Performance of Variable Phase Cycle in Geothermal and Waste Heat Recovery Applications

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Keywords

Variable Phase turbine, Flashing Liquid Expander, Triangular Flashing Cycle, Variable Phase cycle, organic Rankine cycle, ORC, binary cycle, geothermal utilization, low temperature geothermal, efficiency, thermodynamic cycle, axial impulse turbine, waste heat recovery

ABSTRACT

Previous information has been presented about a method of generating power from low-temperature resources using a cycle called the Variable Phase Cycle [Welch et al 2009, Welch et al 2010, Welch et al 2011) which is similar to the Triangular Flashing Cycle. A typical Variable Phase Cycle uses a single-component fluid which is pumped as a liquid to a heat exchanger where it remains a liquid close to saturation. This liquid then continues to a set of discrete converging-diverging nozzles where it flashes partially or completely to vapor which then passes through a turbine. This cycle uses an axial impulse Variable Phase Turbine (also known as a flashing liquid expander) which by design accepts any vapor fraction from 0-100%. The liquid/vapor is then passed on to a condenser where the cycle is completed. The Variable Phase cycle excels over Organic Rankine and steam cycles at lower temperatures where it can realize a 30-50% power advantage over other approaches.

In this paper, two applications of the Variable Phase cycle are discussed. The first application is a geothermal power plant operated in the geothermal field near Coso Junction, California. In this application, the brine separated from the wellheads is sent to the Variable Phase Cycle heat exchanger before being re-injected into the ground as per usual. This approach eliminates the need to drill a new geothermal well by using the sensible heat of the separated brine that was previously being wasted. The turbine is a 1 MW turbine which to date has produced 800 kW. The working fluid in this system is R134a.

The second application is a waste heat recovery application. Most diesel engines use a turbocharger to improve power and efficiency. The air output of the turbocharger needs to be cooled before going into the diesel engine and a convenient way to do this and improve overall efficiency at the same time is to use a heat-recovery cycle to cool the air. A Variable Phase Cycle was implemented in lieu of the intercooler for a 10,000HP (7.5MW) diesel ship propulsion engine. The recovered heat was used to generate 150 kW of power using a Variable Phase Turbine (flashing liquid expander). The working fluid in this system is R245fa.

Introduction

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

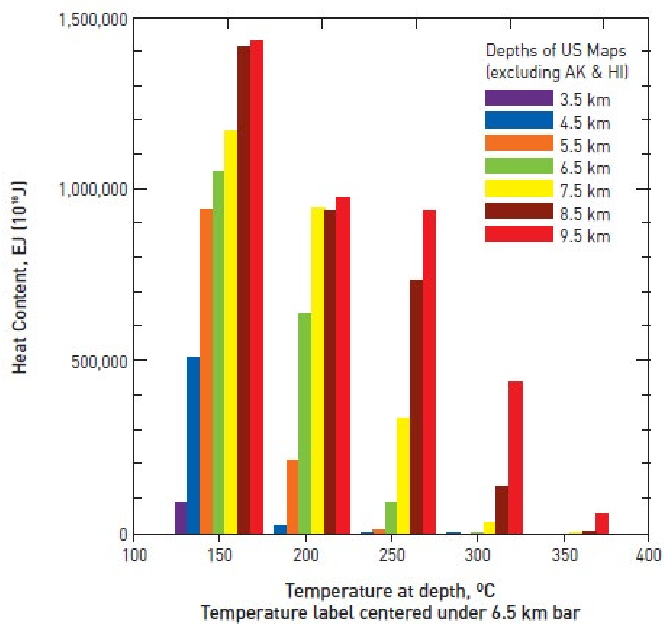

Figure 1. Estimated available heat at various depths for contiguous United States [MIT, 2006].

are generally far more resources at a lower temperature in a given area than there are high temperature resources [Massachusetts Institute of Technology, 2006].

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). Figure 1 estimates the total amount of heat at various depths and temperatures for the continental United States. From the table it is immediately obvious that the vast majority of land is below 200˚C in temperature at current realistic geothermal well depths.

Notice that for shallower and more accessible depths the total heat available drops by around an order of magnitude for every 50˚C 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 100-120˚C 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, 2006]. 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.

Finally, a great many industrial processes generate significant amounts of heat as a byproduct of their processes. An estimated 20-50% of industrial energy input is lost as heat with 60% of that heat loss being low-temperature heat, here defined as below 232˚C (450˚F) [U.S. DOE, 2008] Examples are too numerous to cover all but some of the largest include cement kilns, metal smelting plants, glass plants, boilers, incinerators and exhaust from conventional combustion-based power plants burning coal, methane, etc.

Background

In conventional Organic Rankine and Kalina cycles the temperature profiles in the heat exchanger gives 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 Kalina cycle 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.

To remove the pinch point temperature limitation a Variable Phase Cycle (Triangular Flashing Cycle) can be applied. Figure 2 shows a Variable Phase Cycle (Triangular Flashing Cycle) typical T-H diagram.

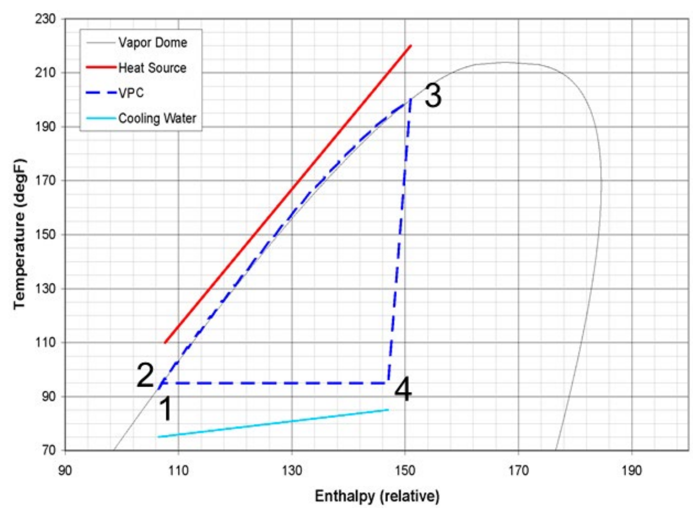

Figure 2. T-H diagram for the Variable Phase Cycle (“VPC”) also known as Triangular Flashing Cycle.

Note 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 3 shows a typical process

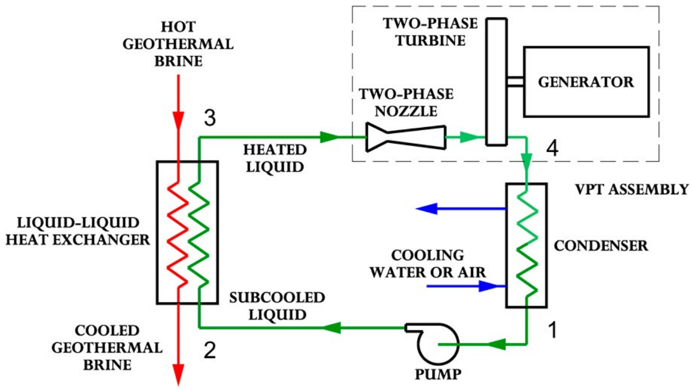

Figure 3. Generalized process diagram for a Variable Phase Cycle.

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 2.

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 (Schuller, 2011) with continuous liquid impact. Additionally the presence of liquid has been found to reduce performance and cause excessive vibrations.
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 4) 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.

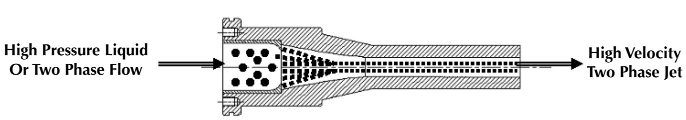

Figure 4. Cross-section of the two-phase converging-diverging nozzle.

The Variable Phase Turbine itself is a single-stage axial impulse turbine. A shroud is typical but not required. 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 3). 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 5). Provided that the liquid droplet size is small, a large fraction of the droplets follow the gas phase in turning through the blade passages. Less liquid actually impacts the blades—which is desirable because such impacts and the resulting flow of the collected liquid results in a frictional loss. The liquid on the blades does transfer some energy to the moving blade and is flung off the blades at the exit.

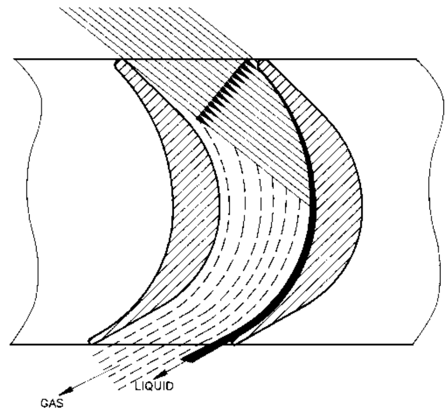

Figure 5. Axial impulse Variable Phasae Turbine for handling two-phase flow.

Geothermal VPC

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 110C. 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 (Figure 6). During startup it was found that the brine flow rate was not stable due to the dynamics of the wells and separator. Flow rate would 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. The problem was overcome by a combination of feed forward logic in the control system and the addition of a control valve. The surges no longer impede the functioning of the system during startup or operation.

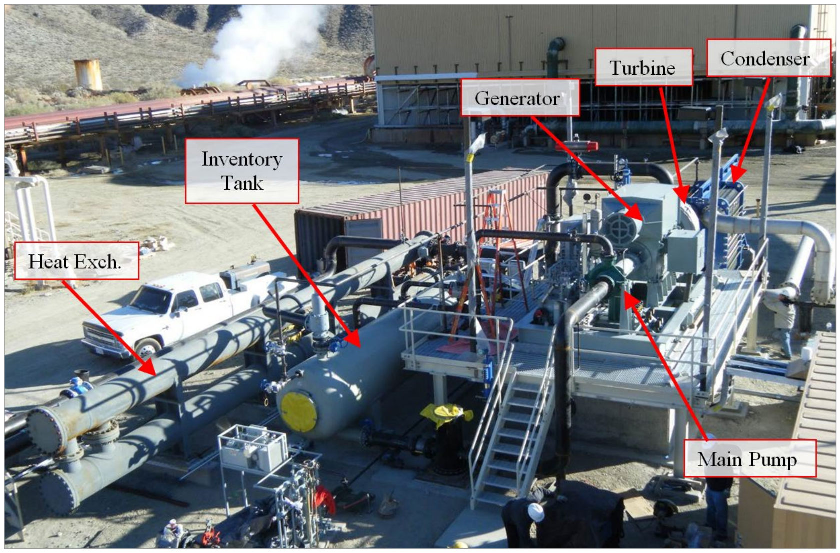

Figure 6. 1MW (net) Variable Phase Cycle geothermal power plant near Coso Junction, California.

The VPC plant is designed as a 1MW net plant with approximately 1.4MW generated from the turbine and 300-400kW in main pump losses. The total pumping power is partially 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 110˚C to approximately 75˚C. The brine temperature could be reduced further but 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 heat exchanger is a shell and tube heat exchanger. This design was selected for ease of cleaning and because the facilities managers already had experience cleaning shell and tube brine heat exchangers. The condenser is a plate and frame heat exchanger. The turbine speed is 3600 rpm and the generator is also a 3600 rpm induction generator so no gearbox is necessary. The generator is double-ended with the turbine on one side and the main refrigerant pump on the other side with one continuous shaft. This approach was selected for efficiency reasons.

In Table 1 representative data from the system is presented. At the time of the data point the net electrical output is 795kW including electrical losses and deducting pumping losses from the main pump that is on the same shaft as the turbine. Turbine shaft power is calculated at 1387kW for these conditions. The refrigerant main pump is by far the largest parasitic loss but as stated above a large fraction of the pump power is recovered as pressure feeding the nozzles. . There are still other parasitic losses to consider such as cooling tower fan and pump losses (unknown), refrigerant boost pump losses (~37kW), generator cooling fan losses (~7.4kW), oil pump losses (~0.5kW), etc. Pressures and temperatures are values reported from sensors and converted from Imperial units. Flows are converted from volume flow rates to mass flow rates based on the fluid density at those conditions. Enthalpies are evaluated in a variety of ways that are discussed further in the Calculation Methodology section.

Table 1. System conditions from a representative data point for geothermal power plant. Electrical power output at this point is 795kW. Power loss from refrigerant pump is already deducted.

	Brine in	Brine out	HX inlet	Turbine inlet	Turbine outlet	Cooling water in	Cooling water out
Flow kg/s	135.1	-	105.5	-	-	-	-
Temp C	111.9	83.3	29.6	99.8	28.4	19.7	29.2
Press. Barg	12.1	12.1	41.4	39.0	7.1	-	-
Enthalpy kJ/kg	466.7	348.5	241.2	375.8	-	-	-

In Figure 7, actual power output is compared to expected power output. The measured power output is the electrical power output of the generator which is the turbine mechanical shaft power less the pump mechanical shaft power losses as they are on the same shaft and then reduced by the generator efficiency. The method for arriving at the Predicted Power values is discussed in the Calculation Methodology section. Significant scatter is present in Figure 7 due to the difficulties in accurately determining flow quality and enthalpy in a two-phase flow and from the errors discussed in the Calculation Error section. Despite the scatter of the plot the system output is very stable for a stable set of brine and cooling water conditions. The overall plant power output matches the predicted plant power output for the given brine and cooling water conditions even if the exact refrigerant conditions diverge from predicted refrigerant conditions for the reasons discussed in the Calculation Error section.

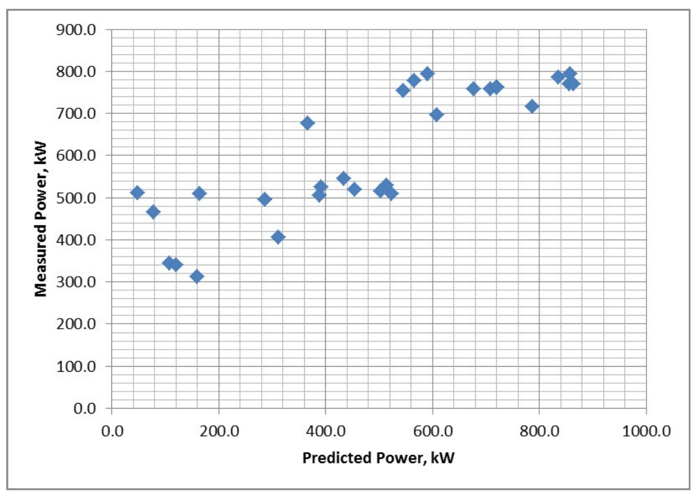

Figure 7. Chart of measured vs. predicted generator electrical output. All power values include losses from the main refrigerant pump.

At this time the power output of the generator is limited to 800kW. This is not due to any fundamental problem but due to concerns about overspeed. It was found over a series of overspeed trip tests that the speed would likely exceed the 20% overspeed rating of the generator at full power. The system is currently artificially limited to 800kW while improvements to the trip methodology are considered and implemented. It is expected that the plant will reach the full 1MW once this issue has been properly addressed.

Calculation Methodology

Calculations with a fluid in or near the two-phase regime can be very challenging. The simple approach of using the pressure and temperature of the fluid in question to calculate enthalpy, entropy, quality, etc. can not be used due to sensor error. If a fluid is saturated and there is even a small error in the pressure or temperature it can cause very large changes in calculations due to the fact that the reported values incorrectly condense or vaporize the entire flow. When it is known that the fluid is well sub-cooled then the temperature and pressure may be used. If that fluid is then run through a heat exchanger then the fluid exit properties--in particular fluid quality--can be estimated based on the enthalpy changes to the secondary fluid (in this case geothermal brine). These calculations are generally more reliable because there is no condensing or vaporizing occurring on the geothermal brine side so small errors in brine sensors do not result in large enthalpy changes.

For these calculations it was assumed that the fluid at the inlet of the nozzle was a saturated liquid. Based on the nozzle inlet conditions and the condenser pressure and temperature the flow and velocity was calculated based on proprietary software. This software in turn was derived from calculations done by the Jet Propulsion Laboratory on the behavior of two-phase flow [Elliot 1982, Elliot et al 1968, Elliot et al 1966]. With flow and spouting velocity it is possible to calculate turbine shaft power. Because the pump is also attached to the generator shaft the pumping losses must be deducted from the turbine power. As the pump is pumping liquid refrigerant this calculation is generally much more accurate and can be done using flow and enthalpy change across the pump. This net power value is then multiplied by generator efficiency to arrive at a predicted power output value. All fluid property calculations were performed with Refprop 9.

Calculation Error

Errors in the pressure and temperature readings will cause errors in calculations. Every effort was made to select and install accurate sensors on this project but even high quality sensors may experience some inaccuracies due to turbulence in flow, poor sensor placement within the flow and scaling of the sensors for geothermal brine readings. For this project it is suspected that the brine flow meter—and possibly other brine sensors--is being scaled due to flow readings slowly diverging from expected values over time and due to the knowledge that the heat exchanger is scaling as evidenced by an increasing pressure drop across the heat exchanger over time. At this time it is not possible to confirm the error in the sensor or estimate how much this affects the readings. For this reason the predicted power calculations did not rely on the brine flow meter for estimating heat input into the refrigerant other than as a general confirmation.

For brine calculations pure water was assumed. Geothermal brine contains dissolved minerals which will affect the density and specific heat of the water. For the types of minerals dissolved in the geothermal brine it is expected that the density would increase and the specific heat would decrease which would tend to minimize error from using pure water in calculations. Never the less the exact values for density and specific heat are unknown and therefore not used for the final turbine power calculations.

Calculations could be made significantly more accurate if the vapor quality of the refrigerant flow were known but at this time no reliable and accurate method for measuring this value has been determined.

Waste Heat Recovery VPC

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 additional heat 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 by 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 200˚C) 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.

The waste heat recovery Variable Phase Turbine output 150kW electrical power during qualification although the turbine and generator are sized in excess of 200kW. The power output was simply limited by the amount of heat available which was inadequate to reach maximum turbine power. The turbine and generator spin at 3600rpm with no gearbox necessary. The heat exchanger is a two-pass shell and tube heat exchanger with hot compressed air heating liquid R245fa. Unlike the geothermal application there is no limitation on the return temperature of the heat source flow (hot air) and in fact a low temperature is desirable. The return temperature is therefore dictated only by the temperature of the cooling flow to the condenser which in this case is sea water. Ambient conditions strongly affect both the hot air and cooling sea water temperatures but a common range would be 190-210˚C for the hot air inlet and 8-20˚C for the sea water temperatures with the hot air return temperature generally being 40˚C above the cooling water inlet temperature. Carnot efficiencies are again dependent on ambient conditions but will be in the roughly 30% range.

In Table 2 representative data from the system is presented. At the time of the data point the net electrical output is 151kW. This does not include system parasitics such as refrigerant main pump (~34kW), cooling water pump losses (unknown), refrigerant boost pump losses (19kW), etc. There is no generator cooling fan or oil pump on this system. Pressures and temperatures and flows are values reported from sensors. Enthalpies are evaluated in a variety of ways that are discussed further in the Calculation Methodology section.

Table 2. System conditions from a representative data point for waste heat recovery power plant. Electrical power output at this point is 151kW.

	Hot air in	Hot air out	HX inlet	HX outlet	Turbine outlet	Cooling water in	Cooling water out
Flow kg/s	11.8	-	9.0	-	-	-	-
Temp C	192.3	55.3	13.7	125.7	21.7	9.1	14.0
Press. Barg	3.0	3.0	21.8	21.1	0.1	-	-
Enthalpy kJ/kg	467.7	328.4	218.3	401.8	-	-	-

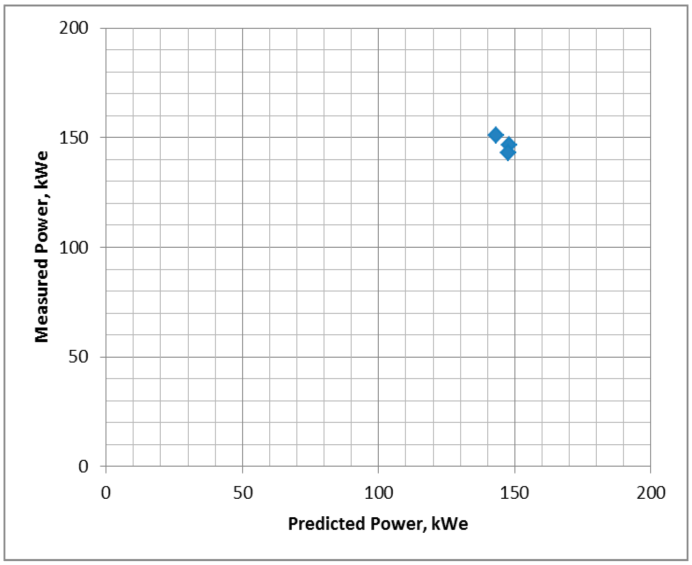

Figure 8. Predicted vs. measured power of waste heat recovery VPT. All data taken at maximum diesel engine load.

Figure 8 shows measured vs. predicted power for the waste heat recovery VPT. Limited information on air flow rates was available at conditions other than maximum diesel engine load which is why the graph is lightly populated. Agreement is good between measured and predicted values at maximum load.

Calculation Methodology

The calculations were performed in a similar way to the geothermal calculations and also face the same challenges of reliable numbers when at or near the two-phase regime. The refrigerant inlet conditions to the heat exchanger can be directly calculated because it is well sub-cooled. The heat exchanger outlet conditions for the refrigerant were not always performed using the reported temperature and pressure because it was known that in some conditions vapor was being formed in the heat exchanger. Note that this is not a problem for the Variable Phase Cycle or Turbine because the nozzle and turbine can accept any combination of vapor and liquid at the nozzle inlet, nozzle outlet and turbine. In fact a small amount of vapor can actually be helpful for certain scenarios such as a very low condenser pressure or excessive nozzle cross-sectional area. Calculations were performed on the hot air using the standard air mixture available in Refprop. The calculated heat loss of the air across the heat exchanger was assumed to all go into the refrigerant from which the refrigerant outlet conditions were calculated. Heat exchanger outlet conditions were compared to values calculated by proprietary system modeling software that uses a heat balancing method and turbine/nozzle geometry and the outlet conditions of the heat exchanger were found to be in good agreement. For the data reported in Table 2 both methods agreed that there should be approximately 15% vapor content at the heat exchanger exit.

Hot air flow rate was a value reported by the diesel engine operator and was not measured directly by the system. Pumping parasitic lost was determined by pump motor name plate values and, for the main pump which used a VFD, the speed at which the pump was operating. All air and fluid property calculations were performed with Refprop 9.

Calculation Error

Similar to the geothermal system errors in the pressure and temperature readings will cause errors in calculations. Every effort was made to select and install accurate sensors on this project but even high quality sensors may experience some inaccuracies due to turbulence in flow, poor sensor placement within the flow. Scaling is not an issue for this application.

For hot air calculations the default mixture in Refprop was used. This mixture assumes a specific mixture of dry gases. Due to the very close proximity to the ocean it is expected that there is some water vapor content in the air and possibly trace amounts of other, unknown impurities. This moisture content is not expected to significantly affect the air enthalpy calculations. Direct measurement of the air flow rate was not possible. Reported air flow rates are supplied by the diesel engine operator and are based on their own calculations that indirectly report air mass flow based on such parameters as air temperature, turbocharger speed, etc. No error margin was supplied with these reported air flow rates. Heat balance calculations across all data sets suggest that the supplied air flow calculations may have an error margin of as much as 10%.

Calculations could be made significantly more accurate if the vapor quality of the refrigerant flow were known but at this time no reliable and accurate method for measuring this value has been determined. Calculations could also be made significantly more reliable if some method of measuring air mass flow rate were implemented. Such a method would be out of the scope of control of the Variable Phase Cycle system suppliers and would have to be implemented by the diesel engine operators.

Summary

Two commercial applications of the Variable Phase Cycle and Turbine have been presented in this paper. As far as we are aware these are the first and largest such applications of this cycle technology in the world at nominal design power values of >1MW and 200kW. Performance including efficiency and power output agreed with projected values based on heat source properties. There is still some difficulty in calculating performance from reported sensor values when in or near the two-phase regime because it is necessary to know the quality or enthalpy of the fluid. Further commercial application of this technology seems both technically feasible and economically viable.

References

DiPippo, R. (2007). Ideal Thermal Efficiency for Geothermal Binary Plants. Geothermics 36 , 276-285.

Elliot, D. G. (1982). Theory and Tests of Two-Phase Turbines. JPL Publications 81-105 (DOE/ER-10614-1) . Pasadena, California: Jet Propulsion Laboratory.

Elliot, D. G., & Weinberg, E. (1968). Acceleration of Liquid in Two-Phase Nozzles. JPL Publication 32-987 . Pasadena, California.

Elliot, D. G., Ceromo, D. J., & Weinberg, E. (1966). Liquid-Metal MHD Power Conversion. Space Power Engineering Systems Engineering , 1275-1297.

Massachusetts Institute of Technology. (2006). The Future of Geothermal Energy. U.S. Department of Energy.

Schuller, S. (2011). Best Exergy Point for ORC. Geothermal Resources Council Conference Transactions , 35, 1343-1350.

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.

U.S. Department of Energy. (2008). Waste Heat Recovery: Technology and Opportunity in U.S. Industry. BCS, Incorporated.

Welch, P., & Boyle, P. (2009). New Turbines to Enable Efficient Geothermal Power Plants. Geothermal Resources Council Transactions , 33, 765-772.

Welch, P., Boyle, P., Giron, M., & Sells, M. (2011). Construction and Startup of Low Temperature Geothermal Power Plants. Geothermal Resources Council Transactions , 35, 1351-1356.

Welch, P., Boyle, P., Sells, M., & Murillo, I. (2010). Performance of New Turbines for Geothermal Power Plants. Geothermal Resources Council Transactions , 34, 1091-1096.