources exist in conventional organic Rankine and Kalina cycles ("ORC" and "KC", respectively). The temperature profiles in the heat exchanger result in a point where the temperature difference is minimal. This is the "pinch point" temperature difference and its existence is a fundamental drawback for ORC cycles. The KC reduces the pinch point limitation somewhat, but at the expense of additional complexity and equipment. When designing an ORC cycle,

1. A small temperature pinch leads to "difficult" heat transfer and needs a larger heat exchanger area which can increase capital cost greatly.

2. A large temperature pinch leads to a greater temperature difference between the heat source and working fluid in the heat exchanger and thus less total heat exchange as the cooling and heating curves are not well matched, particular as one moves away from the pinch region.

In general the value of the pinch should be roughly 5 to 10C for an economic optimum.
On the 2nd item the desire to match the cooling curves comes from the desire to reduce entropy generation (the losses). In general the entropy generation S_{gen} where the heat Q is transferred from a high temperature T_H to a low temperature T_L is given from the 2nd law of thermodynamics as,

$$S_{gen} = Q/T_L - Q/T_H = Q(T_H - T_L)/(T_L T_H)$$

So that a small temperature difference T_H - T_L leads to a small S_{gen} and greater efficiency in the heat transfer process.

1.3 Maximizing the Available Energy for Power Production

To remove the pinch point temperature limitation a Variable Phase Cycle (Triangular Flashing Cycle) can be applied. This cycle was first proposed by Hays [Hays and Neal, 1977].
Figure **4** shows a Variable Phase Cycle (Triangular Flashing Cycle) typical T-H diagram.

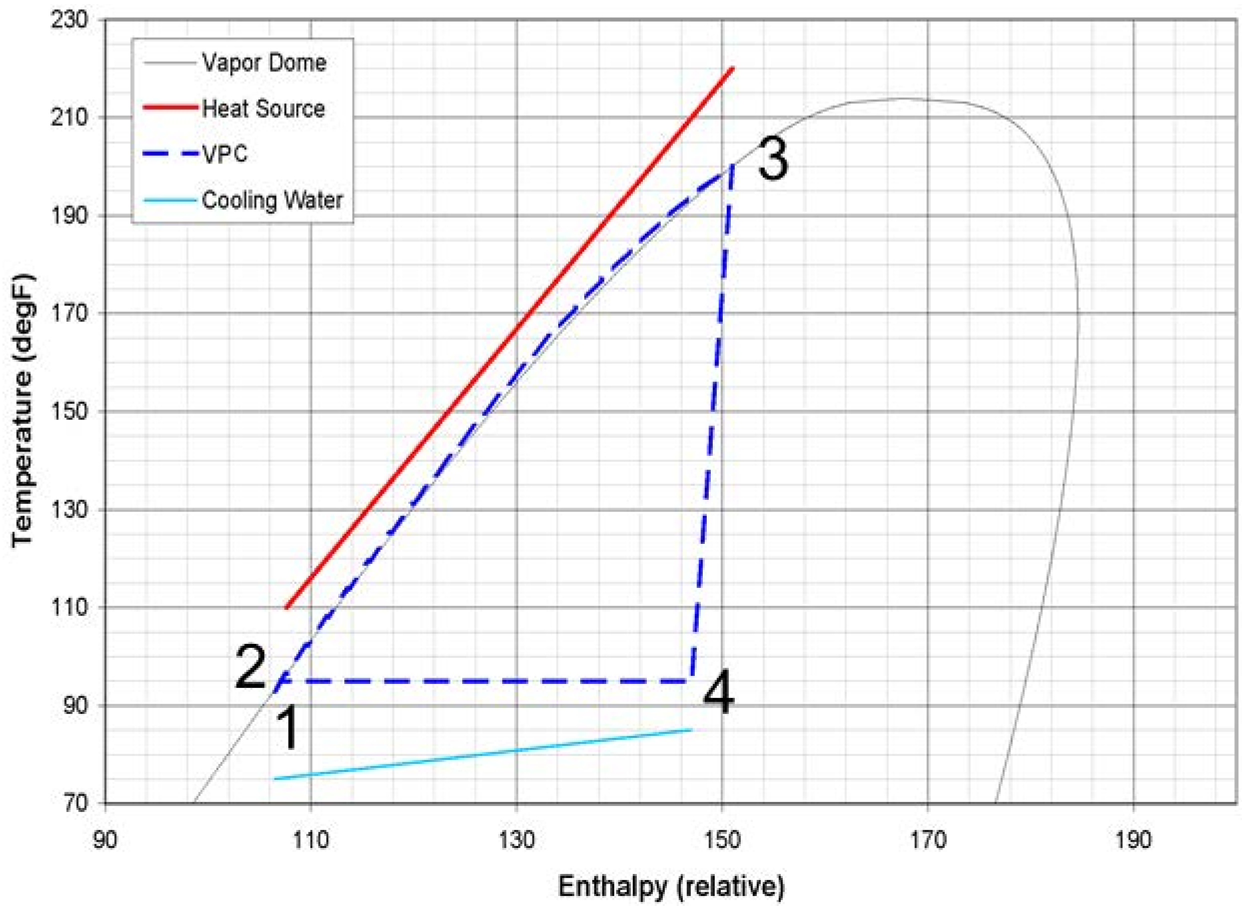


Figure 4 T-H Diagram for the Variable Phase Cycle ("VPC") also known as Triangle Cycle

To note is that the heat source cooling curve is very well matched (runs parallel) to the heating portion of the power cycle, the temperature difference remains nearly constant from state 2 to 3. There is no "pinch point" limitation as seen in boiling power cycles such as a steam or Rankine cycle. This allows the VPC to reduce the resource temperature lower than would be possible for a boiling power cycle which means a larger total heat input into the power cycle. The only limitation on the temperature to which the heat source can be reduced is the temperature of the cooling medium (typically water or air). For this reason the Variable Phase Cycle (Triangular Flashing Cycle) is the most efficient power cycle for sensible heat sources [Stiedel et al 1983,

Dipippo 2007]. But according to Dipippo the triangular flashing cycle requires a flashing liquid expander to generate electricity where the cycle fluid flashes. For this reason the motivation is high to apply flashing liquid expanders in the most efficient cycle to generate electricity in geothermal applications. Figure 5 shows a typical process diagram for a Variable Phase Cycle. Note that the positions 1-4 on the process diagram correspond to the positions 1-4 on the T-H diagram in
Figure **4**.

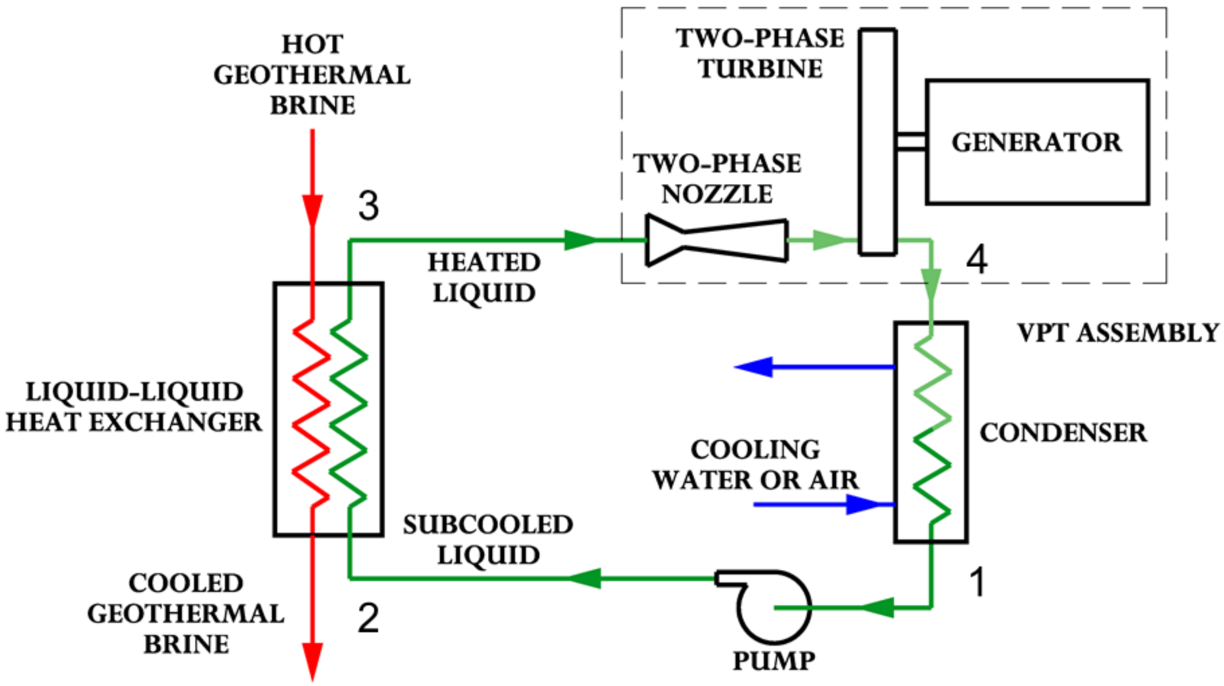


Figure 5 Generalized Process Diagram for a Variable Phase Cycle

Radial inflow turbines are one of the most commonly used turbine configurations used in geothermal power but they are not well suited to long-term use in applications that have significant liquid and vapor components as part of normal operation. The primary problems are:

1. Flow forces tend to push any liquid droplets towards the exit (radially inwards) while centrifugal forces tend to push liquid radially outwards (towards the inlet). As a result it is likely that a recirculation pattern will form with some amount of liquid trapped and recirculating near the turbine inlet or near the nozzle-impeller gap region at the impeller inlet. This can potentially damage the turbine blades with continuous liquid impact. Additionally the presence of liquid has been found to reduce performance and cause excessive vibrations [Kevin's references]

2. The radial inflow turbine blades are relatively more delicate and more susceptible to resonance than other types such as radial outflow or axial flow turbines. This is due to the fact that the blades are thinner than other turbine types.

A successful two-phase turbine and nozzle has been developed that will be called the Variable Phase Turbine ("VPT"). The turbine uses a two-phase nozzle that is based on work originally performed at the NASA Jet Propulsion Laboratory. At the inlet the fluid may be liquid, two-phase or a supercritical fluid. As the fluid travels down the converging-diverging nozzle (see Figure 6) the pressure drops and more vapor begins to form. The working fluid is chosen such that it has a low surface tension and viscosity. These properties allow small droplets, typically on the order of 10 microns, to be sheared from the liquid body by the accelerating vapor. A small droplet size is key for allowing the vapor and liquid phases to couple together well. If the two phases are not coupled and the gas passes around the liquid droplets which are traveling at a lower speed then friction is generated which harms the turbine and cycle efficiency.

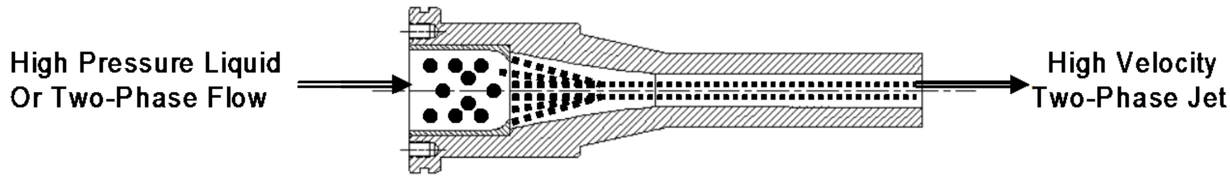


Figure 6 Cross-section of the Two-Phase Converging-Diverging Nozzle

The Variable Phase Turbine itself is a single-stage axial impulse turbine. Because the flow is two-phase a high mass flow can be achieved with a low jet velocity. This enables a relatively low-speed turbine such that a gearbox is not necessary (see Figure 5). A low jet velocity also reduces the potential for erosion and is well below the erosion threshold for steel or titanium which are currently used. Calculations and some experimental data with a 10kW test system suggest that the jet velocities are also below the erosion threshold for aluminum. The two-phase mixture exits the nozzle and enters the axial impulse turbine (
Figure 7). Provided that the liquid droplet size is small, a large fraction of the droplets may follow the gas phase in turning through the blade passages. Less liquid actually impacts the blades—which is desirable because such impacts and flow of the collected liquid result in a frictional loss. The liquid on the blades also transfers energy to the moving blade is flung off the blades at the exit.

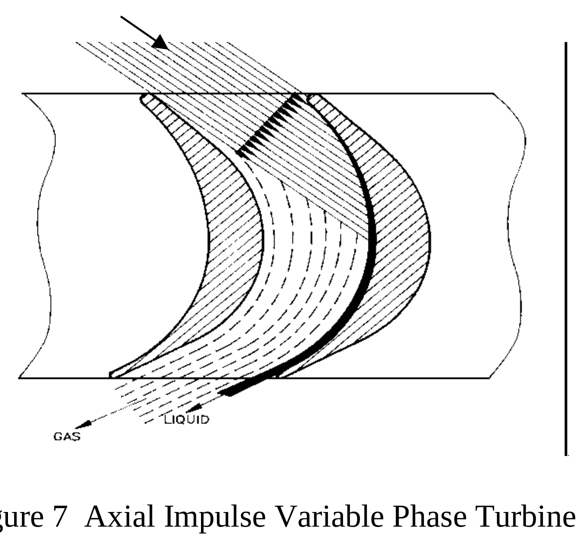


Figure 7 Axial Impulse Variable Phase Turbine for Two-Phase Flow

In addition to the test experience for the two-phase nozzles and turbines, a Variable Phase Cycle pilot plant was built and operated. The pilot plant, shown in figure 8, was operated with R245fa and R227 refrigerant working fluids. The refrigerant was electrically heated in a liquid heat exchanger to a maximum temperature of 240 F at a pressure of 270 psia. The flow was flashed in the turbine. The vapor and liquid mixture was condensed in a water cooled condenser. The condensate was then pressurized by a pump and returned to the heater, following the process illustrated in figure 5.

Figure 8 Variable Phase Cycle Pilot Plant Operating with R245fa

Operation was of the pilot plant, which was the first closed cycle two-phase power plant in the world, was smooth and controllable with no vibration or excessive noise. Refrigerant flow rate and temperature were routinely varied with no problems.

The Variable Phase Turbine shown in the figure generated a maximum of 7 kWe, the measured power agreeing closely with predictions, figure 9. The design codes from which the predictions were made, are those used to design the VPC power plant for high power applications.

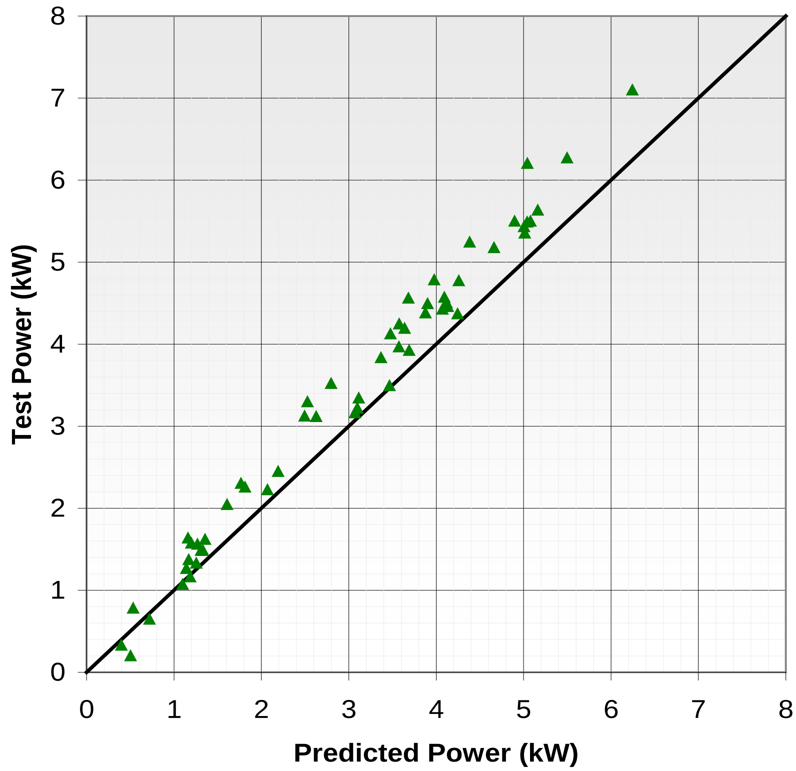


Figure 9 Performance of Variable Phase Turbine in Pilot Plant
Compared to Predicted Performance

1.4 Demonstration of Commercial Viability of Variable Phase Cycle

To confirm the commercial viability of the Variable Phase Cycle, a 1 megawatt demonstration plant was proposed. The project objectives were to design, build, install and operate a 1 megawatt geothermal plant at the Terra-Gen BLM East power plant, at Coso Geothermal Resource, California. The geothermal plant used separated brine for the heat source. A maximum of 1,000,000 lb/h at 235 F was stated to be available. The Variable Phase Turbine Cycle was to use refrigerant R-134A as the working fluid.

A commercial geothermal demonstration plant was constructed at an existing facility near Coso Junction, California, United States. The existing facility has multiple flashing steam turbines spread over a geographically large area. A collection system gathers the output of multiple wellheads for the use of one or more flashing steam turbines. There is liquid and vapor as part of the wellhead flows so it is necessary to separate out the liquid brine prior to the steam turbines. This liquid is reduced in pressure to feed some additional steam to lower-pressure steam turbines but a significant amount of atmospheric-pressure geothermal brine is generated with a temperature of 235 F. Previously this brine was simply being sent to re-injection wells. A low-temperature Variable Phase Cycle was installed to convert some of this wasted heat into electricity prior to re-injection. The results of the project are reported herein.

2.0 Accomplishments

The project successfully demonstrated the operation and performance of the first commercial scale Variable Phase Cycle ("VPC") power plant. This cycle, also known as the "Triangle Cycle", was demonstrated to maximize the energy extraction from a given geothermal brine source at 235 F. The maximum power generated was 800 kW. The power plant operation was smooth and controllable for well-behaved brine sources.

The measured power output of the Variable Phase Turbine agreed closely with the predicted performance. The maximum power tested, 800 kW, was within 3% of predicted. At part load the power was increased by drawing down the brine temperature to ~145 F, lower than could be reached with an ORC. At 50% of brine flow the power was increased to 530 kW.

The durability of the power plant was demonstrated when sudden excursions of more than 50% of the brine flow were experienced in less than 15 seconds. Controls were implemented which successfully accommodated these severe burst inputs of energy into the system. This result enables reliable power production from unsteady sources of waste energy, such as are likely to be found in flash steam plants having two-phase gathering systems.

The results support the conclusion that application of the Variable Phase Cycle to generate power from low temperature geothermal resources and enhanced geothermal resources can increase the power production by 15-30%. This result can reduce the cost per megawatt of new projects and increase the power produced from available geothermal resources.

3.0 Project Activities

3.1 Resource Characterization

The resource at Coso geothermal field supplies high temperature two-phase flow to flash steam plants having an installed capacity of 240 megawatts. The downhole temperatures are in the range of 550-600 F. The high downhole temperatures result in high concentrations of minerals in the liquid phase. The energy source for the VPC power plant is brine from one of the flash steam separators. The separated brine is at a temperature of 235 F. A typical liquid analysis is provided in Table 1. The high silica content, 528 ppmw, poses a scaling problem for heat exchangers.

Table 1 Brine Composition from Demonstration Separator, ppmw

Na	6500
K	1015
Ca	229
Li	46.2
Sr	23.6
B	222
SiO_2	528

Cl	11660
F	2.74
SO4	63.8
HCO3	54.8

In order to determine the scaling potential, a plate and fin heat exchanger and tube and shell heat exchanger were tested with the geothermal brine at the Coso site ("Coso") to determine fouling potential due to scale deposits. Figure 10 shows the heat exchanger test system.

Figure 10 Heat Exchanger Test System

The shell and tube heat exchanger was qualified for the brine chemistry. The plate and fin heat exchanger had scale deposits that blocked the internal passages. Some initial restrictions were present. However, after 1 month the passages were completely blocked. Cleaning attempts failed. Figure 11 is the plate-fin heat exchanger after a 1 month test period showing the silica deposits.

Figure 11 Plate – Fin Heat Exchanger Deposits

Figure 12 shows the shell and tube passage after 1 month exposure. A thin coating of silica occurred which was readily cleanable.

Figure 12 Sections of Shell and Tube Heat Exchanger Test Articles

As a result of the testing the shell and tube approach was selected. As a result the power plant was field erected rather than factory built, resulting in additional cost and time.

The measured brine flow from the separator at the time of the Phase 1 study, January 2005, is shown in figure 13. The maximum flow is 1,300,000 lb/h with excursions to a low of about 500,000 lb/h during operation. The average was about 800,000 lb/h (1,684 gpm) and that was selected as the design point. Concerns about scale deposits in the reinjection wells caused the operator to limit the re-injection temperature to 175 F for long term operation. This limit reduced the available energy for the VPC. However, short duration testing was later permitted when the temperature was lowered to values as low as 120 F to demonstrate the ability of the VPC to maximize the exergy efficiency.

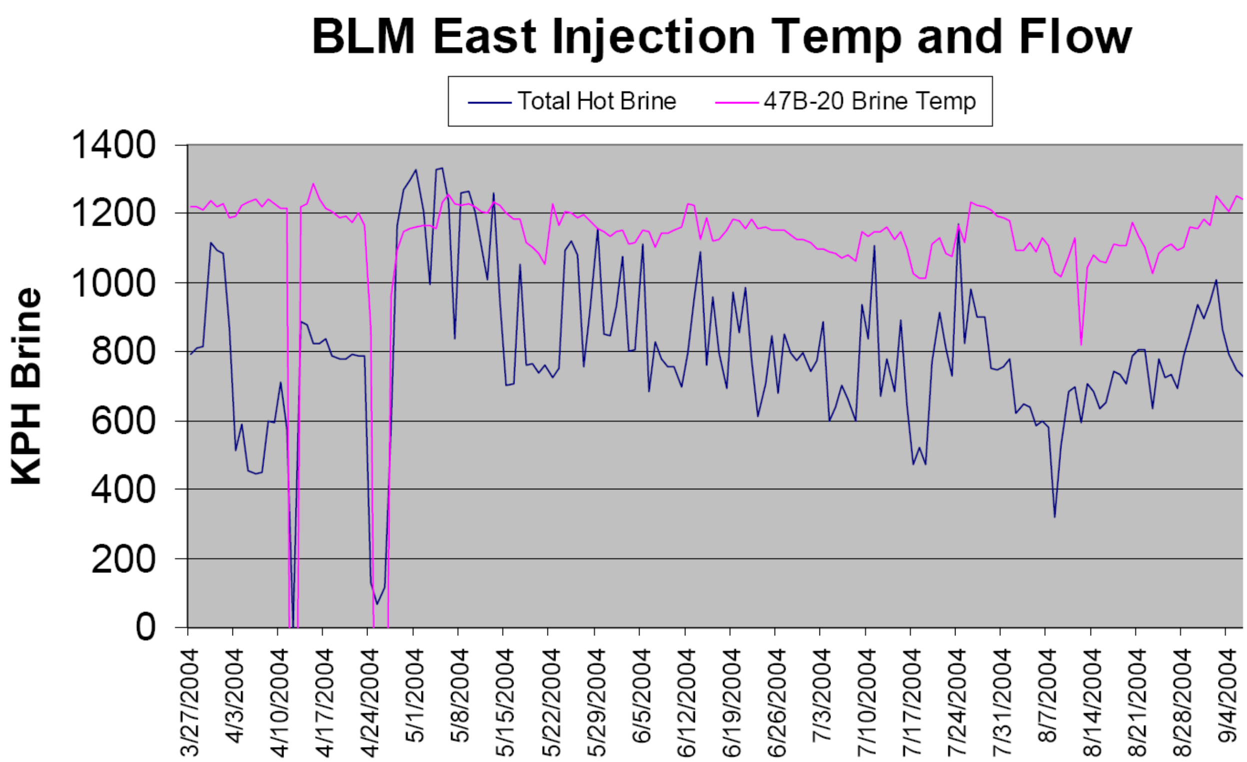


Figure 13 Brine Flow Measurements for BLM East at the Time of Project Start

Initiation of the second phase was delayed until July, 2009. During this time the brine flow had declined due to a decline in the well field. The current maximum flow is about 500,000 lb/h with an average of about 300,000 lb/h.

3.2 System Analysis

Power Analysis and design of the power plant was carried out with the Variable Phase Cycle system code. The code inputs are shown below:

3.2.1 System Code Inputs

Heat Source

Type (liquid, gas, or two-phase); temperature; flowrate, permissible heat exchanger outlet temperature

Variable Phase Cycle

Type (liquid, gas, supercritical, two-phase); working fluid (e.g., R-134A, R245fa; Isobutane,etc.)

Heat Exchangers

Minimum approach temperature difference for heat exchanger and condenser; cooling source for condenser (water or air, inlet and outlet temperature); estimated values of overall heat transfer coefficient, U, for heat exchanger and condenser

Variable Phase Turbine

Nozzle length; number of nozzles; nozzle setting angle; turbine speed; turbine U/C or mean line diameter

Pumps

On turbine shaft or electric motor driver; boost pump or not; efficiency algorithm or specified efficiency

Piping System Pressure Drops

Specified pressure drops in valves and piping

3.2.2 System Code Output

The system code performs a heat balance to match the working fluid conditions to the heat source for the specified heat exchanger process inputs and approach temperatures. The system code calls up the two-phase nozzle code sub-routine and calculates the two-phase nozzle performance and geometry for the specified inputs of nozzle length and number. The system code subsequently uses the turbine input parameters to calculate the turbine performance and geometry. The system code determines the turbine output power, the net VPC cycle power, taking into account the pumping requirements, and the estimated net power plant power including estimates of cooling tower fan power, cooling water pump power or, in the case of an air-cooled condenser, the air fan power requirements.

A cost algorithm is subsequently used the estimate the cost of the power plant

3.2.3 Analysis for Coso Resource

The results of the analysis for the brine at the BLM East separator at Coso Resource is shown in Table 2. At the design conditions of 235 F temperature and 819,470 lb/h of brine the Variable Phase Turbine produces 1039.5 kWe from the generator. Since the generator is shaft driven through the generator, the generator electric output is the net for the cycle, except for auxiliaries. The minimum heat source temperature is 175 F, limited by the geothermal operator to avoid scaling of the reinjection wells.

Table 2 System Code Output for Design Case

7/8/2010 9:03 VPT13 Coso R134a 1MW 175F 100603 design.xls

R134a

Net Pow: **1039.5 kW**

1000
819470

Axial Turb File:	vptcycle
Run AxTurb as:	DLL
OverWriteCheck:	No

1.3

	Temperature	Pressure	Enthalpy	Density	Quality	Flow Rates			Pipe Size	Velocity	Loss/100	Efficiency	Power		Delta P	ΔHeight
	°F	psia	BTU/lb	lb/ft^3	-	lb/s	lb/hr	gpm/CFM	in	ft/s	psi	-	kW	BTU/lb	psig	ft
Heat Source Inlet	235.00	130.0	203.79	59.2550	0	227.631	819,470	1724.2	12	4.9	-0.2	Use 98.4% Cp correction factor			10	0
Heat Source Exit	175.00	120.0	143.39	60.7087	0			1682.9	8	10.7	-1.5	-	-	-	-	-
Heat-X Inlet	92.68	599.60	106.28	74.5659	0	251.51	905,436	1513.9	8	9.7	-1.50	-	-	-	4.91	0
Heat-X2 Inlet	152.20	594.69	127.69	65.5011	0	251.51	905,436	1723.4	12	4.9	-0.22				2.38	
Heat-X Exit	213.70	592.31	160.95	40.9346	0	251.51	905,436	2757.7	8	17.6	-2.68	-	Windage		2	8
Nozzle Inlet	213.31	588.33	160.95	40.7525	0	251.51	905,436	2770	8	17.7	-2.69	0.9595	7.29	Axturb	-	-
Nozzle Exit	89.00	117.15	153.76	3.6622	0.66053	251.51	905,436	4120.6	18	38.9			5.36	Theoretical	-	-
Rotor Exit	89.00	117.155	155.23	3.5599	0.68051	251.51	905,436	4239	18	40.0		0.7950	1514.63	5.7079	1.5	0
Condenser Exit	87.98	115.65	104.68	73.8612	0	251.51	905,436	1528.3	10	6.2	-0.50	-	-	-	0.5	-3
Boost Pump Inlet	87.98	116.69	104.68	73.8642	0	251.51	905,436	1528.3	12	4.3	-0.20	82.0%	-31.268	-0.1178	38.6	-
Boost Pump Exit	88.34	155.26	104.80	73.9277	0	251.51	905,436	1527	8	9.7	-1.51				3	12
Main Pump Inlet	88.34	146.10	104.80	73.9013	0	251.51	905,436	1527.5	8	9.7	-1.51	78.0%	-395.6	-1.4908	-	-
Main Pump Exit	92.68	610.49	106.28	74.5946	0	251.51	905,436	1513.3	6	17.2	-6.28				15	-8
Cooling Water Inlet	68.00	36.0	36.20	62.3202	0	851.373	3,064,944	6131.6	14	12.8	-1.2	Pump Effic.	-77.7	-0.0864	9.56	25
Cooling Water Exit	83.00	15.6	51.14	62.1872	0			6144.7	14	12.807	-1.132	70.0%			-	-

HEAT EXCHANGER PARAMETERS

Heat-X Exit T (°F)	213.70
Max Heat-X Exit T (°F)	213.86
R134a Critical Pressure (psia)	588.75
HX Heat Xfer (MMBTU/hr)	49.499
U (BTU/hr ft^2F)	200
Total Area (ft^2)	7373
LMTD (°F)	44.95
Avg TD (°F)	33.57
Min. ΔT pinch desired (°F)	10.00
Min. ΔT pinch actual (°F)	19.26

CONDENSER PARAMETERS

Condensing T. (°F)	88.18
Condensing P. (psia)	115.65
Subcooling (°F)	0.20
Cond. Heat Xfer (MMBTU/hr)	45.776
U (BTU/hr ft^2F)	100
Total Area (ft^2)	39147
LMTD (°F)	11.62
Avg TD (°F)	11.69
Min. ΔT pinch (des)	6.00
Min. ΔT pinch (actual)	6.00

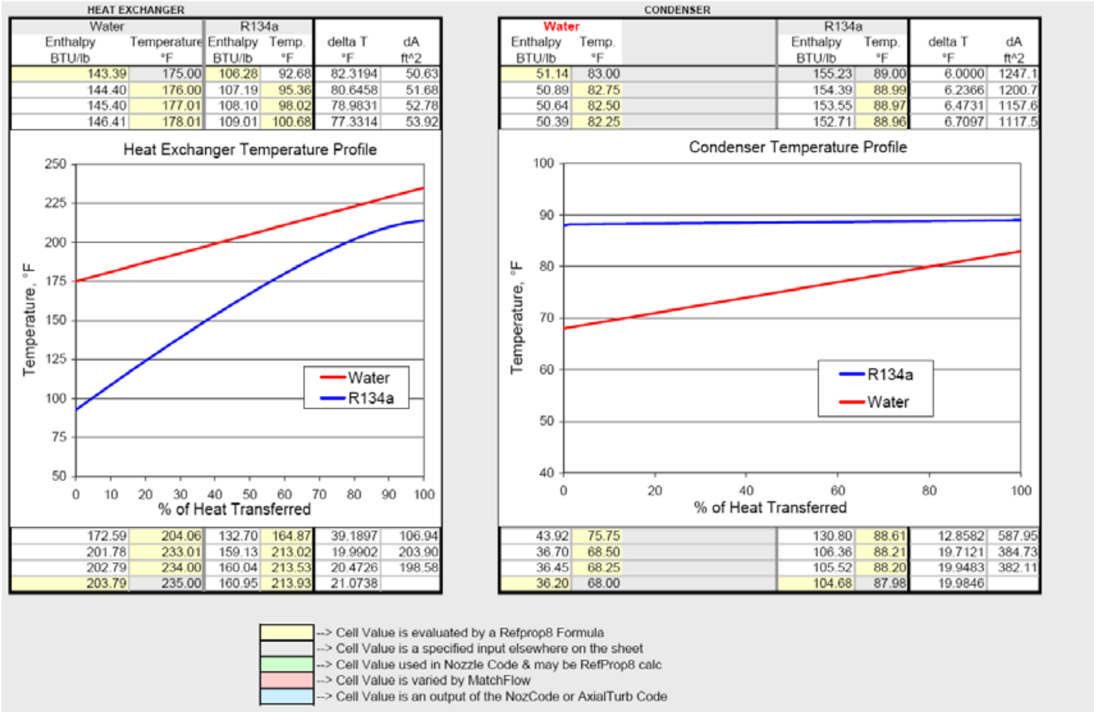
MISCELLANEOUS PARAMETERS

904.92

HeatSource fluid	Water	
	Choices	
Type of cycle?	Liquid	
Vary HXExitT to get HX pinch?	No	
Vary Cond. T. to get pinch?	No	1119.0
Main Pump on Turbine Shaft?	Yes	1074.3
Use Boost Pump?	Yes	
	Efficiencies	
Generator Efficiency	96.0%	
Inverter Efficiency	100.0%	
HX Heat Xfer Efficiency	100.0%	
Condenser Heat Xfer Efficiency	100.0%	
	Other Parameters	
Generator Flow Fraction	0.0%	
Rotor Speed (rpm)	3672	
u/c	0.497	
Number of Nozzles	10	
Nozzle Angle (deg)	20	
Nozzle Length (in.)	12	

OTHER WORKING CYCLE INFORMATION

	Required R134a flow rate		251.51	lb/s
	Heat rate to working fluid		13750	BTU/s
	Heat-X Exit Pipe ID		12	in
	Heat-X Exit Pipe Velocity		7.82	ft/s
	Nozzle Exit Enthalpy (Isentropic)		153.46	BTU/lb
0.8864	Nozzle Exit Velocity (Noz Code)		600.05	ft/s
1.3071	Nozzle Exit Diameter		1.4370	in
	Rotor Exit Vapor Flowrate		171.16	lb/s
	Rotor Exit Liquid Flowrate		80.355	lb/s
	Mean Line Rotor Diameter		1.4583	ft
	Nozzle Rotor Coverage Angle		27.668	°
	Boost Pump Inlet NPSHA		2.7365	ft
	Pump NPSHA/Stages	58.79	ft	2
	Pump RPM / Ns / Eff	3636	1448.5	82.8%
	Generator Power Dissipation		57.424	BTU/s
	Gross Shaft Power Rate		4.3249	kW/(lb/s)
	Gross Electric P (Turb&Mpump)		1074.3	kW
	Cycle Efficiency		7.5%	
	Plant Net Thermal/Exergy Effic.		7.2%	2.6%
	Power Rate (kW/MMBTU/hr)		21.00	



HEAT EXCHANGER

Water Enthalpy BTU/lb	Water Temperature °F	R134a Enthalpy BTU/lb	R134a Temp. °F	delta T °F	dA ft^2
143.39	175.00	106.28	92.68	82.3194	50.63
144.40	176.00	107.19	95.36	80.6458	51.68
145.40	177.01	108.10	98.02	78.9831	52.78
146.41	178.01	109.01	100.68	77.3314	53.92
172.59	204.06	132.70	164.87	39.1897	106.94
201.78	233.01	159.13	213.02	19.9902	203.90
202.79	234.00	160.04	213.53	20.4726	198.58
203.79	235.00	160.95	213.93	21.0738	

CONDENSER

Water Enthalpy BTU/lb	Water Temp. °F	R134a Enthalpy BTU/lb	R134a Temp. °F	delta T °F	dA ft^2
51.14	83.00	155.23	89.00	6.0000	1247.1
50.89	82.75	154.39	88.99	6.2366	1200.7
50.64	82.50	153.55	88.97	6.4731	1157.6
50.39	82.25	152.71	88.96	6.7097	1117.5
43.92	75.75	130.80	88.61	12.8582	587.95
36.70	68.50	106.36	88.21	19.7121	384.73
36.45	68.25	105.52	88.20	19.9483	382.11
36.20	68.00	104.68	87.98	19.9846	

3.3 Variable Phase Turbine ("VPT") Design

The VPT for this project was designed with 10 two-phase nozzles discharging to a titanium alloy rotor. The arrangement is shown in figure 14, a solid model of the VPT. The liquid enters a volute at high pressure. The volute feeds flow to the nozzles which accelerate the flow to a pressure near that of the condenser. The two-phase jets drive the impulse rotor at 3600 rpm. The flow discharges from the rotor in a horizontal exit pipe.

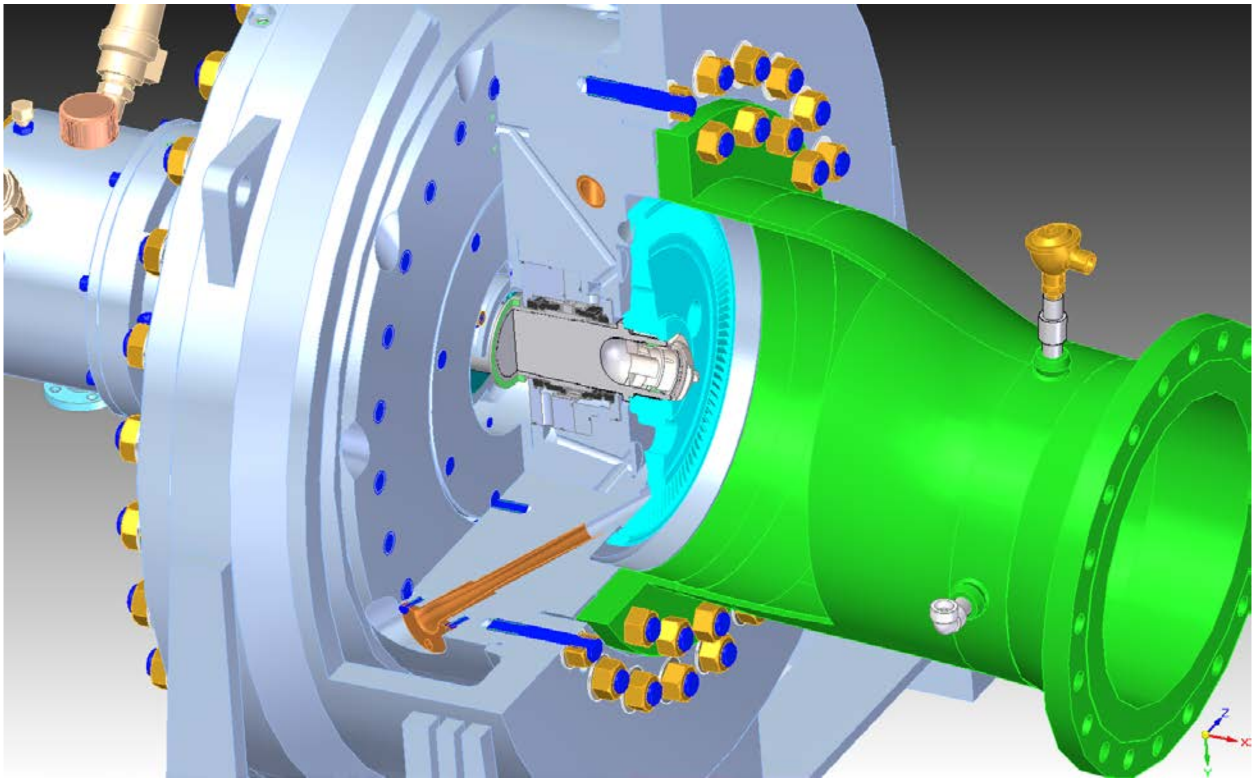


Figure 14 Variable Phase Turbine Solid Model

The nozzle profile and the blade profile were determined by the proprietary design codes which are included in the system code.

Figure 15 shows one of the nozzles. The nozzle flow is supersonic as a result of the low speed of sound in two-phase flow. The geometry has a converging diverging profile for efficient acceleration of the flashing refrigerant.

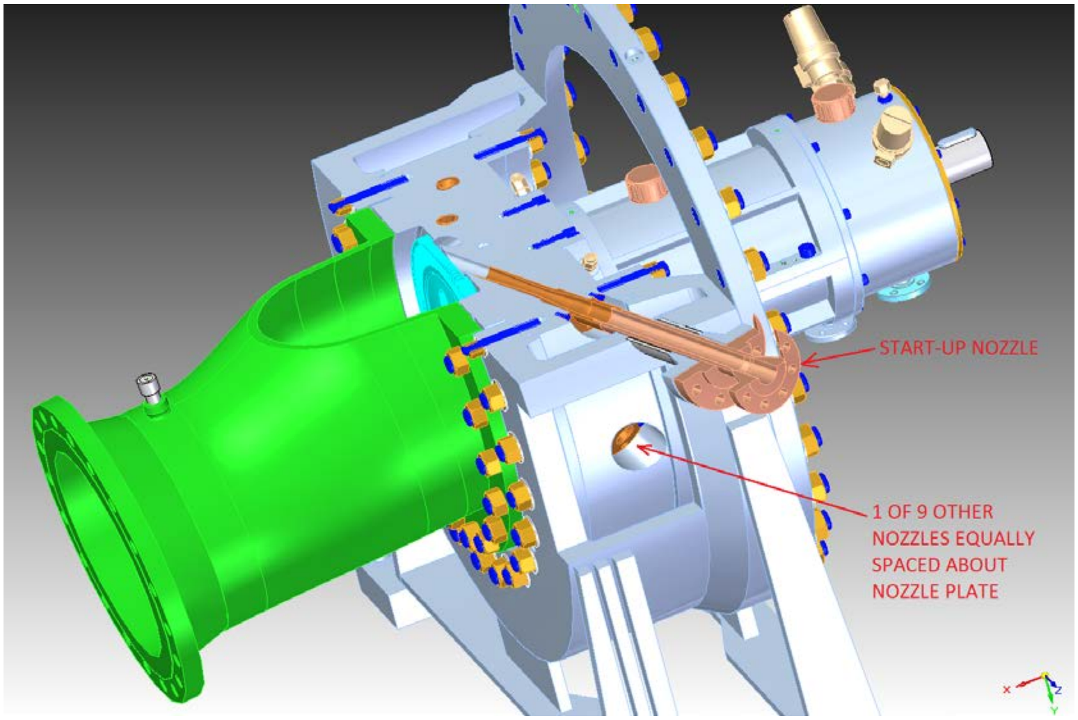


Figure 15 Variable Phase Turbine Nozzle Design

As shown in figure 16, a cross section of the assembly, the turbine is supported by two, tilting pad bearings. A thrust bearing is supplied to accommodate the axial thrust. The refrigerant working fluid is sealed by the pressurized oil seal shown. The oil pressure is maintained above the refrigerant pressure to ensure there is no refrigerant leakage.

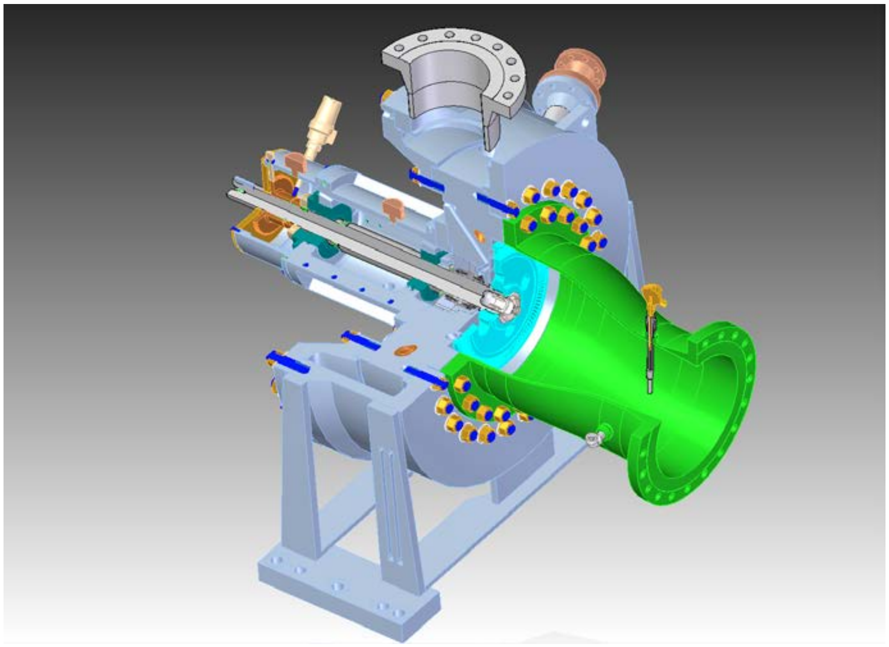


Figure 16 Variable Phase Turbine Assembly

The volute contains an inventory of high pressure refrigerant at the temperature produced by the heat exchanger. A quick closing trip valve is provided in the pipe feeding the volute. The volute contained enough high pressure refrigerant to cause rotation after the valve closed. To reduce the speed a pipe with a quick opening valve was added to direct flow from the volute to the back side of the wheel in the reverse direction of rotation. The resulting drag force on the wheel reduced the speed.

3.4 Power Plant Design

A solid model of the Variable Phase Cycle power plant is shown in figure 17. Heat is transferred in the Brine Heat Exchanger to the refrigerant R134A working fluid. The heated refrigerant flows to the Variable Phase Turbine where it flashes producing power. The turbine drives a Generator and the main refrigerant Pump. The two-phase refrigerant leaving the turbine flows to the water cooled Condenser. The condensate flows into the Inventory Tank which serves as a hot well. A Boost Pump transfers the refrigerant to the main Pump which returns the refrigerant at high pressure to the Brine Heat Exchanger.

The complete one megawatt Variable Phase Cycle power plant was designed to utilize waste brine from the flash steam separator at Coso. Process and Instrumentation Diagram and electrical single line drawing.

Equipment specifications were prepared for the heat exchanger, generator, pumps, condenser and balance of plant components. Manufacturer submittals were used to design the power plant.

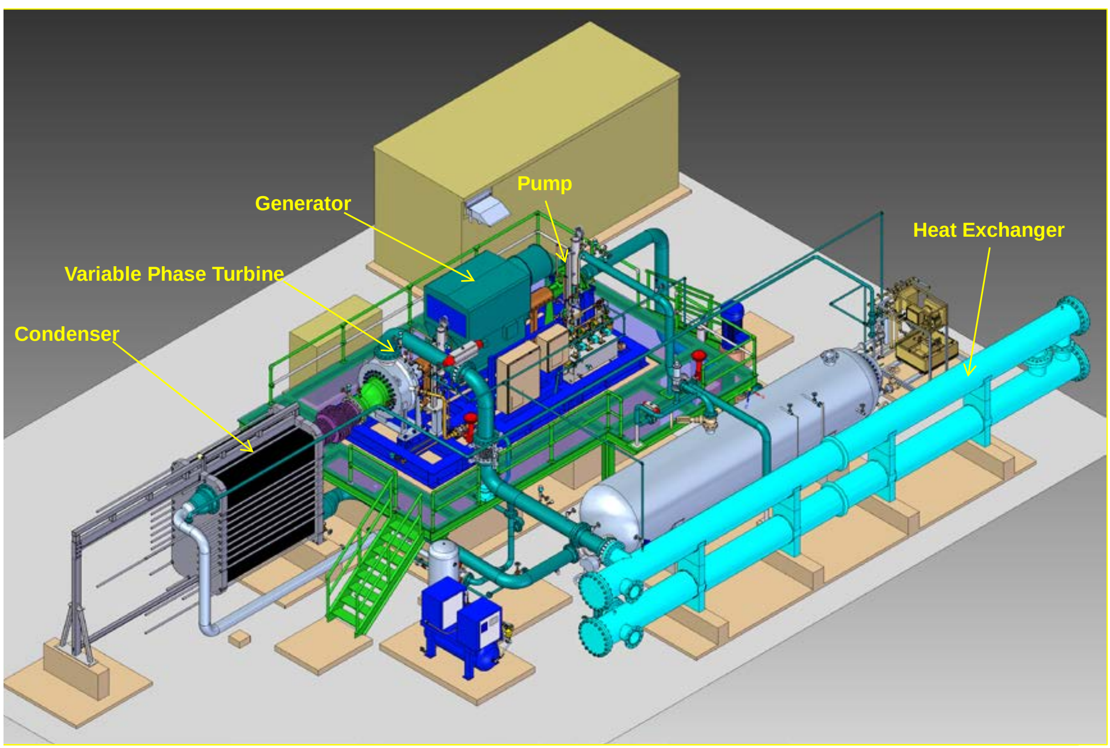


Figure 17 Variable Phase Cycle 1 Megawatt Power Plant

Unique features of the power plant are:

1. A liquid heat exchanger is utilized, reducing construction costs and controllability relative to a vaporizer. The liquid heat exchanger eliminates the pinch point limitation to heat transfer found in ORC vaporizers.

2. The VPT rotates at 3,600 rpm, enabling direct drive of a generator, eliminating a gearbox and gearbox lube oil system.

3. The main pump is driven by the double ended shaft of the generator eliminating generator and electrical motor inefficiencies from the pumping power requirements and enabling a smaller generator.

3.5 VPT Manufacturing

3.5.1 Variable Phase Turbine Rotor

The titanium alloy turbine blisc is shown in figure 18. The outer diameter is 16 inches. A shroud was applied to reduce liquid splash and the resultant windage losses. The rugged construction, as contrasted to thin, three-dimensional radial blading, proved to be very durable.

Figure 18 Variable Phase Turbine Rotor

3.5.2 Nozzles and Casing

The nozzle and casing assembly is shown in figure 19. The casing is welded construction, designed to operate at 650 psig. High pressure refrigerant enters the top port, is expanded to low pressure and exits through the nozzles (the ellipses surrounding the hole for the shaft). The smaller port at the side was used for a startup nozzle.

Figure 19 Nozzle and Casing Assembly

3.5.3 Variable Phase Turbine Assembly

The turbine rotor installed in the casing is shown in figure 20. A mechanical device with an interior expanding clamp was used to secure the rotor to the shaft. The two-phase flow leaving the blades flows in an axial direction to the condenser.

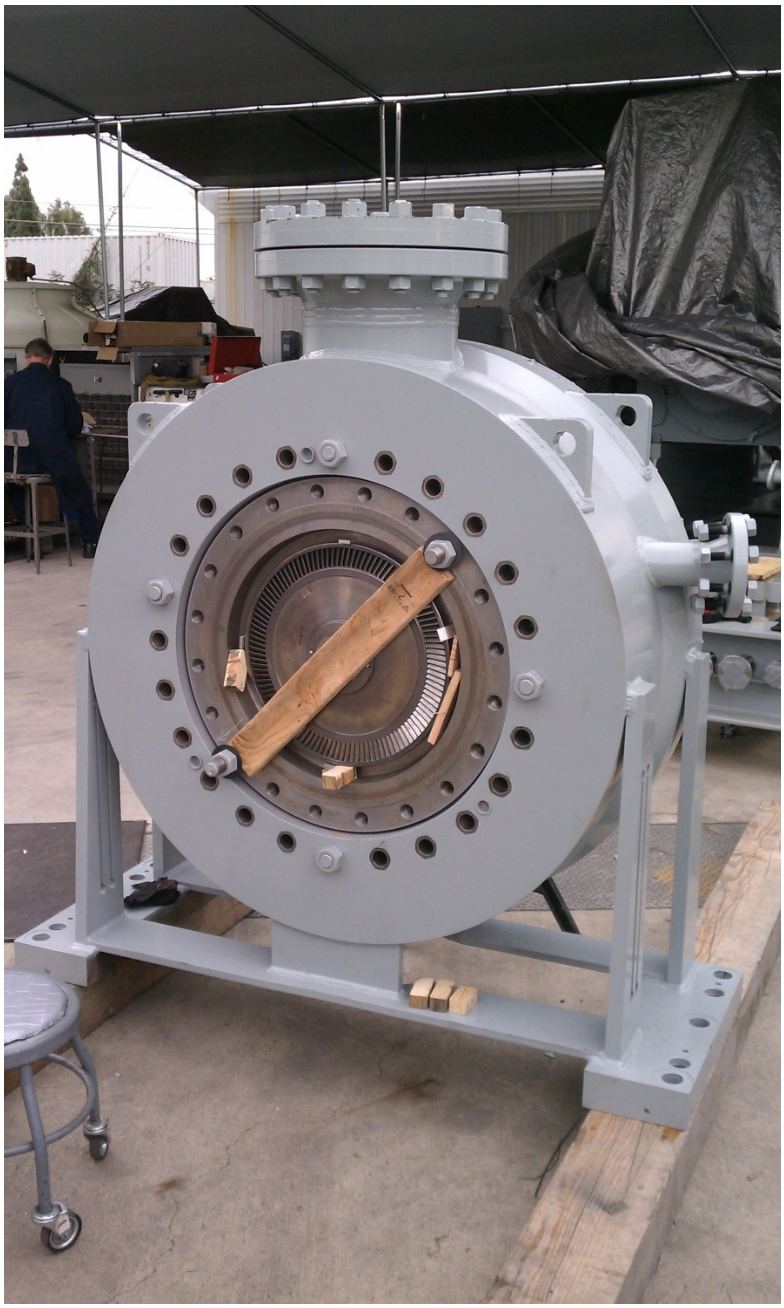


Figure 20 Variable Phase Turbine Rotor Installed in Casing

3.5.4 Rotating Assembly

The completed rotating assembly consists of the Variable Phase Turbine, generator and pump. The assembly is shown on the skid with the lube oil system and instrumentation junction box in figure 21. This unit was factory tested with nitrogen to verify mechanical performance before shipping to the site.

Figure 21 Rotating Assembly Mounted on Skid for Factory Testing

3.6 Power Plant Manufacturing

The VPT was manufactured and inspected. Balance of plant components were purchased from vendors and delivered to Energent. The VPT was assembled with the generator and main pump on a skid and tested at Energent.

The brine heat exchanger was a shell and tube construction. Two forty (40) foot long exchangers in series were utilized. Figure 22 shows the heat exchanger after downloading at the site. The Variable Phase Turbine assembly completed tests at the Energent factory with nitrogen. The unit was shipped with the inventory tank, compressor, boost pump and other auxiliary equipment to the Coso site.

The condenser was a plate and fin, compact design. The condenser is shown after downloading at the site in figure 23. The unit has gasket construction.to facilitate cleaning.

Figure 22 Shell and Tube Heat Exchanger at Site

Figure 23 Plate Fin Condenser

Figure 24 shows the turbine generator skid loaded on a truck for shipping to the site.
Figure 25 shows the inventory tank loaded for shipment. The equipment was received at the site and unloaded by Terra-Gen in the staging area.

Figure 24 Variable Phase Turbine Assembly Loaded for Shipping

Figure 25 Inventory tank loaded for Shipping

3.7 Power Plant Installation

The power plant was field erected at the geothermal site. Original planning was to factory build it with a plate fin (compact) heat exchanger. However, scale tests at the site resulted in extensive deposits in a test plate fin heat exchanger, rendering that design unusable. As a result the shell and tube geometry was adopted. The shell and tube unit was 40 feet in length and required long piping runs and a large inventory tank. As a result, field erection was required and increases in cost and installation time were experienced. A companion project at Coso is exploring methods to alleviate scale deposits in heat exchangers. This project when complete should enable compact heat exchangers to be used, resulting in factory assembly with reduced cost and time. In addition a goal of the heat exchanger program is to enable lower brine return temperatures, resulting in increased energy extraction from geothermal resources.

Installation of the cooling water system and the brine supply system were completed by Terra-Gen. Figure 26 shows the cooling water pumps and outlet line next to the plant cooling tower. Two 100% pumps are provided to ensure uninterrupted operation of the Variable Phase turbine power plant. Figure 27 is a view of the cooling water lines attached to the condenser.

Construction of the brine supply piping is shown in figure 28. The completed lines leading to the heat exchanger are shown in figure 29. Figures 30 shows the electrical power conduit leading to the breaker room and the finished power line to the soft starter. Figure 31 is the turbine pedestal.

Figure 26 Cooling Water Pumps and Cooling Water Line

Figure 27 Cooling Water Lines and Condenser

Figure 28 Installation of Brine Lines

Figure 29 Brine Lines Leading to Heat Exchanger

Figure 30 Completed Electrical Connection to Soft Starter

Figure 31 Turbine Pedestal

The VPT power plant construction was completed. All high pressure welds were tested by ultrasound. The complete system was helium leak checked. Electrical and instrumentation loop checks were completed. Insulation was applied to the piping and components. Brine and cooling water systems were operated. Refrigerant was loaded into the system.

Figures 32 and 33 show views of the completed power plant. The existing steam-brine separator for the flash steam power plant was utilized as a source of the brine. The existing cooling tower was utilized as a source of cooling water for the refrigerant condenser. The existing breaker for the power plant auxiliary power system was utilized to deliver the power produced.

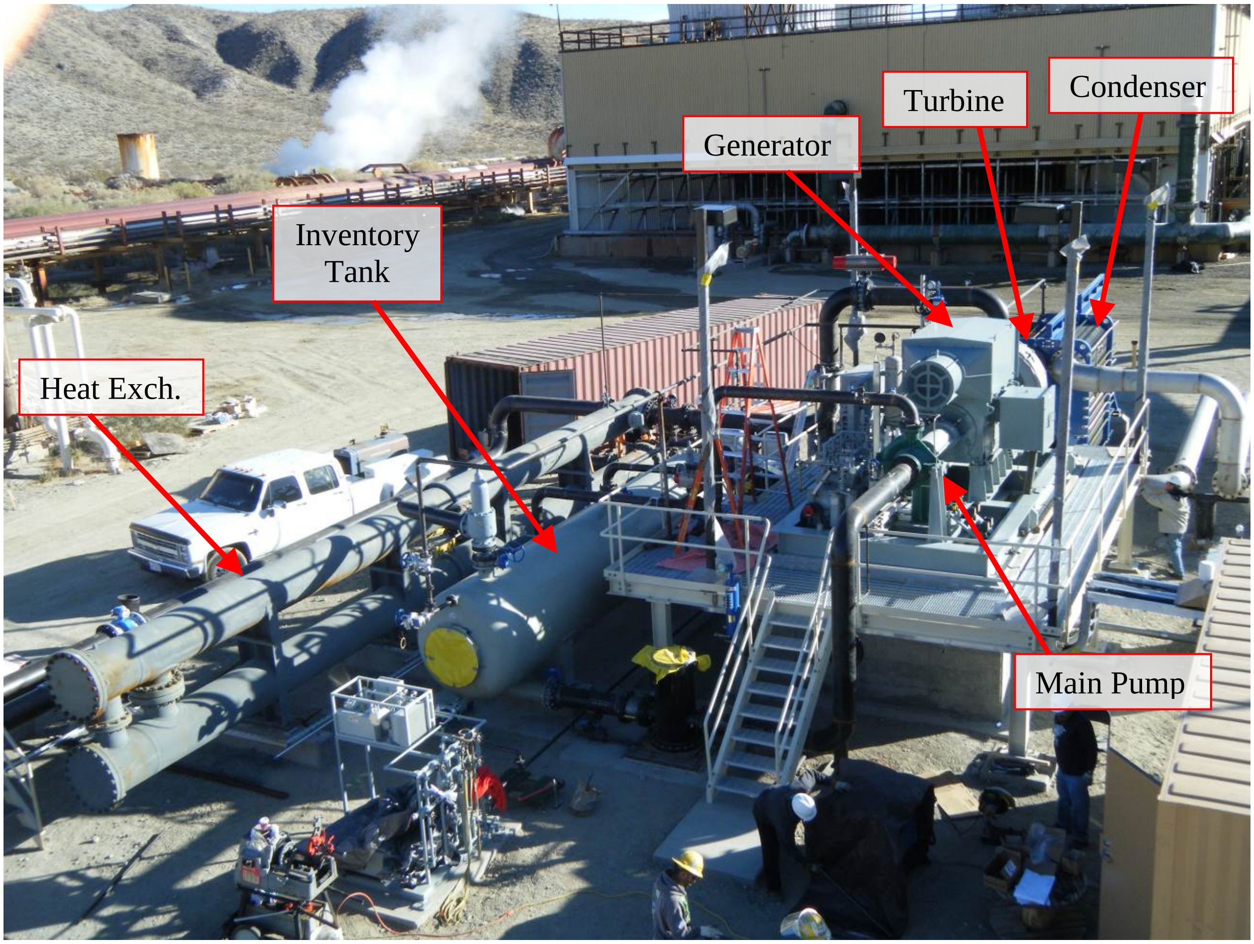


Figure 32 Overhead View of Completed 1 MW Power Plant at Coso Geothermal Resource

Figure 33 Side View of Completed 1 MW Plant at Coso Geothermal Resource

3.8 Power Plant Commissioning

Construction of the Variable Phase Cycle power plant was completed. Mechanical and electrical construction was completed. Refrigerant piping was completed, leak checked and insulated. Brine lines to and from the heat exchanger were installed. Cooling water lines to and from the condenser were installed. The control system was completed. Instrumentation installation and loop checks were completed.

The R134a was loaded into the system. Some minor leaks were corrected. The system pumps were started and refrigerant circulated in the system. Brine was admitted to the heat exchanger increasing the temperature of the refrigerant. The refrigerant was flashed in the nozzles causing turbine rotation. The turbine was synchronized with the grid. The initial power generated was 260 kW. A nuisance trip caused the unit to overspeed to 4,200 rpm. Analysis of the records showed the trip valve required about 4 seconds to close instead of the required <1 second.

The valve was returned to the vendor and a larger exhaust valve installed. The valve was reinstalled and testing continued. During the next startup a surge of brine from the Terra-gen separator occurred, resulting in an over speed trip before the grid connection was established.

During the tests high vibrations were recorded on the generator. A consultant was retained to determine the cause of the vibration. The generator was found to be unbalanced. The manufacturer had balanced the rotor without the cooling fan rotor installed on the shaft. The generator was re-balanced in the field and as a result the vibration levels were within the recommended limits.

During startup it was found that the brine flow rate was not steady due to the dynamics of the wells and separator. The surface gathering system consists of long runs of two-phase piping from the wells to a central separator. Brine collects in the piping until the steam pressure build up to a value high enough to dislodge the brine. This action results in a sudden surge of brine to the separator. The brine outlet valve from the separator opens suddenly to maintain a level in the vessel.

Flow rate could periodically surge by as much as 50% in the space of 10-15 seconds before returning to normal. The added energy to the heat exchanger resulted in temperature increases and flashing of the refrigerant in the heat exchanger. As a result, speed and power excursions were experienced by the turbine. Surges presented a significant problem to commissioning the power plant.

Several problems were caused by the unanticipated surges in brine flow to the VPC heat exchanger from the Terra-Gen separator system. The multiple starts and stops and thrust reversals caused damage to the thrust bearing, allowing contact of the rotor with the casing. This caused damage to the turbine rotor and shroud.

The sudden speed increases and torque induced by the surges contributed to mechanical damage of the VPT. The sudden torque caused loosening of set screws retaining a spacer which contacted the non-active face of the thrust bearing. The resulting high axial forces caused damage to the thrust bearing which in turn resulted in contact of the rotor with the stationary structure. This caused damage to the blades and shroud.

A replacement bearing and new rotor and shroud were manufactured and installed to complete commissioning.

Extensive modifications were made to the control system and another control valve was added to mitigate the effects of the surges. The modifications finally succeeded after several weeks of delay caused by the reprogramming and ordering and installation of the new valve. The new control algorithms will be useful for future waste brine energy recovery systems at Coso and elsewhere.

The power plant was started again. Stable flow conditions were realized and power generated to the grid. The power was kept at 230 kW for 20 minutes and a planned trip was executed. The turbine speed only reached 3,827 rpm.

Commissioning of the Variable Phase Cycle power plant (“VPC”) was completed. The system was operated for more than 200 hours with electric power sent to the grid. Very stable operation was demonstrated when the brine flow to the heat exchanger was steady. Figure 34 is a picture of the control screen during operation at 700 kWe net power. The maximum power tested was 796 kWe.

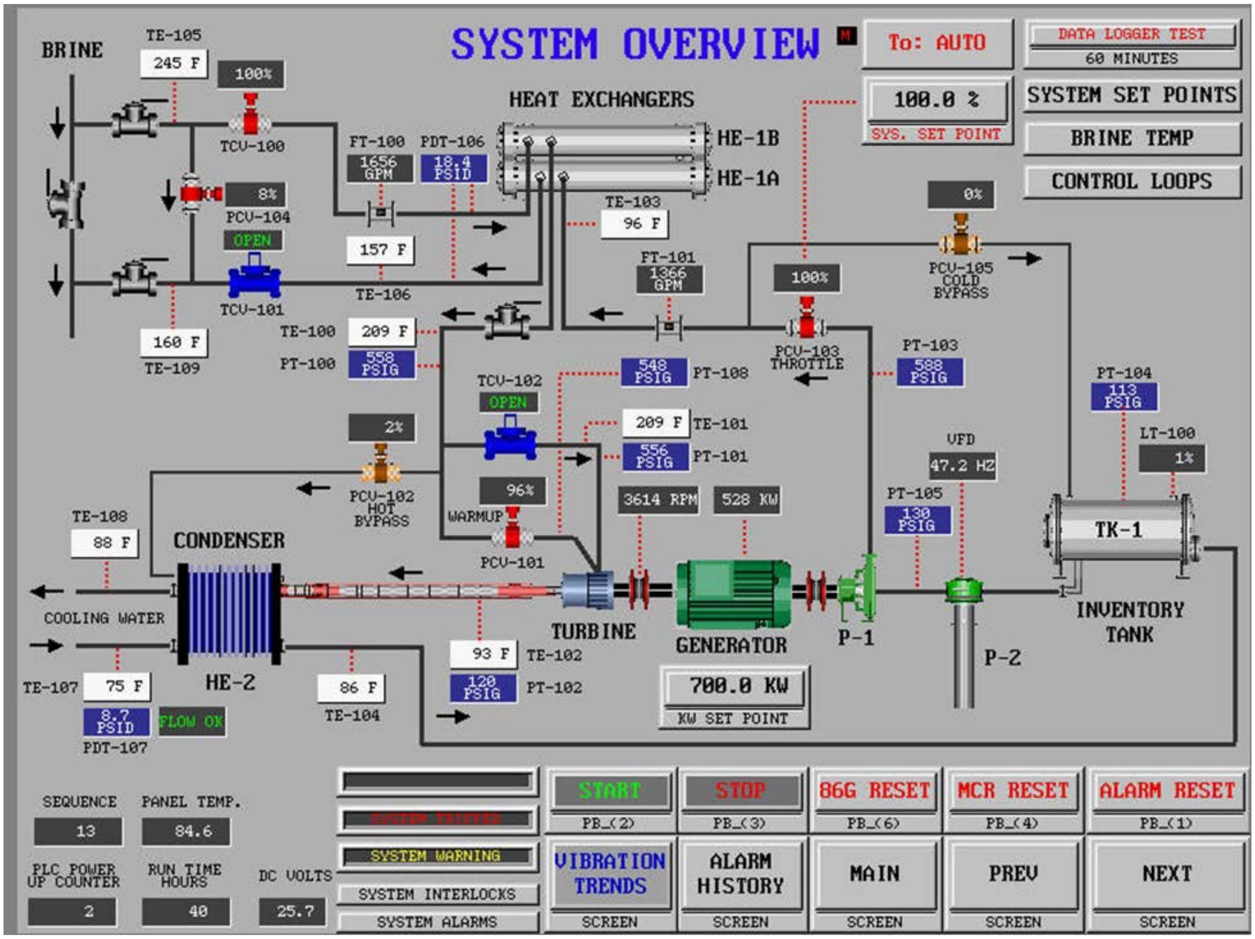


Figure 34 Control System Screen during Operation at 700 kW

3.9 Variable Phase Power Plant Performance

The VPC plant is designed as a 1MW net plant with approximately 1.4MW generated from the turbine and 300-400kW for the main pump. The pumping power is mostly regained by the pressure energy provided to the nozzles. The working fluid is R134a. A liquid-liquid heat exchanger reduces the heat of the brine from approximately 235 F to approximately 175F. The Variable Phase Cycle can reduce the brine temperature further without any pinch point limitations. However, the existing facility managers requested that the brine return temperature be restricted to this minimum value to avoid scaling of the re-injection wells. This return temperature is therefore higher than the temperature to which the Variable Phase Cycle is capable of reaching. With this limitation the Carnot efficiency is 9.1%.

The Variable Phase Turbine power train is shown during operation in figure 35.

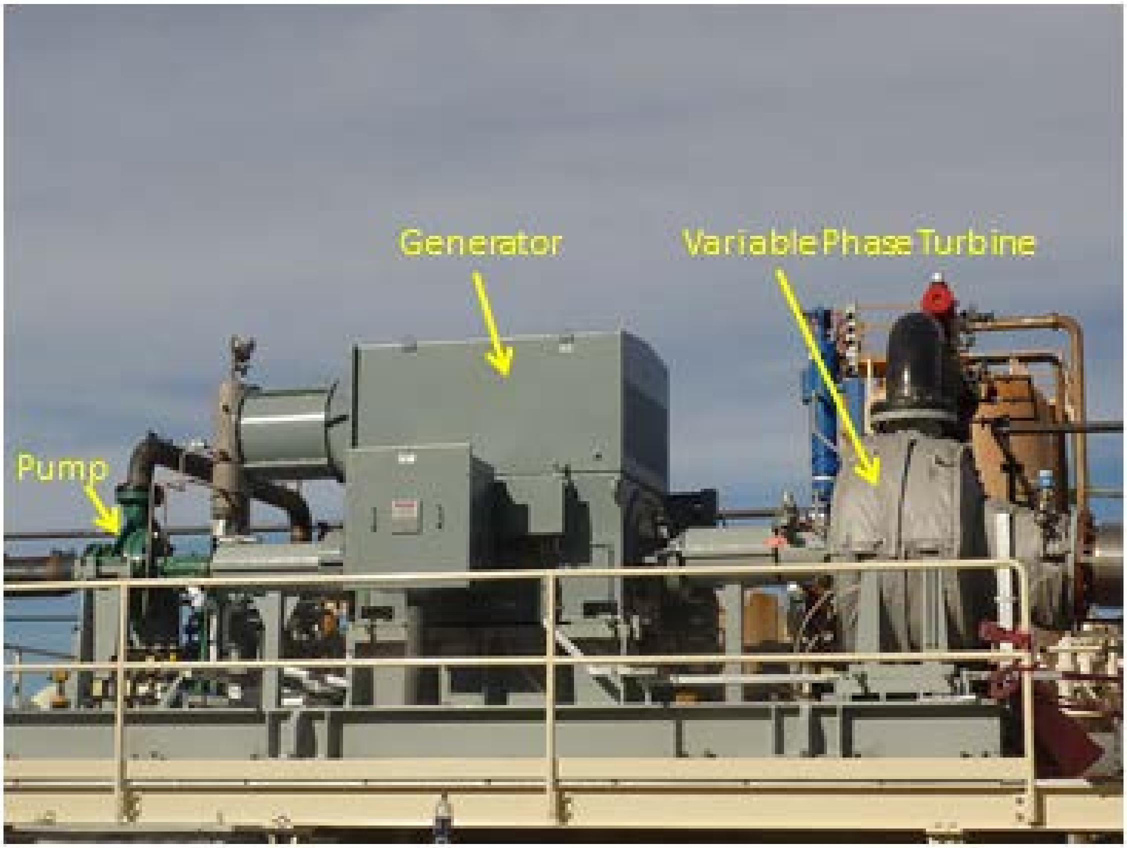


Figure 35 Variable Phase Turbine Power Train during Operation

Generation of 794 kWe into the grid was accomplished. The power generated agreed to within 1% with the power predicted by the design codes. The power generated was only limited by the brine flow available at the time and the permissible brine outlet temperature.

Table 3 is the system code prediction for the maximum power case. The input brine flow was corrected to produce the measured refrigerant flow of 822,000 lb/h (A heat balance confirmed that the indicated brine flow, measured with an electromagnetic flowmeter, was too high). For an indicated flow of 2252 gpm and an exit temperature of 180 F, the electric power measured was 794 kWe. The code prediction is 803 kWe with a shaft power of 1279 kWs, the extra power used to drive the pump. The predicted turbine isentropic efficiency is 76%. The measured shaft power was 1242 kWs. The isentropic turbine efficiency for the maximum power case was therefore 73%.

The refrigerant flow rate measured, 822,000 lb/h, was less than the design, 905,000 lb/h, leading to off-design performance.

Table 3 Code Prediction for Maximum Power Case

R134a	Turb Shaft	1279 kW	7/6/2014 9:59 DOE Coso Ax Turb Highest Power Point P
	CycleP:	803 kW	
	ProjectP:	753 kW	

Axial Turb File:	vptcycle	Unit set
Run AxTurb as:	DLL	English
OverWriteCheck:	No	v4.3

	Temperature	Pressure	Enthalpy	Density	Quality	Flow Rates			Efficiency	Power		Delta P	ΔHeight
	°F	psia	BTU/lb	lb/ft^3	-	lb/s	lb/hr	gpm/CFM	-	kW	BTU/lb	psi	ft
Heat Source Inlet	232.00	125.0	200.8	59.33	0	229.167	825,000	1734	-	-	-	4.3511	0
Heat Source Exit	180.00	120.6	146.4	60.60	0			1697	-	-	-	-	-
Heat-X Inlet	90.3	623.1	105.5	74.93	0	228.529	822,705	1369	-	-	-	34.5	0
Heat-X Exit	211.8	588.6	158.0	44.48	0	228.529	822,705	2306	-	Windage		4	0
Nozzle Inlet	211.4	582.9	158.0	44.281	0.000	228.529	822,705	2316	95.4%	5.9	Axturb	1.7	-
Nozzle Exit	88.1	115.5	151.3	3.779	0.629	228.529	822,705	3628		5.1	Theoretical	-	-
Rotor Exit	88.1	115.5	152.7	3.675	0.648	228.529	822,705	3731	79.2%	1278.9	5.3076	3.48	0
Turb Bypass Out	88.1	115.5	158.0	3.33	0.720	0	0	0	-	-	-	-	-
Outlet Mixed	88.1	115.5	152.7	3.675	0.648	228.529	822,705	3731	-	-	-	-	-
Condenser Exit	85.3	112.0	103.7	74.23	0	228.529	822,705	1362	-	-	-	1	-4
Boost Pump Inlet	85.3	113.1	103.7	74.24	0	228.529	822,705	1362	75.0%	-29.07	-0.1206	36.3	-
Boost Pump Exit	85.6	149.4	103.9	74.29	0	228.529	822,705	1381				0	10
Main Pump Inlet	85.6	144.2	103.9	74.28	0	228.529	822,705	1381	73.0%	-391.8	-1.6262	-	-
Main Pump Exit	90.3	623.1	105.5	74.93	0	228.529	822,705	1369				0	0
Cooling Water Inlet	66.9	29.2	35.1	62.33	0	714.567	2,572,440	5146	Pump Effic.	-49.9	-0.0662	14.504	-
Cooling Water Exit	82.6	14.7	50.7	62.19	0			5157	65.0%			-	-

HEAT EXCHANGER PARAMETERS

Heat-X Exit T (°F)	211.8
Max Heat-X Exit T (°F)	212.0
R134a Critical Pressure (psia)	588.7
HX Heat Xfer (MMBTU/hr)	43.1849
U (BTU/hr ft^2F)	70
Total Area (ft^2)	16393.8
LMTD (°F)	46.65
Avg TD (°F)	37.39
Min. ΔT pinch desired (°F)	20.00
Min. ΔT pinch actual (°F)	20.0

CONDENSER PARAMETERS

Condensing T. (°F)	86.2
Condensing P. (psia)	112.0
Subcooling (°F)	0.90
Cond. Heat Xfer (MMBTU/hr)	40.264
U (BTU/hr ft^2F)	282
Total Area (ft^2)	13006
LMTD (°F)	10.67
Avg TD (°F)	10.99
Min. ΔT pinch desired (°F)	5.5
Min. ΔT pinch actual (°F)	5.5

MISCELLANEOUS INPUTS

HeatSource fluid	Water
	Choices
Type of cycle?	Liquid
Vary HXExitT to get HX pinch?	Fluid
Vary Cond. T. to get pinch?	Yes
Main Pump on Turbine Shaft?	Yes
Use Boost Pump?	Yes
	Efficiencies
Generator Efficiency	94.0%
Rotor Efficiency Method	AxTurb
	Other Parameters
Generator Flow Fraction	0.0%
Rotor Speed (rpm)	3600
u/c	0.505
Number of Nozzles	10
Nozzle Angle (deg)	20
Nozzle Length (in.)	12.0

OTHER WORKING CYCLE INFORMATION

Required R134a flow rate		228.53	lb/s
Heat rate to working fluid		11996	BTU/s
Heat-X Exit Pipe ID		8	in
Heat-X Exit Pipe Velocity		14.72	ft/s
Nozzle Exit Enthalpy (Isentropic)		150.97	BTU/lb
Nozzle Exit Velocity (Noz Code)		578.92	ft/s
Nozzle Exit Diameter		1.369	in
Nozzle Exit Vapor Flowrate		143.86	lb/s
Nozzle Exit Liquid Flowrate		84.67	lb/s
Mean Line Rotor Diameter		1.46	ft
Nozzle Rotor Coverage Angle		26.399	°
Boost Pump Inlet NPSHA		5.2	ft
Pump NPSHA/Stages	64.31	ft	1
Pump RPM / Ns / Eff	3550	784.4	75.7%
Generator Power Dissipation		72.776	BTU/s
Gross Shaft Power Rate		3.7543	kW/(lb/s)
Gross Electric P (Turb&Mpump)		834	kW
Cycle Efficiency		6.8%	
Plant Net Thermal/Exergy Effic.		6.3%	2.0%
Power Rate (kW/(MMBTU/hr)		18.59	

The brine flow was varied to determine further off-design points. Figure 36 shows the variation of power with brine flow. The power ranged from 100 kWe to 794 kWe as the indicated brine flow ranged from 600 gpm to 2252 gpm.

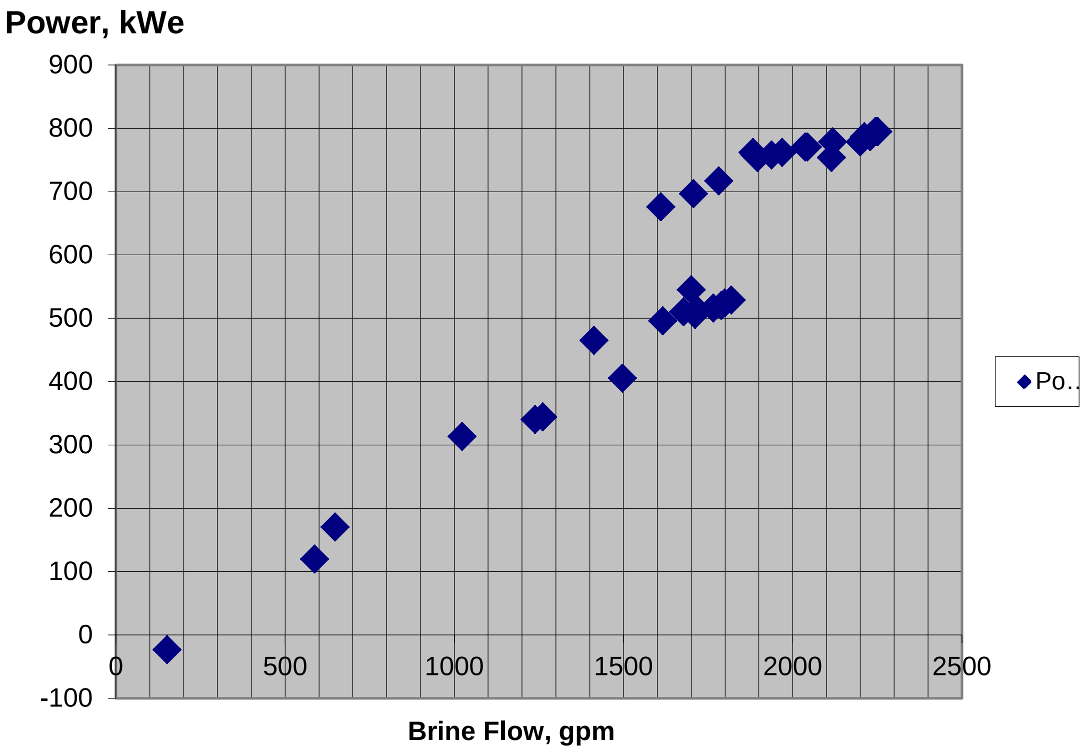


Figure 36 Variation of VPT Power with Indicated Brine Flow Rate

Currently the brine flow has declined to about 800-1000 gpm. As shown in the chart the power generated at this flow is in the range of 200-300 kWe. The low power is a result of limiting the heat exchanger outlet temperature to 175 F to avoid scaling. The power plant is not designed for long term operation at these low power levels. The concurrent heat exchanger scaling project is evaluating methods to avoid scaling at lower temperatures. If successful the power could be increased by extracting more energy from the brine.

To determine the effect of heat exchanger outlet power production a series of tests were conducted to lower the outlet temperature by increasing refrigerant flow rate for short periods of time (~8 hours). The results are shown in figure 37. As the heat exchanger outlet temperature was decreased from 180 F to 150 F the power increased from 183 kWe to 546 kWe.

Extrapolation of the results to 125 F, the goal of the heat exchanger program, indicates about 800 kWe could be realized even at the low brine flow rates.

Figure 38 plots the power produced versus the indicated brine heat input. The maximum power point is also shown for reference. This figure illustrates the influence of lowering the heat exchanger exit temperature to increase the heat input to the power conversion cycle. It is one of the benefits of the Variable Phase Cycle. For low temperature geothermal resources with low dissolved solids a substantial power increase can be achieved.

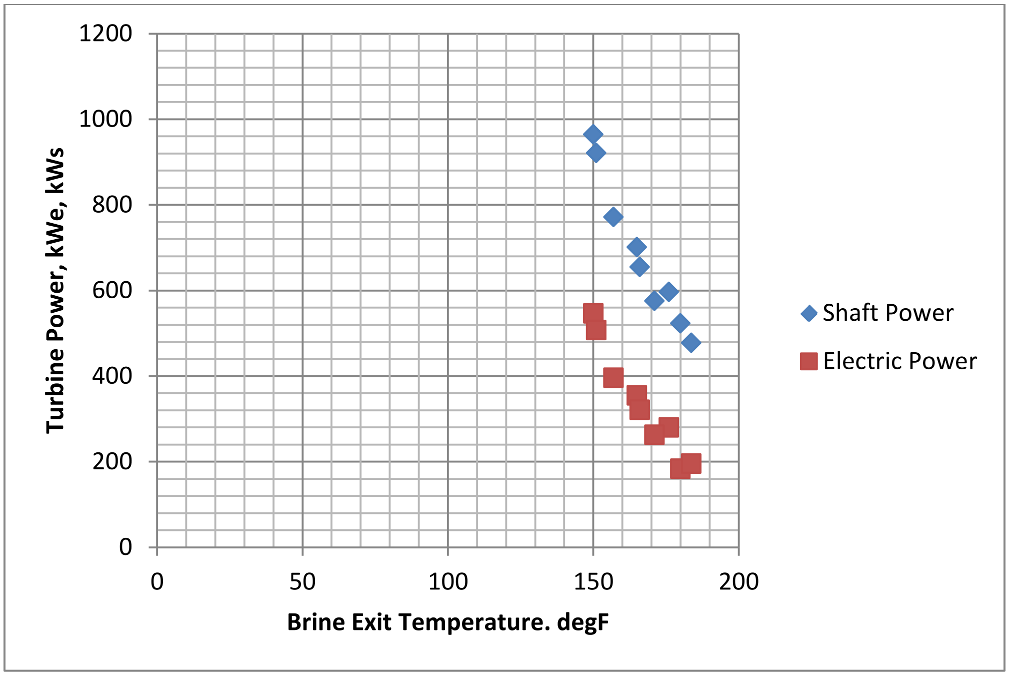


Figure 37 Turbine Electric and Shaft Power versus Brine Temperature Leaving Heat Exchanger

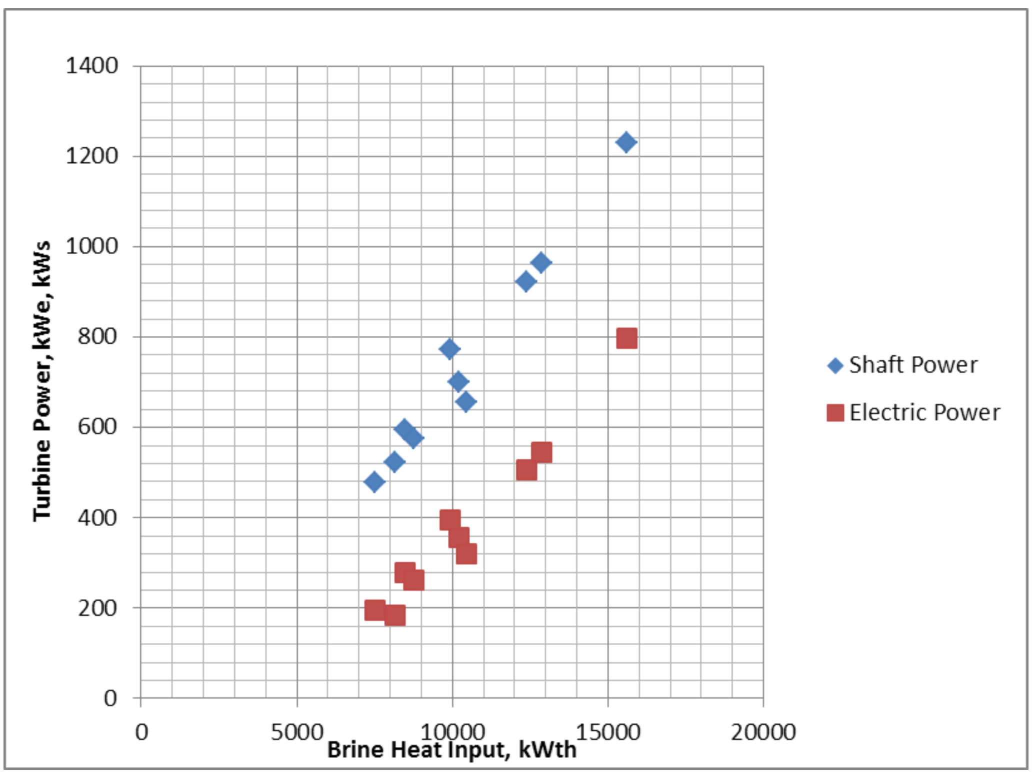


Figure 38 Turbine Electric and Shaft Power versus Indicated Brine Heat Input

The tests at higher exit temperatures were achieved by throttling the refrigerant flow. As shown in figure 39 the throttling produced two-phase flow at the inlet to the turbine, rather than flashing from liquid to two-phase inside the nozzle. The lowest temperature, 150F, enabled liquid with no vapor to be supplied to the nozzle. Limiting the flow rate to limit the temperature to 175 F to 185F resulted in vapor quality of 60-70%. The ability of the Variable Phase Turbine to operate stably with vapor at the inlet was demonstrated. This result confirms the descriptive name given is appropriate. The axial impulse design can also be used for vapor turbines as confirmed by a hermetic VPT provided for a ORC waste heat system.

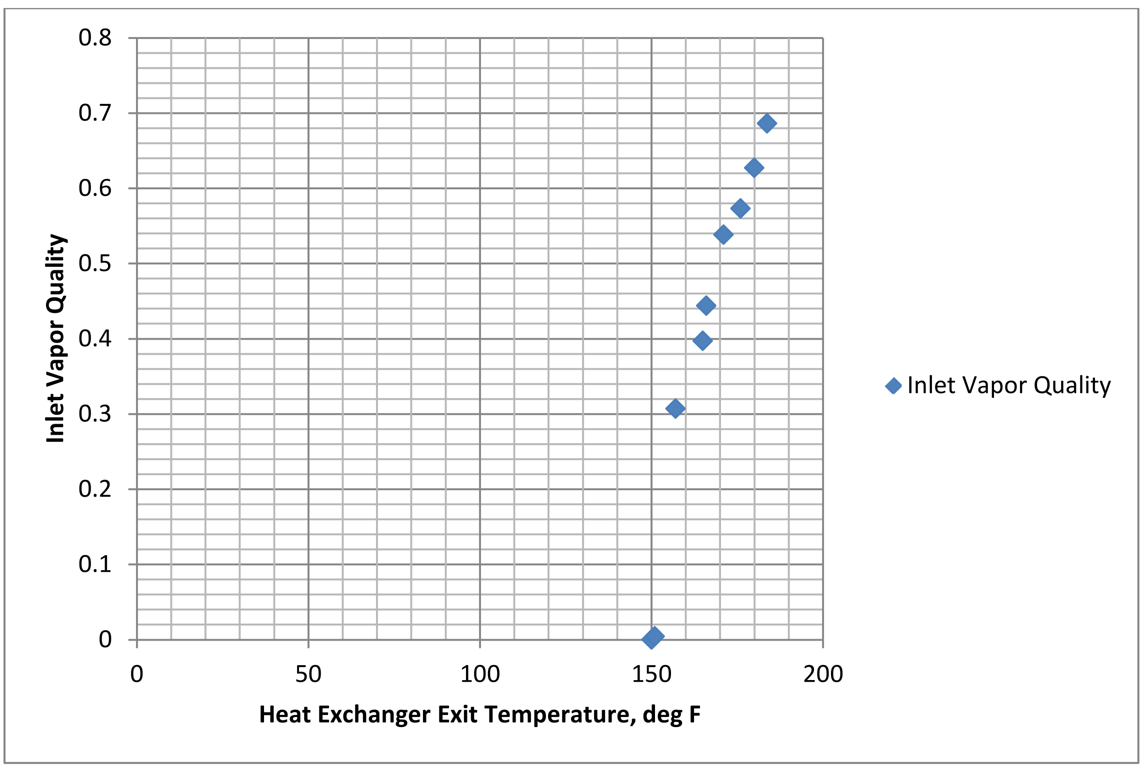


Figure 39 Inlet Vapor Quality versus Heat Exchanger Exit Temperature

3.10 Power Plant Operation

Operation of the power plant was smooth and controllable when the brine flow was stable. However, unanticipated brine surges from the host separator station were experienced. Increases of as much as 50% in a 15 second duration occurred. The rapid increase in heat input to the refrigerant caused rapid temperature and pressure increases which led to rapid increases in the speed of the turbine, or in the power if the generator was connected to the grid.

Extensive modifications were made to the control system and another control valve was added to mitigate the effects of the surges. The modifications succeeded after several weeks of delay caused by the reprogramming and ordering and installation of the new valve. The new control algorithms will be useful for future waste brine energy recovery systems at Coso and elsewhere.

After the modifications stable, although slowly fluctuating, flow conditions were realized and power generated to the grid.

Commissioning of the Variable Phase Cycle power plant ("VPC") was completed. The system was operated for more than 200 hours with electric power sent to the grid. Very stable operation was demonstrated when the brine flow to the heat exchanger was steady.

4.0 Applications of Technology Demonstrated by the Project

4.1 Geothermal Power Plants

The major application of the Variable Phase Turbine is generation of power from low temperature geothermal resources. The advantages of low cost and simplicity (no boiler, no separator, and no gearbox) are in addition to maximizing the power production.

The technology has application to surface power plants for Enhanced Geothermal Resources. The additional power produced can effectively leverage downward the capital cost per megawatt for the expensive infrastructure investment. In addition the Variable Phase Turbine has been tested for supercritical CO_2 expansions should that fluid be used for heat extraction from Enhanced Geothermal Resources.

Currently three low temperature geothermal projects are under development in Japan. Interest has also been expressed by developers in the U.S. and Italy.

4.2 Waste Heat Power Systems

Another Variable Phase Cycle system has been designed and supplied for a waste heat recovery system. Fundamentally, waste heat recovery is similar to geothermal in the temperatures typically experienced and the power size ranges used.

In this case an engineering and shipbuilding firm wished to recover heat energy from the propulsion engine of large container transport vessels. For these engines a turbocharger compresses air to feed into a two-stroke diesel engine. The air must be cooled before being used in the diesel engine—a task which is normally accomplished using an intercooler and a seawater cooling medium. Instead, a Variable Phase Cycle is installed in place of the intercooler to reduce the air temperature to the diesel engine.

Because the temperatures are somewhat higher R245fa is used instead of R134a. R134a reaches high pressure at higher temperatures (in this case around 200C) and so R245fa is a more appropriate fluid as it has similar properties except at higher temperatures and lower pressures.

This Variable Phase Cycle application generates approximately 150kW depending on exact conditions. The test engine size is 10,000HP (~7.5MW) and was run at full capacity for qualifying the Variable Phase Turbine. For this application the main refrigerant pump is run off of a VFD. The very tight space requirements made putting the main pump on the shaft unfeasible. The electricity generated will be used to power electrical loads on board the ship, reducing the amount of diesel used to power the ship.

Figure 40 shows the Variable Phase Turbine during operation. Figure 41 is a picture of the ship engine with the waste heat system installed.

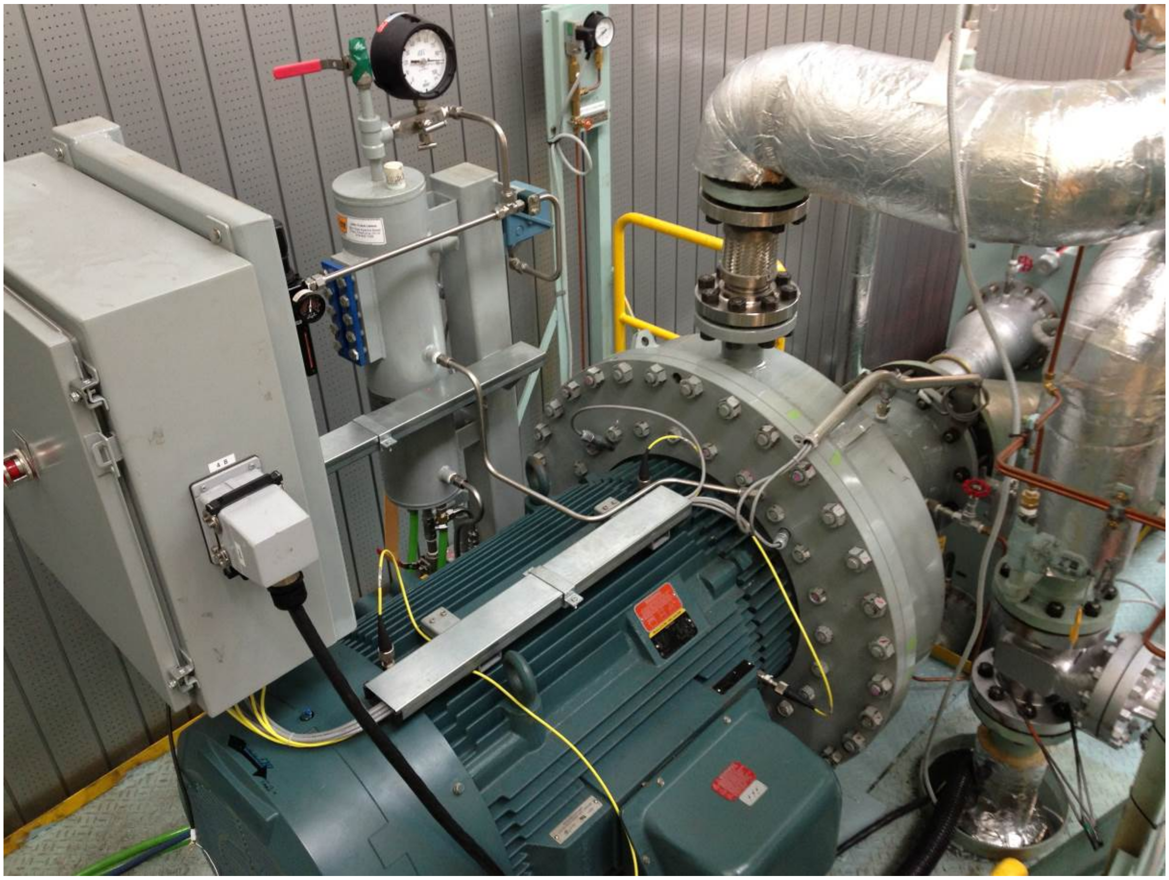


Figure 40 Variable Phase Turbine Operating in Waste Heat Recovery System

Figure 41 Ship Engine with Variable Phase Cycle Waste Heat Power System

4.3 Two-Phase Industrial Power Generation

A great many industrial applications exist for Variable Phase Turbines The Variable Phase Turbine (VPT) is a new turbine which can efficiently convert the energy in variable phase flows to power. Some examples of variable phase flows include gas and liquid mixtures, flashing liquids and supercritical or saturated liquids that expand into the dry gas region.

Identified applications include industrial refrigeration processes such as LNG, ethylene, and ammonia production. Replacement of the function of two-phase Joule-Thomson valves in these and other refrigeration processes by the VPT will produce power from energy previously wasted in the pressure letdown. The removal of energy from the process by generating power, instead of frictional heating in a valve, can produce more refrigeration, resulting in a greater product yield. For the processes above, improvement in the output of from 5-15% was identified.

For example, for a retrofit LNG case analyzed it was assumed that the plant design is well balanced. When the plant is running at its design capacity, all its subsystems are at their respective design capacities. Then, when the Mixed Refrigerant system capacity is raised by 5% due to the VPT application, the capacities of the propane system and the front-end preparation must also be raised by the same amount to supply the additional natural gas feed to the cold box. This can be accomplished by applying the VPTs in the propane system and the front end. This not only generates the 5% more capacity for the Mixed Refrigerant requirement but also has extra capacity remaining in these areas to produce additional 0.94% of LNG. Thus the overall LNG production capacity can be increased by 6.5%.

In addition to the increase in production, Variable Phase Turbines generate a total of 4.7 megawatts of power which can offset the main compressor power requirement.

Application of Variable Phase Turbines to ethylene production and to methanol production has also been studied. For an ethylene plant with production of 2.4 million metric tons per year Variable Phase Turbines would produce 2 megawatts of electric power and increase production by 4.6 % (101,000 MT/yr). For methanol production 5.4 megawatts can be generated and a steam savings of 15,000 lb/h would be realized.

Other refrigeration applications include air separation plants, ammonia production and petroleum refining processes.

5.0 Technology Transfer Activities

5.1 Publications

Welch, P., and Boyle, P. New Turbines to Enable Efficient Geothermal Power Plants, GRC Transactions, Vol. 33, 2009

Welch, P., Boyle, P., Giron, M., and Sells, M., Construction and Startup of Low Temperature Geothermal Power Plants, GRC Transactions, Vol. 35, 2011

Boyle, P., Hays, L., Kaupert, K., and Welch, P., Performance of Variable Phase Cycle in Geothermal and Waste Heat Recovery Applications, GRC Transactions, Vol. 37, 2013

Kaupert, K., and Hays, L., Flashing Liquid Expanders for LNG, ConocoPhillips LNG Technology Conference, 2010 and 2012, Houston Texas

Hays, L., and Zhang, G., Maximizing the Power from Geothermal Resources and Waste Heat, International Conference on Sustainable and Clean Energy, Dalian, China, 2011

5.2 Displays

Turbomachinery Symposium, 40^{th}, 41^{st} and 42^{nd} , George Brown Convention Center, Houston Texas

5.3 Website

www.energent.net

5.4 Licensing Agreements

Mitsui Engineering & Shipbuilding Co. Ltd. - Variable Phase Turbine for Geothermal Power and Waste Heat Recovery in Japan

ConocoPhillips - Variable Phase Turbine for LNG Production Worldwide

Shanghai Shenghe New Energy Resources Technology and Development Ltd, - Variable Phase Turbine and Euler Turbine for Power Production in China

Chevron – Power Recovery Separating Turbine for Power Recovery in Amine Systems Worldwide

ACD LLC – Variable Phase Liquid Turbine for Air Separation Plants Worldwide

6.0 Computer Code

6.1 System Analysis

Analysis and design of the power plant was carried out with the Variable Phase Cycle system code. The code inputs are shown below:

6.1.1 System Code Inputs

<u>Heat Source</u>

Type (liquid, gas, or two-phase); temperature; flowrate, permissible heat exchanger outlet temperature

<u>Variable Phase Cycle</u>

Type (liquid, gas, supercritical, two-phase); working fluid (e.g., R-134A, R245fa; Isobutane,etc.)

<u>Heat Exchangers</u>

Minimum approach temperature difference for heat exchanger and condenser; cooling source for condenser (water or air, inlet and outlet temperature); estimated values of overall heat transfer coefficient, U, for heat exchanger and condenser

<u>Variable Phase Turbine</u>

Nozzle length; number of nozzles; nozzle setting angle; turbine speed; turbine U/C or mean line diameter

Pumps

On turbine shaft or electric motor driver; boost pump or not; efficiency algorithm or specified efficiency

Piping System Pressure Drops

Specified pressure drops in valves and piping

6.1.2 System Code Output

The system code performs a heat balance to match the working fluid conditions to the heat source for the specified heat exchanger process inputs and approach temperatures. The system code calls up the two-phase nozzle code sub-routine and calculates the two-phase nozzle performance and geometry for the specified inputs of nozzle length and number. The system code subsequently uses the turbine input parameters to calculate the turbine performance and geometry. The system code determines the turbine output power, the net VPC cycle power, taking into account the pumping requirements, and the estimated net power plant power including estimates of cooling tower fan power, cooling water pump power or, in the case of an air-cooled condenser, the air fan power requirements.

A cost algorithm is subsequently used the estimate the cost of the power plant

7.0 Impact of Results on Low Temperature Geothermal Power Generation

The demonstration was the first commercial scale operation of the Variable Phase Cycle (also known as the triangle cycle. Good turbine efficiency and controllable operation were established. The operation demonstrated the ability of the cycle to eliminate the pinch point limitations of organic Rankine cycles. For low temperature resources the potential exists to generate 20-40% more power from a given resource than with conventional ORC technology.

Implementation of the Variable Phase Cycle for low temperature resources can add more than 10 gigawatts to the nation's geothermal power production. The technology has already been demonstrated to add to the nation's export potential,

Conclusions and Recommendations

A new power cycle was successfully demonstrated at the commercial level for low temperature geothermal power production. This power cycle can increase power production by 20-40% from low temperature resources, leveraging the capital cost of the entire project. Application of the Variable Phase Cycle (aka Triangle Cycle) depends upon a high efficiency two-phase expander. The Variable Phase Turbine was successfully demonstrated for this application and as well as for waste heat applications.

The current demonstration site has encountered declining brine flow rate resulting in part load operation of 20%. Improvement of the power output at the present site is possible if scale control methods are successful in a companion program. The current power plant should be maintained in a standby condition with periodic operation until scale control methods can be verified and implemented on the power plant heat exchanger.

Parallel to this effort a low temperature resource with low scaling potential should be sought to install a duplicate of the power plant to fully realize the potential of the new cycle..

References

DiPippo, R, (2007) Ideal Thermal Efficiency for Geothermal Binary Plants, Geothermics 36 (2007), pp276-285.

Stiedel, R.F., Brown, K.A., & Pankow, D.H., (1983) *The Empirical Modeling of a Lysholm Screw Expander*, Proceedings of the Intersociety Energy Conversion Engineering Conference, Orlando, FL, USA: 1983.

Appendix
Power Plant P&ID and Operation

The power plant process and instrumentation diagram ("P&ID") is provided in figure A-1. A description of the process design and operation follows.

1 Process description

1.1 Process Overview: The power system is a Variable Phase Cycle (VPC) and uses R134a as the working fluid. A refrigerant main pump, P-1, is fed by a boost pump, P-2, and provides the pressurized flow of liquid to the shell-and-tube heat exchanger, HE-1, which uses hot geothermal brine (water) to heat the liquid. The pressurized liquid, two-phase, or supercritical fluid (depending on the conditions of the heat source) then flows to the turbine, VPT-1. The low pressure refrigerant leaving the turbine enters the condenser vapor header. The condenser is a plate-and-frame heat exchanger that uses cooling tower water to condense the refrigerant. The liquid that is condensed gravity drains to the inventory tank which feeds the boost pump, and subsequently main pump, to close the power cycle.

1.2 Process Details – Inventory Tank –TK-1: The inventory tank, TK-1, stores the refrigerant in the system during maintenance and provides adequate refrigerant surplus capacity for operating under widely varying heat exchanger conditions. The refrigerant boost pump, P-2, draws from the bottom of the inventory tank.

1.3 Process Details – Refrigerant Boost Pump – P-2: The refrigerant boost pump, P-2, is a variable-speed, vertical, multi-stage, canned pump. The suction of the first-stage impeller is at the bottom of the pump, approximately 12' 6" below grade – this is the low point in the entire system. The purpose of this pump is to provide adequate NPSH to the main pump, P-1.

1.4 Process Details – Refrigerant Main Pump – P-1: The refrigerant main pump, P-1, supplies refrigerant to the heat exchanger, HE-1, and is coupled to the turbine-generator shaft. During startup, the pump naturally ramps up in speed as the turbine speed increases once warm refrigerant starts expanding in the turbine. Once interconnect speed has been reached, the pump operates at a fixed speed. To adjust the head-flow curve, the main throttle valve, PCV-113, is used to regulate the flow of refrigerant, FT-101, supplied by the pump.

The pump throttle valve located immediately downstream of the pump, PCV-103, is used to prevent backflow – which can occur during trip conditions or during startup before interconnection – and to keep the pressure between the pump outlet and PCV-103, PT-103, below an upper limit. FT-101 has a minimum control value based on the speed of P-1 that is connected to the position of PCV-103 to prevent dead-heading the pump. However, during upset or trip conditions, PCV-103 will close to prevent backflow through the pump even if the flow measured at FT-101 is below the limit.

1.5 Process Details – Heat Exchanger – HE-1A & HE-1B: The heat exchanger is shell-and-tube type with 2 identical heat exchangers in series. It is designed to be nearly counter-current for optimum heat exchange. The brine flows through the tubes (so that the ends can be opened periodically and the tubes cleaned of silica build-up. The heat exchangers are configured such that the flow path of the refrigerant is always upwards, to not trap any refrigerant vapor, so the system is capable of an operating mode where there is two-phase flow at the exit of the heat exchanger. The temperature and pressure of the refrigerant leaving the heat exchanger are measured by TE-100 and PT-100, respectively. The brine temperature leaving the heat exchanger is measured by TE-106 and is required to be kept at or above 175 degF to limit silica dropout.
Maximum pressure in the Heat Exchanger sector (the area between PCV-103 and PCV-113/PCV-101/PCV-102) is limited by both the cold bypass valve (PCV-105) and the hot bypass valve (PCV-102). Of the two, PCV-102 has a higher set point so that the two valves do not fight each other.

1.6 Process Details – Variable Phase Turbine: Upstream of the turbine trip valve, TCV-102, and PCV-113 is the Warmup Throttle Valve, PCV-101. PCV-101 regulates flow to a single nozzle, providing ‘topping’ control, so that the remaining turbine nozzles can operate closer to design point by having a wider range of operating conditions under which PCV-113 can remain 100% open. The flow from PCV-113 feeds the 11 remaining circular converging-diverging nozzles that efficiently convert the high pressure, high temperature refrigerant into low pressure, high velocity, two-phase flow. This high velocity flow spins an axial impulse turbine.

1.7 Process Details – Condenser: The low pressure two-phase refrigerant leaving the turbine enters the condenser vapor header. The condenser is a plate-and-frame heat exchanger that uses cooling tower water to condense the refrigerant. The liquid that is condensed gravity drains to the inventory tank.
The turbine hot bypass exhaust, PCV-102, enters the opposite end of the condenser vapor header to limit disturbance to the turbine rotor.

2 Control Scheme

2.1 Brine Valve Overview: The brine loop has two control valves: the throttle valve, TCV-100, and the bypass valve, PCV-104. Prior to VPC startup, all of the brine is flowing through the fully open bypass valve and the throttle valve is closed. The brine reinjection well often has upward and downward surges in flow. Because of this, the primary control loop for the throttle valve is controlling on flow through the heat exchanger (fast acting). The bypass valve is positioned inversely of the position of the throttle valve so that the sum of the positions, in percentage, is always 100%.

2.2 Refrigerant Valve Overview: The refrigerant loop has five positionable valves: pump throttle valve PCV-103, cold bypass valve PCV-105, hot bypass valve PCV-102,

warmup throttle valve PCV-101, and throttle valve PCV-113.
Originally, PCV-103 controlled the refrigerant flow through the system, but the addition of PCV-113 made this unnecessary, so the role of PCV-103 is that of a check valve – to prevent backflow, which can occur in upset conditions as well as after a trip. This valve also is must not dead-head the main pump or cause too high of a pressure at the pump discharge (PT-103), so there is a minimum flow limit (FT-101) and an upper pressure limit.
PCV-105 and PCV-102 are both normally closed during regular operation. They serve to limit overpressure in the heat exchanger and relieve pressure after a trip. The set point of PCV-102 is higher than PCV-105, so that the cold bypass is opened first.
PCV-101 regulates flow to a single nozzle, providing 'topping' control, so that the remaining turbine nozzles can operate closer to design point by having a wider range of operating conditions under which PCV-113 can remain 100% open. The position of PCV-101 is chained with PCV-113 such that once PCV-113 is full open, PCV-101 will start to open for topping control.
PCV-113 controls the refrigerant flow to the set point. It also originally was intended to maintain the pressure in the heat exchanger to be 10 psi above saturation pressure, but early testing determined that this had little to no effect on performance and that operating in the two-phase regime was stable and acceptable.

2.3 Set Point Overview: The primary control values used in normal operation are: brine flow rate, brine heat exchanger exit temperature, and refrigerant flow rate.
Because the brine flow rate is based on the geothermal plant and reinjection well and is the independent variable, the flow rate is a moving set point that is periodically increased/decreased if there is a surplus/deficit of brine flow. During startup, the brine flow rate set point is ramped up slowly along a prescribed curve.
The brine heat exchanger exit temperature is required to be at or above 175F during normal operation. This set point is ramped up from 80F to 175F during startup based on the brine flow rate.
To achieve the brine heat exchanger exit temperature set point and to link the brine system and VPC system, the refrigerant flow rate set point is varied (slow control) based on the brine heat exchanger exit temperature.

3 Refrigerant throttle control valve, PCV-113

3.1 Purpose: The purpose of this valve is to control refrigerant flow rate, FT-101. As FT-101 rises above the set point, PCV-113 closes to reduce the flow rate. PCV-113 is operated on the same control loop as PCV-101 with PCV-113 controlling the first 90%.
3.2 Non-operational position: This valve is closed when the system is not operating
3.3 Startup position: This valve automatically opens during startup to achieve the refrigerant flow rate, FT-101, set point.
3.4 Shutdown position: This valve closes quickly under normal shutdown and trip scenarios
3.5 Fail position: Closed

4 Refrigerant warm-up control valve, PCV-101

4.1 Purpose: The purpose of this valve is to provide topping control for the refrigerant flow rate, FT-101. As FT-101 rises above the set point, PCV-113 closes to reduce the flow rate. PCV-101 is operated on the same control loop as PCV-113 with PCV-101 controlling the top 10%.
4.2 Non-operational position: This valve is closed when the system is not operating
4.3 Startup position: The control loop on this valve is overridden during startup and this valve automatically opens during startup to provide higher power at lower flows to reach interconnect speed faster.
4.4 Shutdown position: This valve closes quickly under normal shutdown and trip scenarios
4.5 Fail position: Closed

5 Brine throttle control valve, TCV-100

5.1 Purpose: The purpose of this valve is to control the brine flow rate, FT-100. As FT-100 rises above the set point, TCV-100 closes to reduce the flow rate. TCV-100 is operated on the same control loop as PCV-104 with the valves operating inversely of each other, namely the sum of the positions, in percentage, is always 100%.
5.2 Non-operational position: This valve is closed when the system is not operating
5.3 Startup position: This valve automatically opens during startup to achieve the brine flow rate, FT-100, set point.
5.4 Shutdown position: This valve closes quickly under normal shutdown and trip scenarios
5.5 Fail position: Closed

6 Brine bypass control valve, PCV-104

6.1 Purpose: The purpose of this valve is to control the brine flow rate, FT-100. As FT-100 rises above the set point, PCV-104 opens to reduce the flow rate to the heat exchanger. PCV-104 is operated on the same control loop as TCV-100 with the valves operating inversely of each other, namely the sum of the positions, in percentage, is always 100%.

6.2 Non-operational position: This valve is open when the system is not operating

6.3 Startup position: This valve automatically moves to a more closed position during startup to achieve the brine flow rate, FT-100, set point.

6.4 Shutdown position: This valve opens quickly under normal shutdown and trip scenarios

6.5 Fail position: Open

7 Refrigerant pump control valve, PCV-103

7.1 Purpose: The purpose of this valve is to prevent backflow through the main pump under upset conditions and trip conditions. The difference between PT-100 and PT-103 is measured to determine if there is potential for backflow.

7.2 Non-operational position: This valve is closed when the system is not operating

7.3 Startup position: This valve automatically opens on startup and remains at 100% during normal operation unless backflow is detected.

7.4 Shutdown position: This valve opens quickly under normal shutdown and trip scenarios to increase flow through the main pump, and therefore the drag on the pump, to slow the rotating assembly down as fast as possible and prevent overspeed. Once the speed of the rotating assembly has fallen below 3500 rpm, PCV-103 closes.

7.5 Fail position: Closed

8 Refrigerant cold bypass control valve, PCV-105

8.1 Purpose: The purpose of this valve is to limit maximum pressure and relieve pressure quickly in the heat exchanger region (based on PT-100).

8.2 Non-operational position: This valve is closed when the system is not operating

8.3 Startup position: This valve remains closed during startup and normal operation.

8.4 Shutdown position: This valve opens quickly under normal shutdown and trip scenarios to relieve the pressure from the heat exchanger section. Once the pressure has fallen below 550 psig, PCV-105 closes. PCV-105 also closes if the pressure in the inventory tank, PT-104, exceeds 130 psi or if TE-103 exceeds 110F, indicating warm refrigerant has reached the heat exchanger inlet and will expand at the inventory tank pressure. During normal operation, PCV-105 starts to open if PT-100 exceeds 665 psig.

8.5 Fail position: Closed

9 Refrigerant hot bypass control valve, PCV-102

9.1 Purpose: The purpose of this valve is to limit maximum pressure in the heat exchanger region (based on PT-100).

9.2 Non-operational position: This valve is closed when the system is not operating

9.3 Startup position: This valve remains closed during startup and normal operation.

9.4 Shutdown position: This valve opens quickly under normal shutdown and trip scenarios when PT-100 is above 600psig to relieve the pressure from the heat exchanger section. Once the pressure has fallen below 575 psig, PCV-102 closes. PCV-102 also closes if the pressure in the inventory tank, PT-104, exceeds 120 psi. During normal operation, PCV-102 starts to open if PT-100 exceeds 675 psig. PCV-102 will not open if the system trips due to loss of cooling water.

9.5 Fail position: Closed

10 Automatic Start Sequence

10.1 Description

Startup consists of moving refrigerant directly through the turbine, before reaching operating pressure and temperature and without passing through the bypass valves, letting the natural flow and heating of the refrigerant to spin up the turbine-generator-pump assembly to rated speed. This is achieved by turning on the refrigerant boost pump, closing the turbine bypass valves, opening the turbine trip valve and turbine warmup valve, enabling the turbine control valve to begin flow control, and incrementing the brine flow rate set point along a prescribed ramp. Over time, as the brine flow rate increases, the refrigerant in the heat exchanger gets hotter and increases the shaft power of the turbine, causing the rotating assembly to increase in speed and therefore increasing the refrigerant flow rate in the heat exchanger. As this process continues, the turbine will eventually reach rated speed. At the moment the turbine reaches rated speed, the generator breaker must close, connecting the generator to the Utility grid. If the breaker does not close, the turbine will continue to increase speed and must be tripped when reaching its overspeed limit. After breaker closure, power output of the generator will be a function of the energy produced by the expanding refrigerant. Once the breaker closes, the turbine warmup valve will resume normal flow control.

10.2 Start Sequence Notes

10.2.1 Note: Waiting to close the generator circuit breaker until the turbine is at rated speed, as opposed to closing the generator circuit breaker at zero speed, improves system reliability, reduces generator transient inrush current recovery time and imposes less stress on all components from the drive train components to circuit breakers and fuses.

10.2.2 Note: PT-1002 limit control and generator kW limit control will be performed by the hot water control algorithm (FICA-1001).

11 Automatic normal plant shutdown sequence

1. Operator selects the automatic plant shutdown sequence.
2. Stop brine flow to heat exchanger: Open PCV-104 / Close TCV-100
3. Wait 15 seconds (PCV-113 will naturally ramp closed as the brine exit temperature drops)
4. Open PCV-105 (Cold Bypass Valve)
5. Close TCV-102 (Trip Valve)
6. Wait 2 seconds

7. Disconnect Turbine from grid (open generator circuit breaker)
8. Close PCV-103 when rotating speed drops below 3500 rpm

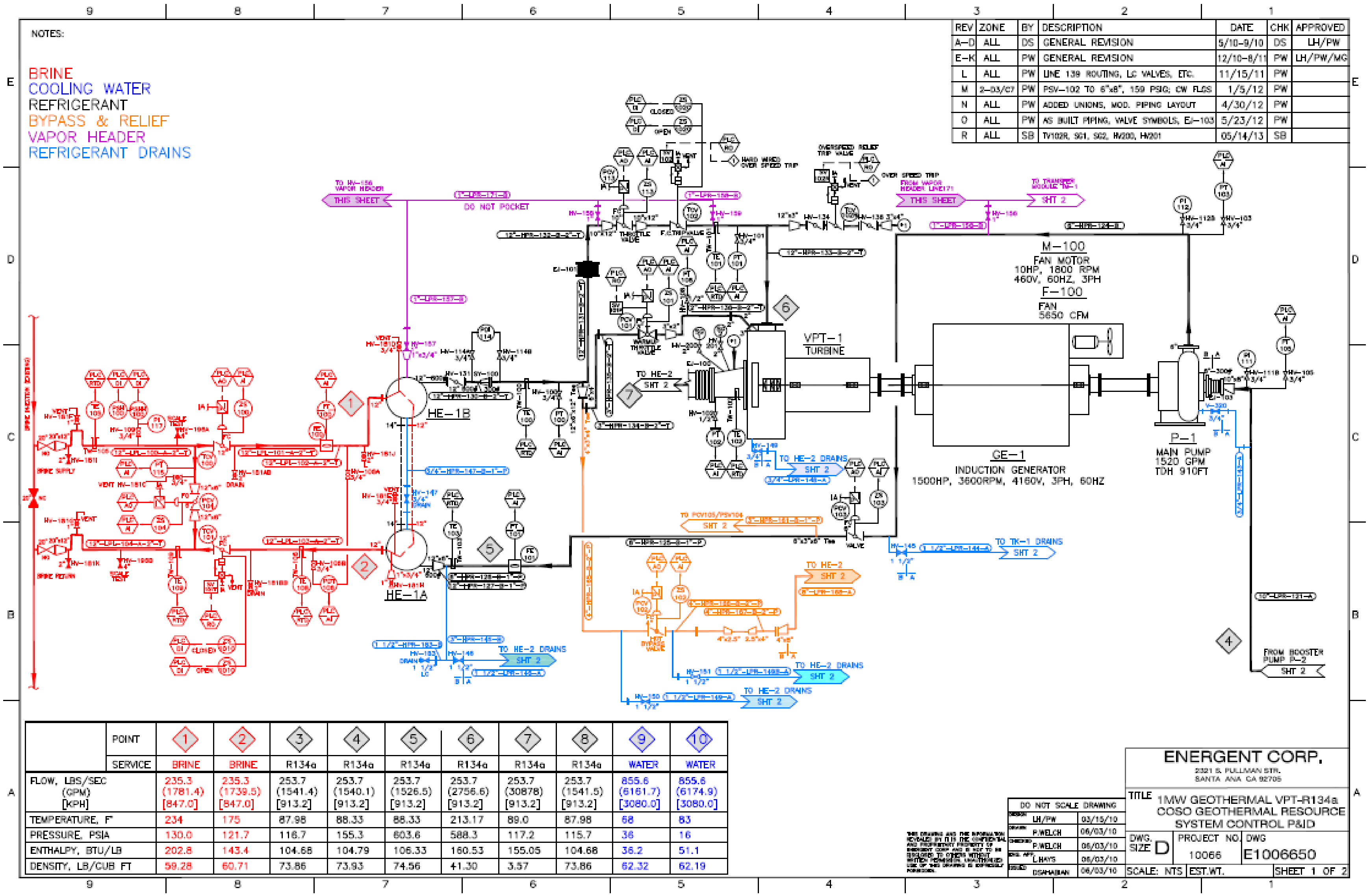
12 Emergency plant shutdown sequence

1. Close TCV-102 (Trip Valve) and PCV-101 (Warmup Throttle Valve)
2. Close PCV-113 (Throttle Control Valve)
3. Disconnect VPT-G from electrical grid by opening generator circuit breaker
4. Open PCV-105 (Cold Bypass Valve)
5. Stop brine flow to heat exchanger: Open PCV-104 / Close TCV-100
6. If PT-104 reaches 130 psig, close PCV-105 (Cold Bypass Valve)
7. Close PCV-103 when rotating speed drops below 3500 rpm
8. Wait 1 minute then turn off cooling water pumps and fans

13 Refrigerant Management during Trip

13.1 Problem: If hot and cold bypass valves are both fully opened after a trip, it is possible there will be a pressure buildup in the inventory tank, causing the relief valve to open. Also, if there is a loss of cooling water, opening the hot bypass will cause an overpressure of the inventory tank sector in 6 seconds.

13.2 Solution: In general, the hot bypass is relieving HOT refrigerant that expands to a factor of 16 when the pressure is reduced, so the cold bypass is a better solution to relieving the pressure in the heat exchanger region since the fluid does not expand much when the pressure is reduced. PCV-105 is sized to pass 180% of design flow, so it is more than sufficient to pass enough flow. Regardless of the method chosen, the high pressure section can remain at high pressure indefinitely, so the cold bypass valve should close once the pressure has dropped below 550psig (and the rotating assembly has dropped below 3500 rpm). Also, if the inventory tank pressure, PT-104 is over 130 psig, the bypass valves should close. Also, if TE-103 exceeds 110F, the cold bypass should close, as this indicates warm refrigerant has reached the heat exchanger inlet and will expand at the inventory tank pressure.



	POINT	1	2	3	4	5	6	7	8	9	10
	SERVICE	BRINE	BRINE	R134a	R134a	R134a	R134a	R134a	R134a	WATER	WATER
FLOW, LBS/SEC (GPM) [KPH]		235.3 (1781.4) [847.0]	235.3 (1739.5) [847.0]	253.7 (1541.4) [913.2]	253.7 (1540.1) [913.2]	253.7 (1526.5) [913.2]	253.7 (2756.6) [913.2]	253.7 (30878) [913.2]	253.7 (1541.5) [913.2]	855.6 (6161.7) [3080.0]	855.6 (6174.9) [3080.0]
TEMPERATURE, F		234	175	87.98	88.33	88.33	213.17	89.0	87.98	68	83
PRESSURE, PSIA		130.0	121.7	116.7	155.3	603.6	588.3	117.2	115.7	36	16
ENTHALPY, BTU/LB		202.8	143.4	104.68	104.79	106.33	160.53	155.05	104.68	36.2	51.1
DENSITY, LB/CUB FT		59.28	60.71	73.86	73.93	74.56	41.30	3.57	73.86	62.32	62.19

Figure A-1 Coso Variable Phase Cycle P&ID

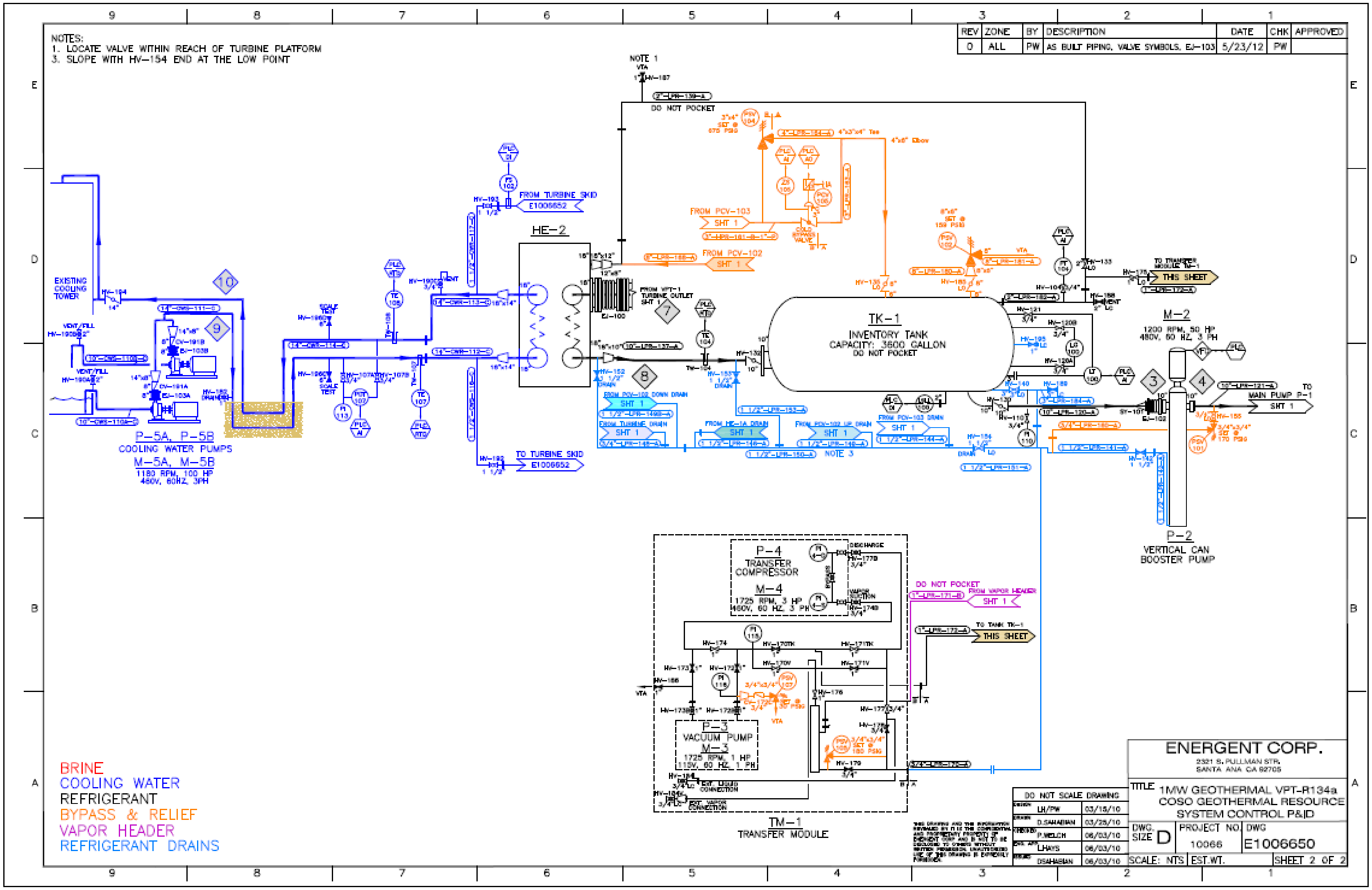


Figure A-1 (Cont.) Coso Variable Phase Cycle P&ID