BINARY POWER PLANT MODELLING AND SENSITIVITY ANALYSIS FOR ELECTRICITY GENERATION FROM AN ENHANCED GEOTHERMAL SYSTEM

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ABSTRACT

A numerical model of a geothermal well field and sub-critical binary Organic Rankine Cycle (ORC) power plant is presented. This model allows the performance of different plant heat rejection systems (i.e. dry air cooling, evaporative wet cooling, once through wet cooling) to be analysed and the equipment to be sized. The model uses NIST REFPROP as source of thermodynamic data, which allows for the performance of commercially available binary working fluids to be readily evaluated and compared.

Of note the model is designed to allow the plant 'off-design' power output to be evaluated across a range of geothermal resource and ambient temperatures in order to estimate the annualised net generation for prospective geothermal projects.

As part of a broader feasibility study for power generation from an Enhanced Geothermal System (EGS) site in Europe, the model is used to compare and contrast performance from different heat rejection options for a brine fed binary power plant. A direct heat cascade use has also been considered.

A sensitivity analysis is presented for each plant option including variation of ambient conditions, monthly ambient data, geothermal brine flow, and geothermal brine temperature. The performance of each plant design for alternative working fluids is presented, as are the water supply requirements. The environmental impacts and site requirements for major equipment are discussed.

Capital and operational cost estimates (+/- 40%, 2012 basis) have been derived as an input to financial modelling. The return on investment for different options is presented to help inform option evaluation of power plant technologies.

1. PROCESS DESCRIPTION

1.1. Overview of Organic Rankine Cycle

Organic Rankine Cycle (ORC) power plants have been used for many years in the power generation industry to utilize relatively low temperature heat sources for electricity generation. ORC plants for geothermal applications have been in operation since 1952 (DiPippo, 2012).

The so-called binary cycle enables the use of lower temperature geothermal resources due to the integration of a secondary working fluid with low boiling point, rather than water. The working fluids are typically hydrocarbons (e.g. propane, butane, pentane) or refrigerant fluids (e.g. 1,1,1,2-tetrafluroethane/R-134a or 1,1,1,3,3-pentafluoropropane/R-245fa). Mixtures of fluids may also be considered. The selection of a particular working fluid for a given application is a function of: cycle feasibility and performance; environmental; health and safety; and operational considerations.

There are various configurations of ORC technology that use different arrangements of heat exchangers and turbines (Kaplan, 2007). The configurations are based on the temperature and state of the geothermal fluid available from the reservoir. However all have the same basic principle of operation. A schematic is shown in Figure 11 (see attachment in rear).

The geothermal fluid is extracted from the reservoir and passes through a series of heat exchangers to transfer the geothermal heat to the working fluid. The heat exchangers typically consist of a pre-heater and vaporiser. The working fluid, which is contained in a closed loop cycle, is heated to saturation point in the pre-heater and then evaporates in the vaporiser to produce a high pressure vapour. The high pressure working fluid vapour is expanded through a turbine generator set to produce electricity. The low pressure turbine exhaust vapour is then condensed. This can be achieved by using a heat rejection system. Heat rejection systems are described in more detail in section 2.2. The condensed working fluid is then returned via a feed pump to the pre-heater, and onto the vaporiser which then closes the cycle.

A recuperator is an optional heat exchanger located at the turbine exhaust. Superheated fluid exhausted from the turbine passes through this heat exchanger and uses sensible heat to preheat the working fluid. This process effectively matches the condenser de-superheating duty to preheating duty. The recuperator will introduce some thermodynamic loss to the cycle, but this is offset somewhat by reducing the condenser and preheater losses. The other consideration is that a recuperative cycle will have a reduced condenser heat load.

Another design choice of the cycle is working pressure; the working fluids typically used have a critical pressure low enough to consider supercritical cycle designs which can have an improved cycle performance over a sub-critical design. However, the majority of plants are of the sub-critical classification.

2. MODEL DESCRIPTION

2.1. Model Overview

A numerical model of a sub-critical ORC power plant and well flow/fluid collection and disposal systems has been developed. The model has been implemented in Microsoft Excel. It is designed to cover an entire binary power plant from feed zones in the production well(s) to the feed zones in the injection well(s) modelling the thermodynamic processes in the equipment in between.

The model uses the themodynamic correlations for different binary fluids, such as iso-pentane (Lemmon and Span, 2006), and these in turn use the industry standard NIST REFPROP (Lemmon, Huber, and McLinden, 2007) library.

Using the REFPROP library the model is able to represent the ORC and determine the following parameters for set 'design point' geothermal brine conditions (source and sink temperature, and mass flow rate), and ambient dry bulb temperature (DBT):

- Gross and Net Power generated from the power plant;
- Estimated parasitic loses from brine circulation and binary fluid circulation pumps, heat rejection system pumps, transformer losses and other auxiliary losses;
- Heat duty and indicative size of each heat exchanger;
- Heat duty, size, and water use requirements for the heat rejection system;

The model does not attempt to optimize the cycle efficiency for a given working fluid. Instead a working fluid is selected, based on a predetermined assessment of the critical temperature of the fluid relative to the geothermal source temperature.

The model is based on a user defined pinch-point temperature difference (PPTD) across the vaporizer heat-exchanger. This is shown in Figure 1 for iso-Pentane.

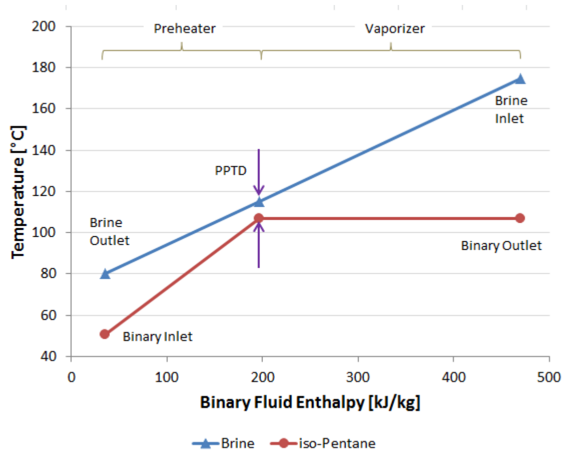

Figure 1: Iso-Pentane Heating and Brine Cooling Curves for Primary Heat Exchanger

Using this set pinch-point the cycle is then separated into five thermodynamic operations relevant for the ORC which are represented in the Excel model.

These processes are:

- Expansion (by binary turbine);
- De-superheating (by recuperator) (optional);
- Condensation (by heat rejection system);
- Pumping (by binary circulation pump);
- Heating/Evaporation (by vaporizer and pre-heater)

The cycle equipment duty and physical sizes are then estimated for the design point.

2.2. Heat Rejection System

In the ORC the power output at the turbine is maximised by maintaining a high working-fluid temperature differential across the turbine, which in turn is major contributor to the overall efficiency of the system. To maintain the high temperature differential required, plant designers will maintain a high inlet temperature by optimising the design of heat transfer equipment whilst maintaining a low outlet temperature by rejecting as much heat as possible at the condenser. The heat rejection system is an integral part of the cycle because it allows heat to be rejected from the cycle at the condenser.

The SKM model is able to represent three different heat rejection systems: air cooling; wet evaporative cooling; once through cooling in a modular fashion.

2.2.1 Dry Air Cooling

Dry cooling, also known as air cooled condensation (ACC). It is typical of binary plants that are located in areas without a ready supply of water. In this process heat is removed from the working fluid by passing air, by fans, over a bank of tubes containing the working fluid, in a 'fin-fan' heat exchanger structure. Dry cooling systems require a significant physical footprint to achieve the required heat exchange area, and are significantly affected by rises in the ambient DBT.

2.2.2 Wet Evaporative Cooling

In wet cooling, heat is removed from the working fluid by exchanging it in a non-contact shell and tube condenser with cold cooling water from a cooling tower (CT). Heat from this cooling water is then removed by evaporation of a portion of this cooling water at the CT. A CT can cause a visual plume in certain conditions. This may rule out this option from a consenting perspective depending on the project location and local requirements.

Wet cooling is more efficient than dry cooling because there are two mechanisms contributing to the overall transfer of heat: sensible heat transfer due to difference in temperature levels; and the latent heat equivalent of mass transfer resulting from evaporation of a portion of the circulating water. Wet cooling systems respond to ambient wet bulb temperature.

2.2.3 Once Through Cooling (OTC)

The OTC system also uses a non-contact shell and tube condenser. Cooling water flows through the condenser in a 'once-through' fashion to remove heat from the working fluid. Once the cooling water flows through the condenser it is returned to the water source or body at an elevated temperature. It is a non-consumptive use.

If an adequate cold source of water (e.g. a river or the ocean) is available in close proximity to the power plant, this method of cooling can be effective. The flow-rate 'take' and level of temperature increase are important factors to manage from an environmental perspective. A high temperature rise may cause excessive thermal pollution which can impact aquatic life or cause algae growth.

2.3. Treatment of 'Off-Design' Behaviour

The model presented has the capability to represent plant 'off-design' performance and sensitivity for variations of brine flow, brine temperature, and ambient temperature relative to the 'design case' parameters. The 'off-design' performance is modelled on equipment sizes established for the design point.

The off design treatment in the model includes some basic adjustments to the cycle to maximize net power. This includes variation of turbine inlet pressure and minimizing parasitic load, i.e. selected cooling tower fans are switched off when the heat load is lower than design load.

An equipment supplier would represent this behaviour as a 'correction curve' particular to their design. The SKM model provides an approximate representation of the correction curve for variation of brine flow, brine temperature, and ambient temperature.

Mines (2002) identified design features that could maximise off-design performance in the case of lower than design brine temperature. These include management of plant parasitic load using variable frequency drives of motors; and turbines with variable nozzle geometries. These features have been identified as future enhancements to be included in the model.

3. EGS SETTING

The setting assumed is a two production well, one injection well Enhanced Geothermal System petrothermal development. Wells are drilled to 4-5 km depth into a high heat production granite that has been stimulated to achieve sufficient inter-well permeability to support adequate fluid/rock heat transfer.

The amount of water loss (to reservoir) expected for an EGS project is difficult to quantify prior to drilling, reservoir stimulation, and fluid circulation tests. For economic models Tester et. al. (2006) proposes a water loss per total injected water ration of 2% based on current technology, with a figure of 1% in commercial mature years (2026+). For the purposes of this assessment a figure of 2% of water loss (per rate of injection) has been assumed.

The site considered is in Europe and located in an industrial area, and has water sources in the proximity that can be used for power generation purposes (i.e. water cooling).

The average ambient dry bulb temperature (DBT) is taken as 11°C. The OTC system assumes an inlet cold water temperature of 10°C and a maximum allowable temperature rise of 3°C.

4. MODEL SCENARIOS

Table 1 summarises the geothermal brine and plant configurations investigated with the model. They are intended to test the sensitivity of the model to a range of brine temperatures, brine flow rates, and heat rejection systems that might be expected in the EGS setting.

All cases were modelled with a PPTD = 8.2°C. This is a reasonably conservative assumption based on experience with other binary plants.

Case C provides for an additional heat exchanger on the brine side downstream of the preheater. This is to provide process heat to a secondary user for a direct use application such as district heating.

Table 1: Resource and Plant Configuration Cases

Case	Brine Temp. [ºC]	Minimum Rejection Temp. [ºC]	Brine Mass Flow [kg/s]	Condensing System	Direct Use
A	210	45	150	Dry Air	None
B	175	45	150	Dry Air	None
C	175	45	150	Dry Air	Process Heat (80 ºC Lower Limit)
D	175	45	150	Wet Evaporative CT	None
E	175	45	150	OTC	None
F	175	45	110	Dry Air	None
G	130	45	150	Dry Air	None

5. PERFORMANCE RESULTS

5.1. Design Case Results

The cases in Table 1 were modelled at their respective process and ambient design point to achieve a design basis and preliminary equipment sizing for the plant. The results are shown in Figure 2. The working fluid selected for these model runs was n-pentane.

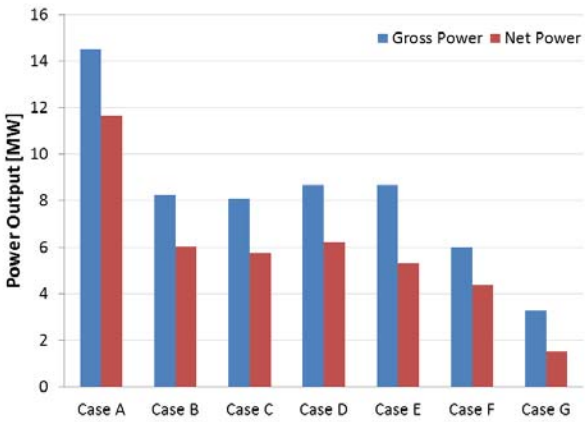

Figure 2: Gross and net power output by design case

The calculation of net power output includes the re-injection pump load which can be significant. In reality there will be some variability in this figure because it will be influenced by the well productivity / injectivity along with the transmissivity of the EGS resource. With the brine system pressures assumed this load is approximately 1.2 MW_e for the cases where the brine flow is 150 kg/s, and 0.89 MW_e for Case F at 110 kg/s.

It is clear that the higher brine inlet temperature in Case A results in a corresponding higher gross and net power output. This is due to higher thermal energy available to the cycle. In addition the gross efficiency also increases with brine inlet temperature. However there is also a higher parasitic load relative to the other cases. This is due to the increased heat rejection area and fan power required by the air cooled condenser.

Cases B and C are very much the same in terms of power output as the optimum brine rejection temperature using n-Pentane as a working fluid is approximately 80°C. This is the limiting set point for usable process heat for Case C - that is the heat rejection temperature from the plant pre-heater. Case C provides for 22 MW_{th} of thermal power for a secondary direct heat user.

Of the 175°C and 150 kg/s temperature and mass flow cases, Cases D (wet air cooling) and E (once through cooling) exhibit the highest gross output. This is due to the closer approach to the ambient wet bulb temperature achievable with these cooling options.

As a result these options are not limited (at this temperature) by ambient DBT, but rather by ensuring the condensation pressure does not fall below atmospheric and permit air leakage into the condenser. The difference between these two cases is the source of parasitic load (cooling water circulation pumps and fans for wet cooling tower, and water circulation pump for the once-through cooling). For the once-through cooling (case E) an assumption was made that the distance between the cooling water source and the power plant is 500 m. Due to the volume of cooling water required, this resulted in a significant parasitic loss (and a lower net power compared to case D). This value will increase or decrease according to the distance of the cooling water source to the plant.

Case F represents a scenario of reduced mass throughput from the EGS reservoir. This case results in a decrease of approximately 1.9 MW_e (gross) or 1.3 MW_e (net) below Case B for a brine reduction of 40 kg/s.

Case G represents a scenario of reduced temperature from the EGS reservoir. This case shows a decrease of 3.9 MW_e (net) as compared to case B for a 45°C drop in resource temperature. The variation in output between Cases A, Case B and Case G highlights the dependence between power plant output and brine inlet temperatures.

5.2. Sensitivity to Brine Inlet Temperature

Each of the design cases was modelled over five points from their design mass flow down to the mass flow at which the cycle produced no net power.

All cases were run at an ambient DBT of 11°C. The variation of net power output with brine temperature is shown in Figure 3.

Case D is more 'robust' (compared to Case B) with respect to a decreased brine inlet temperature. This is due to the increased thermal efficiency of the wet cooling option, in turn due to reduced condenser pressure achieved as well as the decrease in parasitic load with respect to the air cooling. The case E net output will vary depending on the relative location of the cooling water source (and therefore the duty) of the pump.

The design cases have been run from their respective inlet temperature design points down to the cut off value. Figure 3 indicates a cut off temperature of approximately 100°C and 110°C for the cases. Below that temperature the cycle produces no net power.

Case A has been designed to operate at 210°C and this is evident in Figure 3 when compared to the other design cases which are equipment limited.

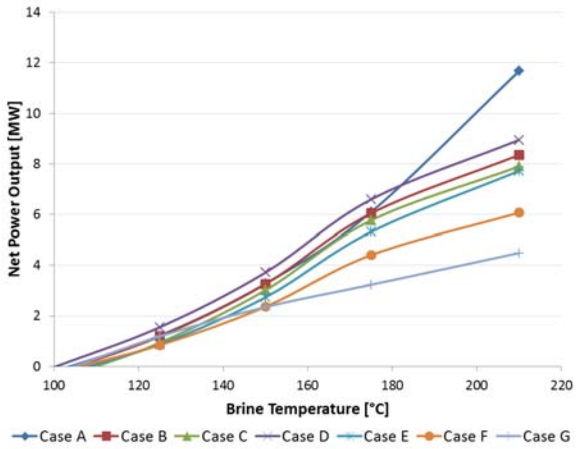

Figure 3: Sensitivity of Net Power Output to Brine Inlet Temperature at Design Ambient Condition

For the other cases, operating above the design temperature results in an increase in power output as the cycle is able to operate at higher pressures due to the higher inlet brine temperature (i.e. through maintaining the constant pinch-point temperature), which results in a significant increase in cycle efficiency. Above the design point the equipment has not been sized to accommodate this increase in temperature, so is limited by the amount of heat that can be transferred from the brine through the pre-heater and vaporiser.

The result of this is that less brine is required to transfer the same amount of heat to the binary fluid and conversely there is a decrease in binary fluid flow due to the increase in enthalpy difference over the pre-heater and vaporiser ($Q = m\Delta h$) due to operation at a higher pressure. As the mass flow of binary fluid is significantly lower than that of Case A, the net output from the other design cases operating at 210°C is significantly lower than case A at the same brine temperature. Equipment over sizing that is more in line with the Case A plant design could be considered. However this should be considered from a technical-economic perspective and factor in the increased capital cost of Case A compared to Case B.

There are a number of implications surrounding operating at higher turbine inlet pressures. The turbine and generator will need to be rated to operate above their design point to cope with the increase in inlet pressure as well as the increased mechanical output from the turbine.

Operating at higher turbine inlet pressures also results in an increased pump load on the binary circulation pumps which in turn decreases net power output. Some of this is offset by the decrease in binary fluid flow, however the increase in head required from the pumps results in a net parasitic increase from the binary pumps operating above the design inlet brine temperature.

The increased brine temperature also has an effect on the cooling load for the various cases. As the same amount of heat is transferred from the brine, and more of this heat is converted into mechanical work (due to increased efficiency), there is a resultant decrease in heat required to be rejected from the cooling system. This results in a decrease in parasitic load for each of the cooling systems.

The other consideration for Case C is that with the decrease in brine flow required for the binary cycle, additional heat is available for the process heat user if brine production is maintained.

5.3. Sensitivity to Brine Mass Flow

Each of the design cases was modelled over five points from their design mass flow down to the mass flow at which the cycle gave no net power. All cases were run at an ambient DBT of 11°C. This is illustrated in Figure 4.

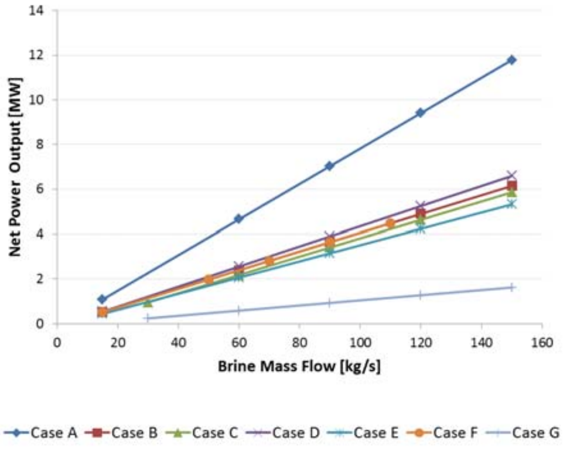

Figure 4: Sensitivity of Net Power Output to Inlet Brine Mass Flow Rate at Design Ambient Condition

The isentropic efficiency of the binary turbine is assumed constant throughout the off design analysis. Modern radial turbine design can integrate variable inlet nozzles into the design to smooth off design conditions allowing inlet conditions to be varied without wasteful throttling. This enables isentropic efficiency to be practically constant throughout seasonal variations (Marcuccilli & Zouaghi, 2007).

From Figure 4 it is clear that the highest temperature option (Case A) results in the highest power output over most of the range of mass rates shown. The gross power output of Case A is higher than that of the other cases however a greater parasitic loss is observed in the cycle resulting in a lower net output at lower brine flow rates. Of the 175°C options the wet evaporative cooling (Case D) and OTC (Case E) display a higher net output at higher brine mass flow rates, however as brine flow rate decreases the effect of the parasitic losses on the cycles becomes less apparent and the net power approaches that of the other 175°C geothermal fluid cases.

Operating at a mass flow above the design point would require consideration of over-sizing at the design point. This has not been considered in the design case and associated modelling. It will also impact the fluid gathering and re-injection system sizing.

5.4. Sensitivity to Ambient Temperature

Each of the design cases was modelled over a range of dry bulb ambient temperatures between 1 and 22°C. These values were chosen as they represent a logical spread given the ambient data for the European region under consideration.

Excursions outside of this range can be inferred from Figure 5 noting the limitation of air cooled condenser size and the ability to reject heat above the design ambient condition. This limitation is due to the additional extended finned-tube surface area required at higher heat duty above design

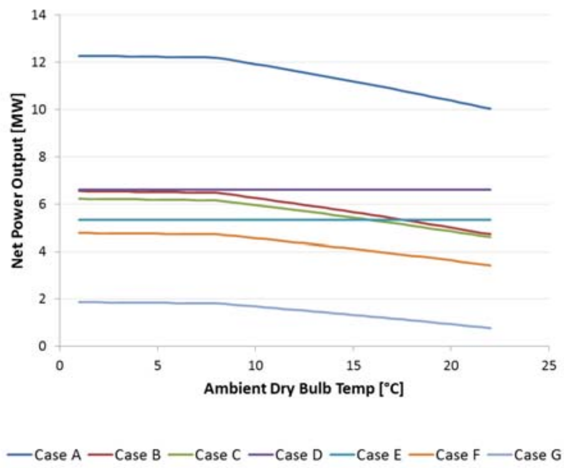

Figure 5: Sensitivity of Net Power Output to Ambient Dry Bulb Temperature

The condenser pressure for the 'once-through' and 'wet' cooling options is not overly affected by ambient temperature over this temperature range (slight decrease in net output for wet cooling not apparent in the figure). This gives a relatively constant net power output against temperature.

All of the dry cooling options exhibit the same trend with ambient temperature. From 1°C the net output decreases slightly due to the increasing parasitic load from the air cooled condensers. Once the ambient conditions reach a temperature of approximately 9°C, the air cooled condenser can no longer maintain a condenser pressure of 1 bara and the turbine output begins to drop significantly with increasing temperature. The same result would be observed for the wet cooling tower (Case D), only at a higher temperature once the approach to the wet bulb temperature could no longer be maintained.

Between 0°C and approximately 9°C the difference in output for the dry and wet cooling options is relatively small. Both cases are able to achieve a condenser pressure of approximately atmospheric and the parasitic load of the wet and dry cooling towers is comparable.

Once the condenser pressure can no longer be maintained by the dry cooling options, the net output of the ACC cases begin to decrease with respect to the wet cooled option.

The output of Case E is dependent on the location of the cooling water source from the plant and the pumping duty. At 500 m from the plant the output of this cycle case only becomes advantageous above the dry cooled options above approximately 16°C.

5.5. Performance Variation by Month

The outputs from the modelling for ambient temperature were overlaid with the monthly ambient temperature data for the region. The 'average' or monthly P50 (between max and min temperatures) was selected for each month and the corresponding power output inferred from the off-design modelling. The process is shown in Figure 6, with net power output per month shown as a graph in Figure 7

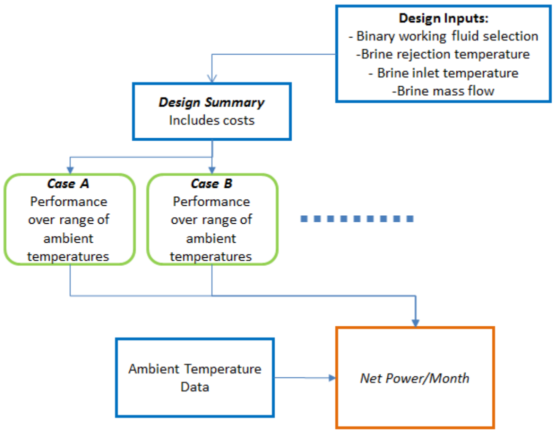

Figure 6: Process to Calculate Monthly Variation

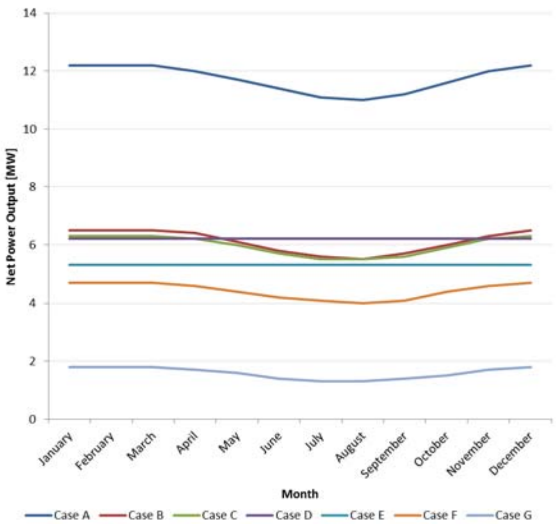

Figure 7: Monthly Net Output

The wet cooling options show little variation in ambient dry bulb temperature. This is correspondingly shown by the flat power output over the year.

As expected the air-cooled options show a decrease in power output over the hotter northern hemisphere months of June to October. As ambient dry bulb temperature is always higher than wet bulb, the performance of dry cooling is more sensitive to fluctuations in the ambient air temperature.

The monthly outputs were combined, in conjunction with an assumed availability factor, to calculate an approximate annual net generation (MWh) of power produced for the different cycles. All cases assume an annual outage of 14 days to occur in September for planned maintenance.

More detailed plant generation figures would require detailed meteorological data to capture diurnal variation (daily) in ambient. In this study an analysis based only on monthly average values was undertaken.

5.6. Sensitivity to Working Fluid Selection

In order to assess the effect that working fluid has on the net output of the power plant and predict potential upside from the use of a different working fluid, the following additional sensitivity investigation was undertaken.

The working fluids investigated include n-pentane, n-butane, iso-pentane and R-245fa. As the model only operates represents sub-critical cycles, those options that require supercritical operation have not been modelled. (Note R-245fa and n-butane would be supercritical cycles at the case A design conditions and therefore have not been included here in this comparison).

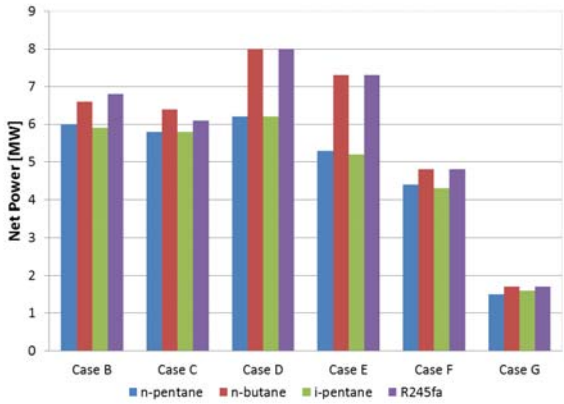

Figure 8: Net Power Output with Various Working Fluids

From Figure 8 it is evident that the wet cooled design cases (case D and Case E) offer significant benefit over the dry cooled options for n-butane and R-245fa. In the model this is due to the condenser pressure being limited to atmospheric pressure or above, for the dry cooled cases, to inhibit air leakage into the condenser and hence into the binary cycle. As the condenser pressure for n-butane and R-245fa is significantly above atmospheric pressure, it does not reach this limitation and is instead governed by the approach to the cooling water temperature.

The model results suggest that n-butane or R-245fa would be good choices for optimising net output for the design conditions. The selection of working fluid for an ORC plant is not a straight forward matter. Aside from net output other considerations such as health and safety, environmental impact and permitting, operational considerations (such as pressurised storage) should be considered. Of note is the increasing industry focus on avoiding fluids that have a high potential for contributing to global warming.

Hydrocarbon working fluids can produce favourable results in terms of brine specific consumption and second law efficiency for medium to low temperature heat sources. Synthetic refrigerants (e.g. R-245fa) have been found to perform slightly better when there is greater difference, in brine supply and rejection temperatures (more than about 60-70°C) (Franco and Villani, 2009). This is evident in the results between case B and case G in Figure 8.

6. WATER USAGE

The water requirements for the operating plant comprise reservoir make-up, potable water supply, and water required for the heat rejection system. A summary of water use requirements is given in Table 2. The heat rejection water requirement, denoted as either OTC or Evaporative Make-up, is dependent on the plant design.

Table 2: Water Usage by Case

Case	Potable Water [l/day]	OTC [l/s]	Evaporative Make-up [l/s]	EGS Reservoir Make-up [l/s]	TOTAL [m^3/day]
A,B,C,F ,G	525	-	-	3	260
D	525	-	24	3	2300
E	525	4600	-	3	400000

7. ESTIMATED CAPITAL AND OPERATING COSTS

Capital and operating costs for the different cases were estimated and are presented in Figure 9 and Figure 10 respectively.

The capital cost estimate presented includes allowances for indirect costs. These include engineering and design (various), owner's costs (15% of total installed costs), growth (10% of direct costs), and contingency (15% of direct costs). The estimate excludes the subsurface costs of the project including the cost of exploration, drilling, and reservoir engineering.

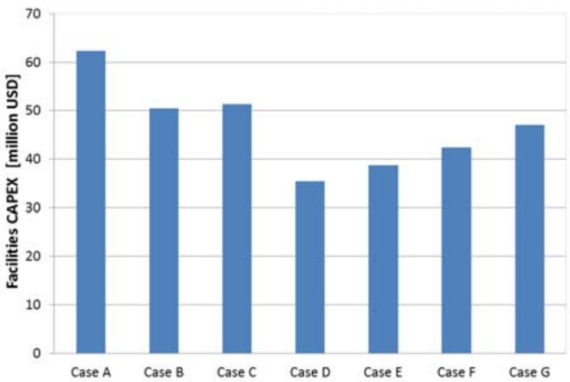

Figure 9: Plant Capital Cost (+/- 40% level of accuracy, 2012 Basis)

These estimates were mainly derived from database information adjusted using a parametric/factored approach. This information was supplemented with supplier information, particularly around the power island.

The estimate of the operations and maintenance costs have been developed by considering: the operational staff roles required; water charges tariff; and our experience for insurance and maintenance costs. Case D in particular has a higher operational cost due to the cost of consumptive make-up water (assumed from municipal supply) for the evaporative cooling tower.

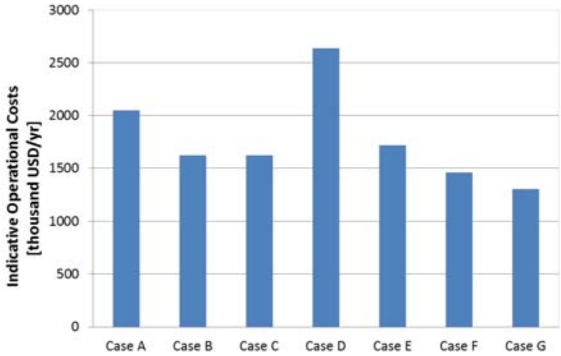

Figure 10: Indicative Annual Operational Cost Estimates (+/- 40% level of accuracy, 2012 Basis)

8. FINANCIAL MODELLING

Financial modelling has been undertaken to assess the effect that differing power plant heat rejection selection choice has on financial viability. Use of estimated capital and operating cost in this analysis is described in Section 7, and has been combined with estimates of costs for subsurface exploration.

The heat rejection method selected for a project has an effect on the capital and operating costs, and also the net plant output. For this purpose, cases B, D, and E have been modelled (using n-pentane as the working fluid) to assess the difference that this technology choice has for a given set of resource conditions.

A summary of the financial model input assumptions is given in Table 3. Debt finance was assumed at 50% to reflect a typical investor's appetite to lend against the project after the resource is proven. The geothermal power subsidy rate is an indicative subsidy figure considered possible for EGS projects to receive in the region (e.g. Groves et. al. (2012)). The nominated power price is indicative and allows comparison across cases.

Table 3: Financial model input assumptions

Parameter	Unit	Value
Debt finance proportion	%	50
Interest rate on loan	%	8
Loan term	Years	10
Geothermal power subsidy	USD/MWh	136
Power price	USD/MWh	104
Development duration	year	5
Project lifetime	year	25
Weighted Average Cost of Capital (WACC)	%	10.5

A summary of financial model outputs is given in Table 4. The results show the lowest required total tariff to achieve a zero project Net Present Value (NPV) is generated by Case D at 265 USD/MWh. This is the electricity tariff required to achieve a project Internal Rate of Return (IRR) to equal the post-tax nominal WACC, 10.5%.

Table 4: Financial model outputs

Model Output	Case B	Case D	Case E
Annual output [GWh]	51.15	51.73	44.22
Project IRR [no subsidy]	-0.17%	-2.47%	-2.07%
Project IRR [with subsidy]	8.79%	9.29%	8.11%
Required total tariff 10.5% IRR [USD/MWh]	282	265	298

The project IRR has been calculated for two scenarios: with an EGS power subsidy; and without. Applying the assumptions given in Table 3, the projects are all NPV negative, i.e. the NPV of cash flows over the life time of the project is less than 0, assuming a discount rate of 10.5%. Therefore these project cases are not considered economic to undertake and unlikely to proceed.

However, a comparison across the cases reveals the highest performance is achieved by Case D when evaluated with an EGS power subsidy. It is interesting to note that Case D is the lowest performer when considered with no subsidy. This is due to the return on investment being extremely low for the no subsidy scenarios. This results in project revenue streams being largely dependent on depreciation as a proportion of capital expenditure, of which Case D has the lowest capital cost estimate.

Financial performance is not the only consideration in ranking the technology plant options. The environmental and community aspects need to be considered in conjunction with financial performance when assessing project viability. For example, Case D will likely have a visible plume from the cooling tower, and Case E requires a non-consumptive water take and return from a natural source such as a river.

Using the same assumptions as detailed in Table 3 and modelling the Case A power plant with a subsidy, the NPV over the life time of the project is USD 54 million, or a project IRR of 14.8%. Under this scenario, Case A would be considered a financially viable project. The improved financial performance is a function of the higher resource temperature.

The financial indicators here are indicative and actual plant performance could be optimised through working fluid selection (i.e. selection of n-butane) and tuning of the PPTD.

9. CONCLUSIONS

1) A relatively simple Excel based model of a sub-critical, brine-fed ORC power plant process has been developed. Different combinations of resource conditions, ambient conditions, binary working fluids, and heat rejection systems can be readily analysed and compared.

2) Wet cooling options for plant heat rejection are attractive from a performance and capital cost perspective compared to dry cooling. However the operational cost, environmental impact, proximity to plant and security of supply need to be considered.

3) At present in an EGS setting a power development appears to be economically marginal. It is viable when sufficient resource temperature and subsidy is present.

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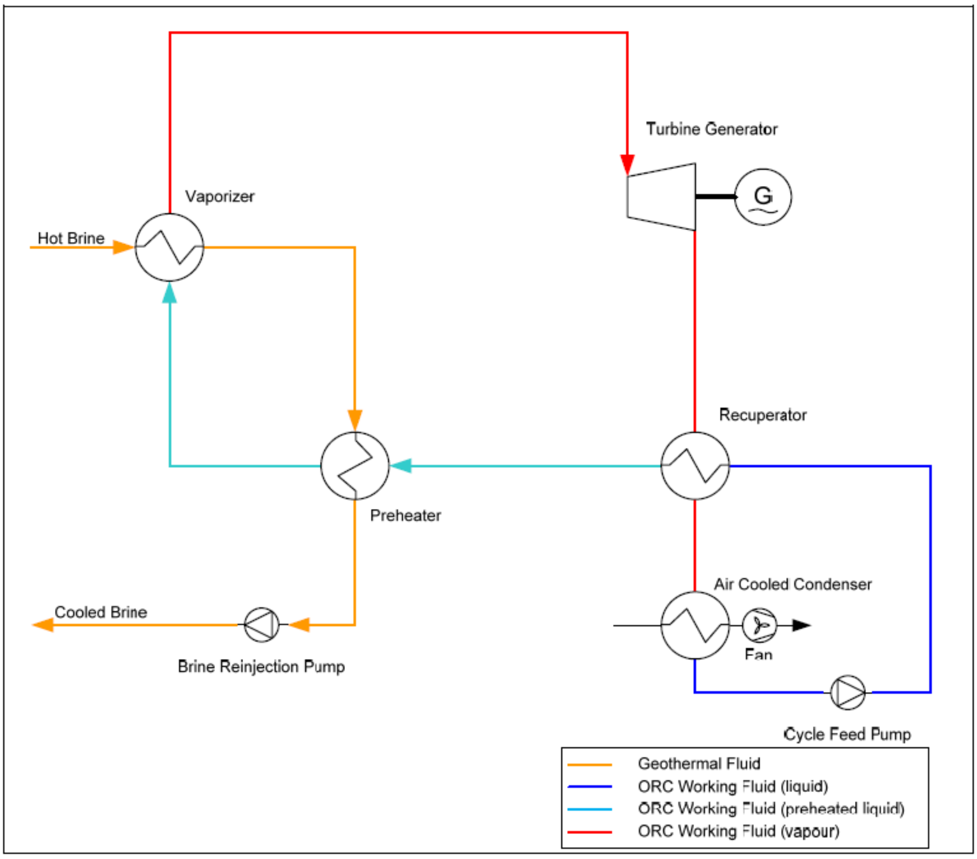

Figure 11: Simplified Cycle Diagram of a Recuperative ORC