Viability Study into the extraction of Geothermal energy through use of an Organic Rankine Cycle

Jay Dickson
s3719855

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1 Introduction and Background

In the following sections consideration is given to the viability of using an Organic Rankine Cycle (ORC) to extract energy from geothermal water to be used elsewhere for primarily farming applications. Due to the nature of the energy source the operating temperatures are relatively low in comparison to mainstream energy production systems, such as coal or natural gas power plants. A limited number of technologies exist to take advantage of these low temperature conditions, the most prominent by far is the Organic Rankine Cycle, other technologies available for low temperature use include thermoelectric generators and Stirling engines.

A thermoelectric generator makes use of a temperature differential in a conductor or semiconductor, this incites the flow of electrons between temperature regions and can therefore generate electricity, the main issue with a thermoelectric generator comes down to its inefficiency and largely limited use in the industry, making this option costly and unproductive.

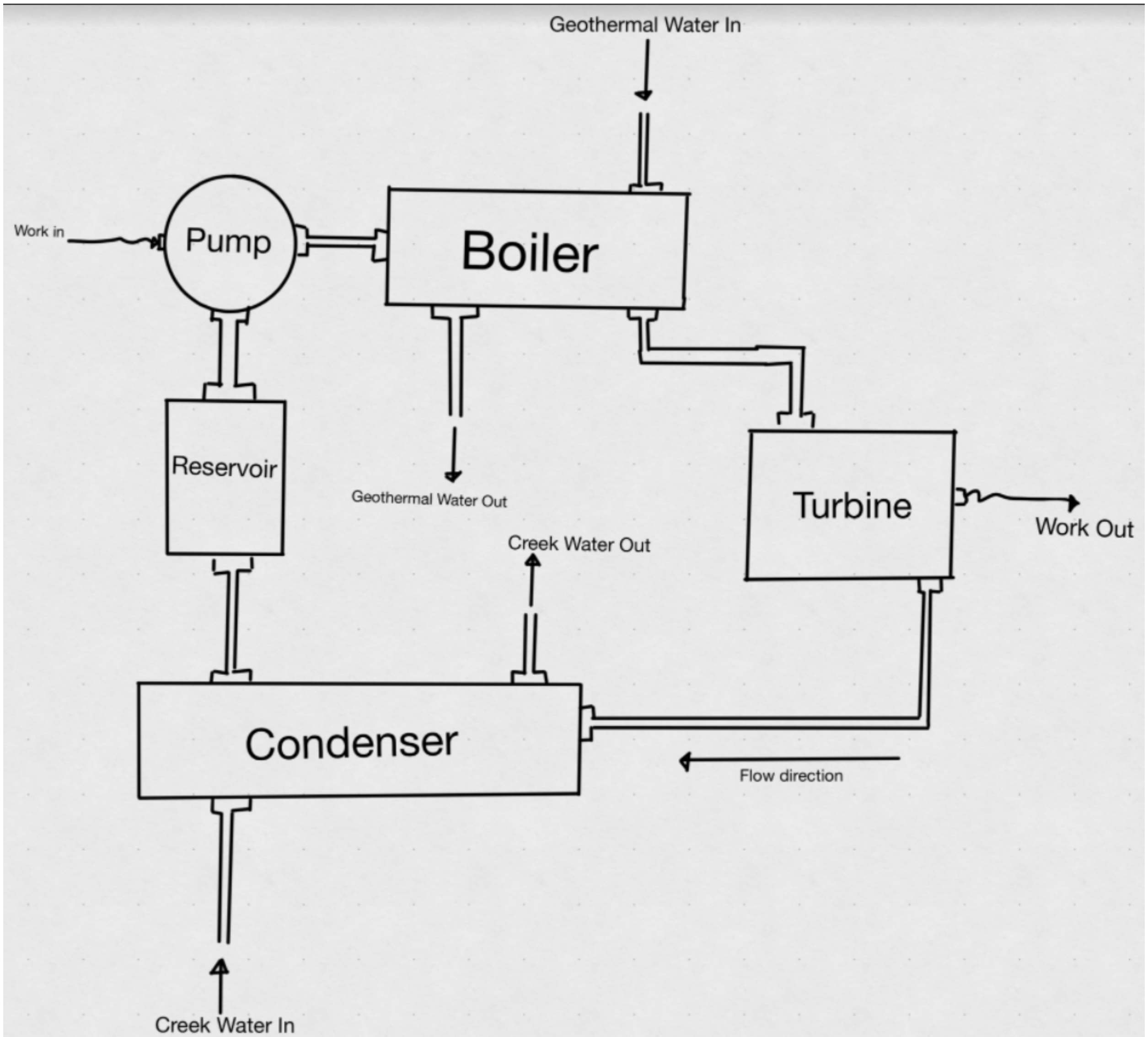
A Stirling engine does not suffer from the same inefficiency problems as a thermoelectric generator and can generate a solid power output, they rely on a temperature differential to expand and contract a working fluid that can then generate rotational motion useful for providing power. Stirling engines however are difficult to build in large enough sizes for the output required for the application being considered and are also expensive and relatively specialised equipment.

An ORC doesn't suffer from many of the same issues. An ORC is versatile due to the variety of working fluids available with properties suitable for most temperature gradients, this is the main advantage over a traditional Rankine cycle – much finer control over the boiling and condensing points of the fluid. Further an ORC consists of widely available components that are relatively cheap and configurable as such this system is highly versatile from a hardware perspective as well. For these reasons an ORC is the most valid and well-suited candidate for the application.

Discussed below is a proposition for one such configuration of an ORC and how it would perform if modelled as both an ideal and realistic system. Calculations of the conditions such as Enthalpy, Temperature, Entropy and Work at various points in the system and across processes are laid out and discussed in the following sections. Finally included is an estimate of the expected power output throughout a year using historical information and a consideration of alternative working fluids and configurations.

2 Layout

Figure 2.1 - Vapor Compression Cycle Layout



Including a reservoir before the pump in the layout allows for the introduction of the initial fluid and allows for pre-heating of the refrigerant before the system is initially turned on. It also serves as a buffer for any small leakages or evaporation. This will ensure the system never runs dry. Placing the reservoir before the pump but after the condenser ensures the contents will be in liquid phase and also at a constant pressure across the container.

3 Ideal Cycle Calculations

Table 3.1 - Assumptions and Initial Conditions.

Variables	Value	Units
Boiler Pressure	1200	kPa (abs)
Condenser Pressure	725	kPa (abs)
Geothermal Water Temp	75.36	Degrees
Water Specific Heat Capacity	4.2	KJ/kg.K
Working Fluid	R134a	n/a

Table 3.2 - Final Calculated Values. Pressure in KPa, Specific Enthalpy in Kj/kg, Specific Entropy in Kj/kg K. Blue indicates mark dependent values.

Process Number	Point	Condition In	Condition Out	Pressure In	Pressure Out	Temp In	Temp Out	Enthalpy In	Enthalpy Out	Entropy In	Entropy Out
4 - 1	Condenser	Superheated Vapour	Saturated Liquid	725	725	50.237	27.875	288.335	90.520	n/a	n/a
3 - 4	Screw expander	Superheated Vapour	Superheated Vapour	1200	725	70.360	50.237	301.020	288.335	0.995	0.995
2 - 3	Boiler	Saturated Liquid	Superheated Vapour	1200	1200	28.678	70.360	90.917	301.020	n/a	n/a
1 - 2	Boiler Feed Pump	Saturated Liquid	Saturated Liquid	725	1200	27.875	28.678	90.520	90.917	0.338	0.338

We first calculate the temperature of the Refrigerant R134a as it exits the heat exchanger/boiler, it is stated that this temperature should be 5 Degrees below the Geothermal Water Temperature of 75.36 Degrees, this gives us a value of 70.36 for the boiler output temperature.

Using this value, we can determine the input temp at the Screw expander/Turbine and given a pressure of 1200 Kpa can find the Enthalpy and Entropy at this point (Çengel, Cimbala & Turner 2012b, pp. 988–992).

As this is an ideal Cycle the expansion process is assumed to be isentropic and adiabatic, given a constant entropy across this process S3 and S4 are equivalent. This can be used to evaluate the quality of the fluid at point 4.

Table 3.3 - Relevant State Function values of R134a at 725 KPa (Çengel, Cimbala & Turner 2012b, pp. 988–992)

Entropy - f	Entropy - fg	Enthalpy - f	Enthalpy - fg
0.3379	0.58181	90.52	175.145

Using the values in **Table 3.3** and the entropy calculated at point 3 in **Table 3.2**, we can find the quality and enthalpy at point 4.

$$x_4 = \frac{S_4 - S_f}{S_{fg}} \tag{3.1}$$

$$h_4 = h_f + x_4 h_{fg} \tag{3.4}$$

Therefore, the quality at point 4 is 1.129 and the relevant enthalpy is 288.335. Given a quality of 1.129 this indicates the fluid at the turbine output is Superheated. Referring to the Superheated Refrigerant Tables (Çengel, Cimbala & Turner 2012b, pp. 988–992) we can determine the temperature of the vapour as it exits the turbine as 50.237 Degrees Celsius.

Table 3.4 - Work and Heat Transfer Values for Unit Mass Flow Rate, Generator and Motor are 100% Efficient and so are equivalent to Pump In and Turbine Out.

Power and Heat Transfer	**Work/Heat 1 kg/s**	**Units**
Qin	210.103	KW
Qout	197.815	KW
W-net	12.288	KW
W_pump-In	0.397	KW
W_turbine-Out	12.685	KW
Efficiency	5.849	Percent

Now given these values we can evaluate the Work and Heat transfer done on and by the system, as well as the Efficiency of the setup at converting heat to power.

First we can evaluate the work in for the Pump, given this is another isentropic and adiabatic process the work per kg can be calculated using the specific volume at 725 KPa, which is 0.0008363 m^3/kg (Çengel, Cimbala & Turner 2012b, pp. 988–992).

$$w = v_f(P_2 - P_1) \tag{3.5}$$

Q-in and Q-out can be calculated as the difference in enthalpy across the boiler and condenser respectively.

$$Q_{in} = h_3 - h_2 \tag{3.6}$$

$$Q_{out} = h_4 - h_1 \tag{3.7}$$

The net work done is equal to the difference between Q-in and Q-out.

$$W_{net} = Q_{out} - Q_{in} \tag{3.8}$$

From this we can find the work output by the turbine by adding the work in at the pump to the net work done.

$$W_{out} = W_{net} + W_{in} \tag{3.9}$$

Finally we can evaluate the efficiency of the cycle at converting heat to power, this is the ratio of Net Work to Heat in.

$$Eff = \frac{W_{net}}{Q_{in}} \tag{3.9}$$

All the final values from the above calculations are listed in **Table 3.4.**

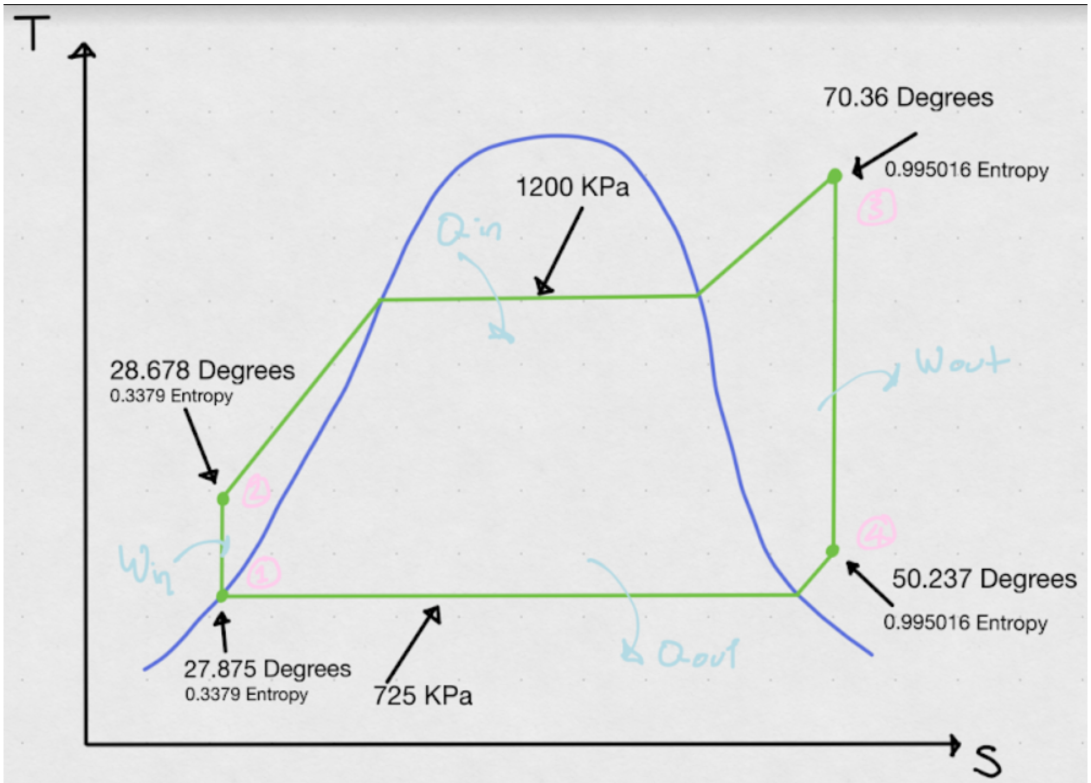


Figure 3.1 - TS Diagram for an ideal Cycle. T in Degrees Celsius, P in KPa, S in Kj/kg.

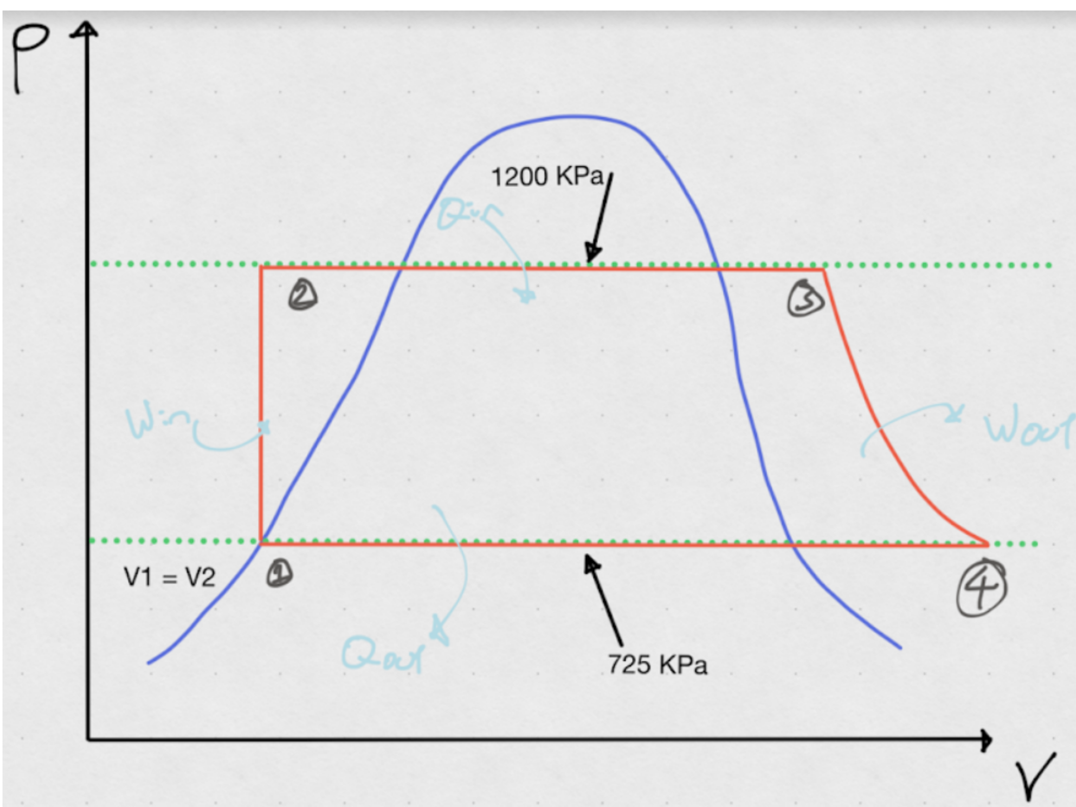


Figure 3.2 - PV Diagram for an ideal Cycle. P in KPa, V in m^3.

4 True Cycle Calculations

Table 4.1 - Assumptions and Initial Conditions.

Variables	Value	Units
Boiler Pressure	1200	kPa (abs)
Condenser Pressure	725	kPa (abs)
Boiler Feed Pump Pressure	725	kPa (abs)
Water Specific Heat Capacity	4.2	KJ/kg.K
Geothermal Water Temp	75.36	Degrees
Screw Expander Efficiency	86.4	%
Boiler Feed Pump Efficiency	93.45	%
Electric Generator Efficiency	78.7037037	%
Electric Motor Efficiency	84	%

Table 4.2 - Final Calculated Values accounting for isentropic efficiency. Pressure in KPa, Specific Enthalpy in Kj/kg, Specific Entropy in Kj/kg K. Blue indicates mark dependent values.

Process Number	Point	Condition In	Condition Out	Pressure In	Pressure Out	Temp In	Temp Out	Enthalpy In	Enthalpy Out	Entropy In	Entropy Out
4 - 1	Condenser	Superheated Vapor	Saturated Liquid	725	725	51.535	27.875	290.060	90.520	n/a	n/a
3 - 4	Screw expander	Superheated Vapor	Superheated Vapor	1200	725	70.360	51.535	301.020	290.059	0.9950	1.0001
2 - 3	Boiler	Saturated Liquid	Superheated Vapor	1200	1200	28.170	70.360	90.945	301.02	n/a	n/a
1 - 2	Boiler Feed Pump	Saturated Liquid	Saturated Liquid	725	1200	27.875	28.170	90.50	90.945	0.3379	0.3393

Calculation of Initial conditions for Turbine and Feed Pump are shown in section 3. First we need to calculate the new entropy, enthalpy and temperature at point 4. Using the efficiency values located in **Table 4.1** and the isentropic ideal values calculated in **Table 3.2** we can evaluate the Enthalpy at point 4 and use this to lookup the associated temperature and entropy in the Superheated refrigerant properties table.

$$h_{4a} = \ -\ (eff_{turb} * (h_3 - h_{4s})\ -\ h_3 \qquad (4.1)$$

This gives us a specific enthalpy value of 290.059, this allows us to determine the corresponding temperature and entropy at 725 KPa. Given in **Table 4.2** (Çengel, Cimbala & Turner 2012b, pp. 988–992).

Moving now to the Pump we can evaluate the significant properties at point 2 with the same general method. Using the isentropic efficiency value to find the entropy at point to and using that value to determine the corresponding temperature and entropy.

$$h_{2a} = \frac{h_{2s} - h_1}{eff_{turb}} + h_1 \tag{4.2}$$

This gives us a specific enthalpy value of 90.945 and corresponding temperature and entropy of 28.170 and 0.3393 respectively assuming a pressure of 1200 KPa (Çengel, Cimbala & Turner 2012b, pp. 988–992).

Table 4.3 - Work and Heat Transfer Values for Unit Mass Flow Rate, accounting for inefficiencies in Generator and Motor.

Power and Heat Transfer	**Work/Heat 1/kgs**	**Units**
Qin	210.103	KW
Qout	197.815	KW
W_pumpIn	2.352467322	KW
W_turbine-Out	10.96008255	KW
Power Out Generator	8.625990895	KW
Power In Electric Motor	2.728862093	KW
Net Power	5.897128802	KW
Efficiency	2.806783153	Percent

Finding the real pump and turbine work output can be done using the efficiency values in **Table 4.1.**

$$W_{In\ Pump} = Eff_{pump} / W_{in\ Pump\ Ideal} \tag{4.3}$$

$$W_{Out\ Turb} = Eff_{turb} * W_{Out\ Turb\ Ideal} \tag{4.4}$$

We can now use the efficiencies for the motor and generator to calculate the actual useful power we get out of the generator and how much of it is required for the motor driving the pump.

$$P_{out} = W_{Out\,Turb} * Eff_{Generator} \tag{4.5}$$

$$P_{in} = Eff_{Motor} / W_{In\,Pump} \tag{4.6}$$

We can then find the net power and the efficiency.

$$P_{net} = P_{out\,gen} - P_{in\,motor} \tag{4.7}$$

$$Eff = \frac{P_{net}}{Q_{in}} \tag{4.3}$$

All the final values from the above calculations are listed in **Table 4.3.**

It can be seen that a reduction in the overall power output and efficiency has occurred when compared to the initial ideal system consideration outlined in section 3. Having considered a more realistic model in this section isentropic inefficiencies in the Compression and Expansion Processes were factored in, as well as the power conversion efficiency of the motor and generator. This is an expected result but shows that the ideal cycle follows reality with little accuracy. The Net power out dropped by over half when inefficiencies were taken into account as did the efficiency.

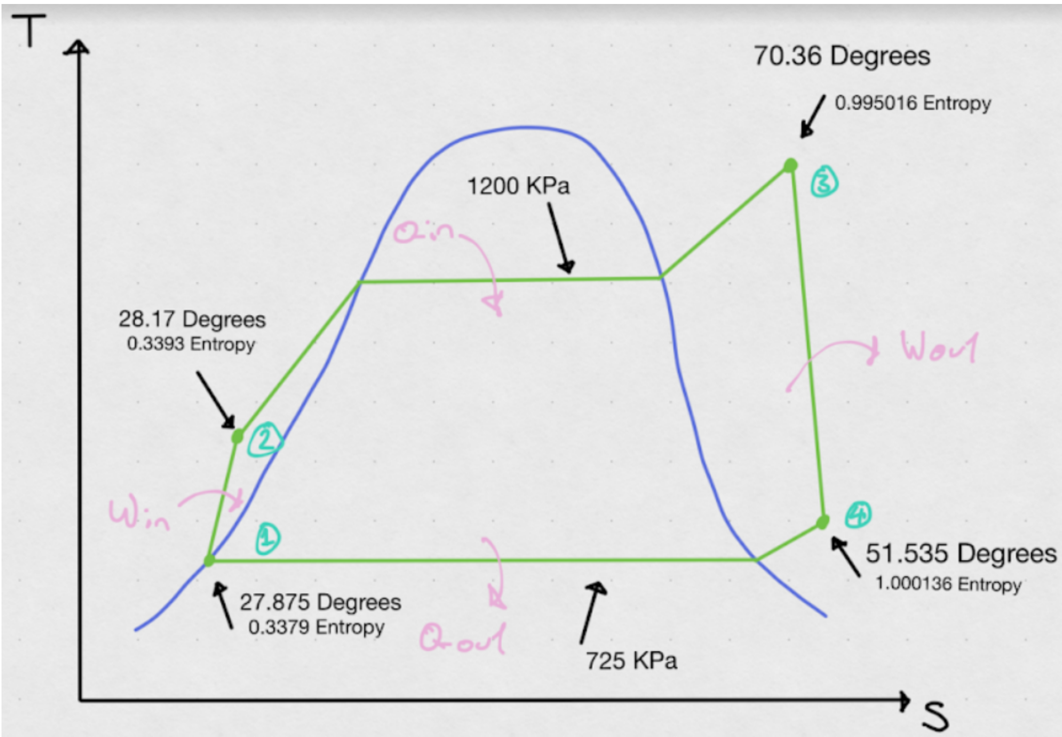


Figure 4.1 - TS Diagram for a non-isentropic Cycle. T in Degrees Celsius, P in KPa, S in Kj/kg.

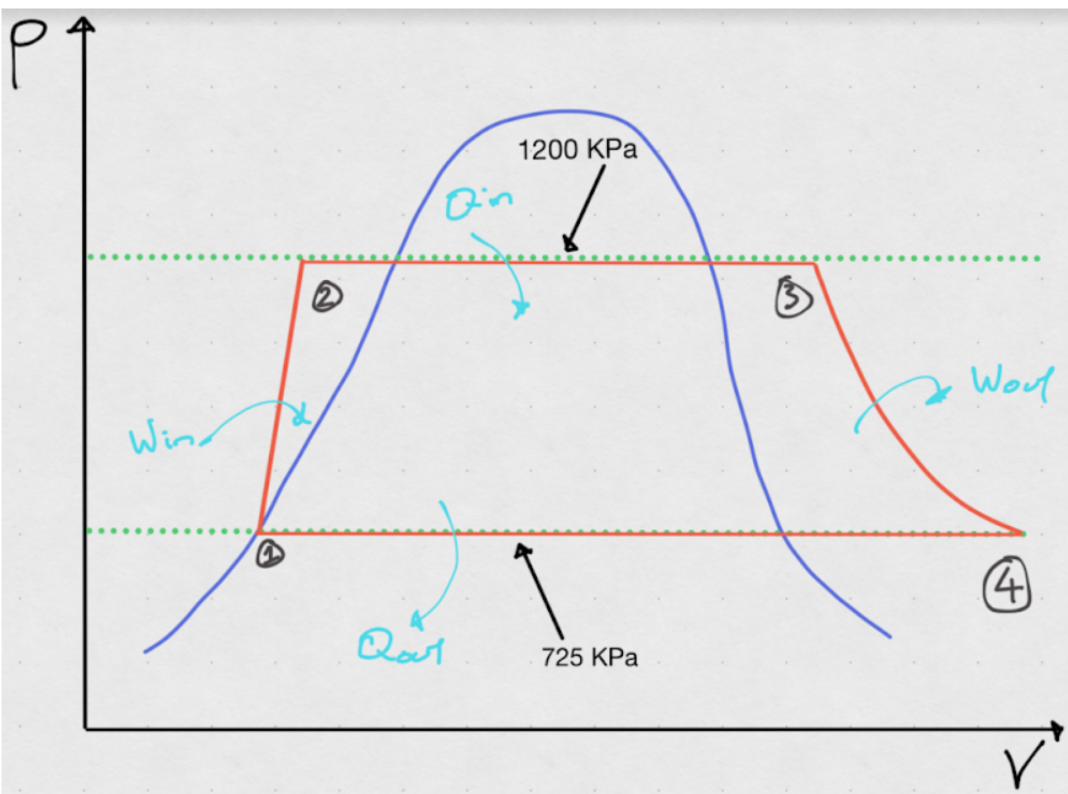


Figure 4.2 - PV Diagram for a non-isentropic Cycle. P in KPa, V in m^3.

5 Monthly Geothermal Variance Calculations

Table 5.1 - Initial values and important properties.

Specific Heat Capacity of Water	Temp Change	Delta-H Boiler	Net Kj/Kg
4.2	5	210.103	5.8972

Table 5.2 - KW of energy from Geothermal Water, related Mass flow rate of refrigerant and Net Output power.

Month	Mass Flow Rate Water (kg/s)	QDot Water (KW)	Mass Flow Rate Ref (kg/s)	Net Power Out (KW)
Jan	44.1	926.1	4.4078	25.994
Feb	45.6	957.6	4.5575	26.878
Mar	46.09	967.89	4.6063	27.167
Apr	45.24	950.04	4.5217	26.666
May	43.63	916.23	4.3608	25.717
Jun	42.34	889.14	4.2319	24.956
Jul	42.22	886.62	4.2195	24.885
Aug	43.37	910.77	4.3348	25.563
Sep	45.01	945.21	4.4987	26.530
Oct	46.04	966.84	4.6017	27.137
Nov	45.76	960.96	4.57371	26.972
Dec	44.37	931.77	4.4348	26.153

Example Calculation for month of January shown below.

$$Q_{Geo} = Q_{boiler} = m_{dot\ geo} * c_{water} * \Delta T \quad (4.7)$$

$$Q_{Geo} = Q_{boiler} = 44.1 * 4.2 * 5 = 4 \quad (4.8)$$

$$M_{dot\ Ref} = Q_{boiler} / \Delta h \quad (4.9)$$

$$M_{dot\ Ref} = 44.1 / 210.103 = 4.4078 \quad (4.10)$$

$$P_{net\ out} = M_{dot} * Specific\ Net\ Work \quad (4.11)$$

$$P_{net\ out} = 4.407 * 5.8972 = 25.99 \quad (4.11)$$

Given the data calculated in **Table 5.2** the pump needs to operate at a flow rate of 3.61 which would eclipse the max value required for complete energy extraction.

Table 5.3 - Percentage of Net Power Out to Total Load. Yearly Average for Power Output and Total Load.

Month (Letters haha)	**Net Power Out (KW)**	**Total Load (KW)**	**Power Out as Percentage of Load**
Jan	25.99358878	59.01	44.049
Feb	26.87772445	58.82	45.694
Mar	27.1665421	58.02	46.822
Apr	26.66553189	56.34	47.329
May	25.7165596	54.89	46.851
Jun	24.95620292	54.67	45.648
Jul	24.88547207	55.8	44.5979
Aug	25.56330942	57.55	44.419
Sep	26.52996442	58.71	45.188
Oct	27.13707092	55.79	48.641
Nov	26.97203226	58.33	46.240
Dec	26.1527332	58.46	44.736
Yearly Averages	**26.21**	**57.19**	**45.851**

7 Conclusion

Considering the values in **Table 5.3** it's quite clear that in the current configuration the system fails to produce enough power to meet the complete load across the whole farm. However as far as viability is concerned; the system could successfully be implemented in a way that could cut reliance on grid electricity by a little under half on average and would incur minimal costs after the initial capital has been put forward, as the heat to run the cycle comes from a natural geothermal source. A different configuration or an alternative working fluid may get more performance out of the cycle, but as is the R134a Organic Rankine Cycle assessed above is clearly not without merit and would be quite viable given the above applications and circumstances.

8 Referencing

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