Optimisation of a Fin Fan Condenser in a Geothermal ORC Plant

Matthew J. Proctor[1], Wei Yu[1]*, Brent R. Young[1], Robert D. Kirkpatrick[1,2]

[1] Dept. Of Chemical and Materials Engineering, University of Auckland

[2] Independent Director, Ngawha Generation Limited

* w.yu@auckland.ac.nz

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ABSTRACT

A case study is performed on the fin fan condenser of one organic Rankine cycle (ORC) unit at the Ngawha Geothermal Power Station, Northland, to optimise the number of fans to maximise the economics of the ORC while maintaining safe operation.

A practical method is shown where first, key parameters of the fin fan condenser are identified followed by the use of these parameters in a plant model to predict the effect of varying the number of fans and condenser area on the power output of the entire ORC. The number of fans is determined using a heuristic that accounts for the trade-off between capital cost and revenue as well as safety issues that arise as a result of increasing the condenser duty. Finally, the economics of investing in additional condenser equipment is evaluated by calculating the payback period. This involves calculating the increased revenue compared to existing operation, taking into account real air temperature data for the plant, and estimating the capital cost of the additional equipment.

1. INTRODUCTION

The organic Rankine cycle (ORC) is a heat recovery technology that is particularly applicable to low temperature heat resources, such as geothermal (Ghasemi et al. 2012, Sohel et al. 2011, Madhawa Hettiarachchi et al. 2007).

A simplified ORC is shown in Figure 1. A working fluid is pumped to high pressure and vaporised by a heat source. It is then expanded, generating work, and condensed to a liquid to complete the cycle. The condenser of an ORC has a significant impact on the process operation as it determines the outlet pressure of the expander and thus has a substantial influence on its power output.

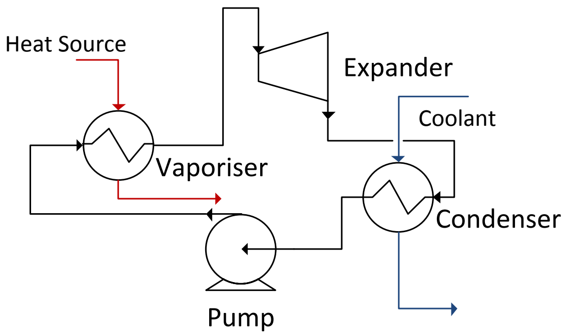

Figure 1. The basic organic Rankine cycle.

In this paper, the condenser of one of the ORC units at the Ngawha Geothermal Power Station will be used as a case study to determine what kind of improvements could be found by increasing the exchanger area and the number of fans. The condenser at Ngawha is air-cooled and so the amount of heat it can extract from the cycle is dependent on the air temperature, which varies from day to night and over the course of the year.

The rest of this paper is structured as follows: First some background on the plant and the model used to optimise the condenser is given. This is followed by the methodology used in this study. Finally the results are presented and the conclusions of the study are stated.

2. BACKGROUND

The ORC under investigation is located at the Ngawha Geothermal Power Station in Northland. It uses n-pentane as its working fluid and generates around 17 MW. The condenser has 45 fans and was originally designed using an inlet air temperature of 15 °C. The premise of this paper is that, based on the range of air temperatures the plant is exposed to, the condenser is undersized and the plant performance could be improved by increasing the number of fans and the surface area.

The effect of increasing the number of fans is to increase the air flow through the condenser, and to increase the surface area available for heat transfer since new tube bundles will also be added along with the additional fans. These changes will result in the cycle operating at a different set of process conditions, and this can be predicted by using a plant model.

A dynamic model of the Ngawha plant has been created (Proctor et al. 2013), and has also been validated against twenty four hours of real plant data, which is not discussed in this paper. The results of the validation show that the model is reasonably accurate, and so can be used to optimise the condenser. The model was constructed in VMGSim, a commercial process simulator, using design data and geometric data provided by the plant operators. A screenshot of the model is shown in Figure 2.

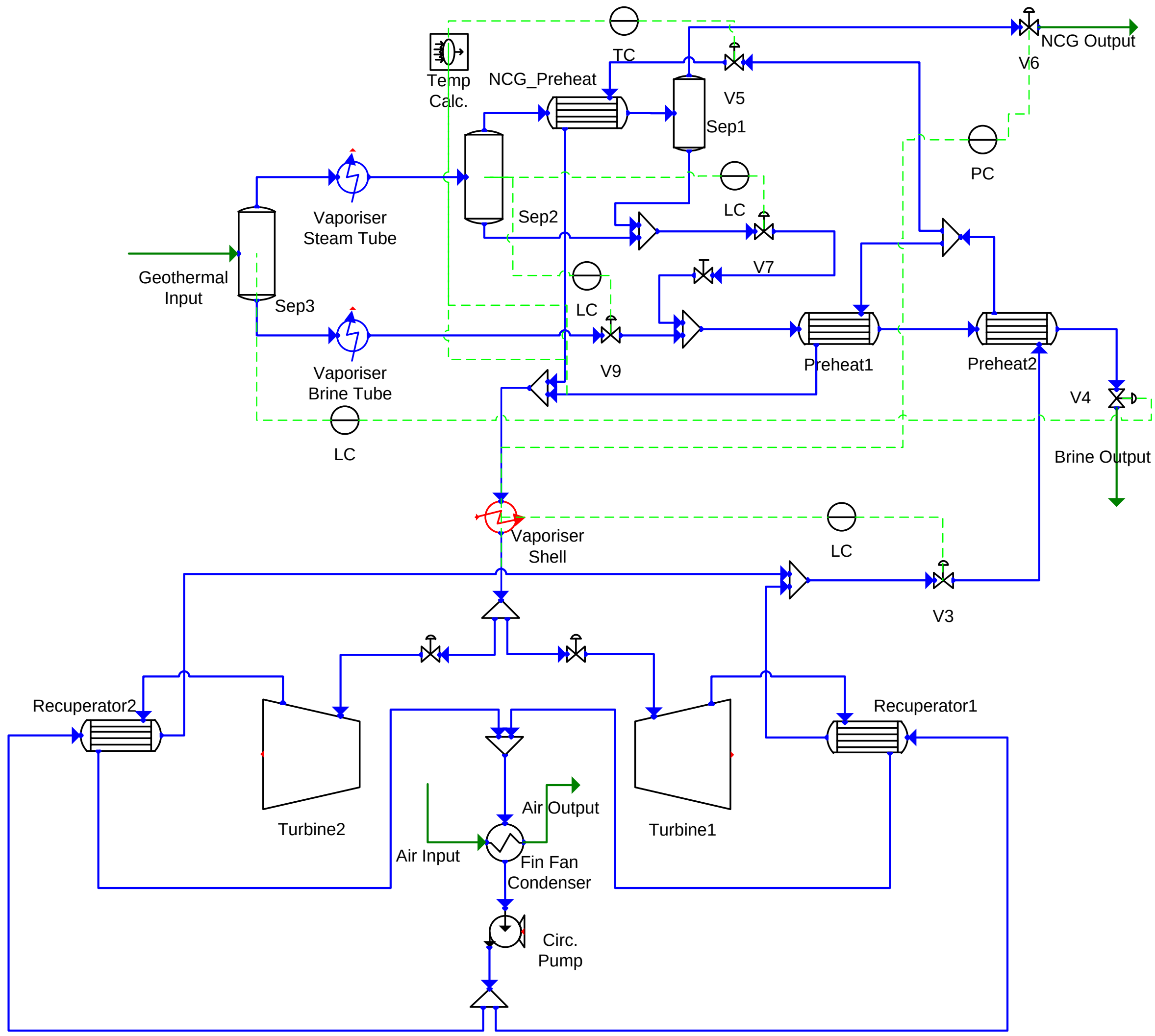

Figure 2. Layout of the plant model, from VMGSim.

Fin-fan condensers are a type of heat exchanger that uses air to cool a process stream. They consist of tube banks through which the process fluid will flow. The tubes have fins attached to their surface in order to increase the area available for heat transfer. Fans will be located either below (forced draft) or above (induced draft) the tubes and will create the air flow needed to cool the process fluid. At Ngawha induced draft is used.

3. METHOD

An outline of the method is as follows:

1. Estimate air temperature rise in fin fan condenser.
2. Using a plant model, determine net power and condenser pressure for a range of air inlet temperatures and number of condenser fans.
3. Find optimal number of condenser fans.
4. Calculate economic viability.

3.1 Air Temperature Rise

The design air temperature rise is unknown and must be determined in order to substitute the condenser overall heat transfer coefficient into the plant model.

The method used to determine the air temperature rise is outlined as follows:

1. Assume a condenser air temperature outlet.
2. Calculate the log mean temperature difference (LMTD).
3. Calculate the heat duty required from the plant design working fluid flow rate, and the heat of vaporisation of n-pentane.
4. Calculate the condenser overall heat transfer coefficient (U), the condenser's outside (finned) area is known.
5. Calculate the air inlet volumetric flow rate from the heat balance and air density.
6. Calculate the inlet air velocity, the inlet area is known.
7. Calculate the kinetic energy imparted to the air flowing through the condenser and compare with the fan design power.
8. Select an air outlet temperature that gives a reasonable estimate of the fan design power.
9. Calculate the air temperature rise.

3.2 Net Power and Condenser Pressure

Values for net power and condenser pressure over a range of inlet air temperatures and number of fans were determined by using a validated dynamic model of the Ngawha plant.

Air inlet temperatures of 8 °C, 15 °C, 21 °C and 25 °C and numbers of fans ranging from 45 to 100 were used in the model to calculate the results. The number of fans was increased in units of five until the maximum net power was reached, and then one or two more data points were calculated in order to confirm the maximum net power for a given inlet temperature.

Net power was calculated by subtracting the parasitic load of the condenser fans from the gross power reported by the plant model. The parasitic load for each fan was assumed constant, and was taken from the design value. This was used to find the parasitic load for the plant for different numbers of fans. The power required by the circulation pump for the working fluid is not taken into consideration because its power use remains at a relatively constant value in comparison to changes in gross power or the parasitic load of the fans.

The area of the condenser was increased from the original area by assuming a constant amount of area would be added for every additional fan. This was multiplied by a constant U value to determine the UA of the condenser for different numbers of fans. This U value is the one calculated in the previous step by assuming an appropriate air temperature rise. It is assumed to remain constant regardless of the flow on either side of the heat exchanger; validation of the model with plant data has shown this to be a reasonable assumption.

The air flow for different air inlet temperatures was calculated by assuming the value of ρv^2 (the air density × the square of the inlet velocity) would remain constant. The air density was calculated for the different air inlet temperatures and so the inlet velocity and thus the air volumetric flow rate were found.

The inputs to the model were the condenser UA, air inlet temperature and the air inlet volumetric flow rate. Using these values the gross power output and condenser pressure were returned by the model. The gross power output was subtracted by the parasitic load to find the net power output.

3.3 Number of Condenser Fans

The number of condenser fans was found by using the following heuristic: select the number of fans that returns a condenser pressure of one atmosphere at the 95 % air inlet temperature. Assuming the 95 % temperature is a commonly accepted design basis for air cooled heat exchangers.

This heuristic balances the increased net power created by adding more fans with the capital cost and also the safety issues created by operating at vacuum pressures. This will be discussed further in the results section.

3.4 Economic Viability

The economic viability of the selected number of fans was assessed by calculating the payback period for the investment in the additional fans:

$$Payback\ (years) = \frac{Capital\ Cost}{Revenue\ (p.a.)}$$

Payback was calculated at the number of fans indicated by the heuristic as well as ten more fans and ten less fans, in order to get some idea of the sensitivity of the economics to the number of fans, and the effectiveness of the heuristic

The capital cost per fan was estimated to be $50,000 NZD. This value includes everything associated with the fan, such as the tube bundle, motor, etc.

The revenue was calculated using a price per MWh of $70, which was decided after examining the prices given at the website www.em6live.co.nz, which provides electricity market information. The net power increase compared to the existing condenser was calculated at the four air inlet temperatures that were tested with the model. The revenue per annum if the air temperature were held constant was calculated, and then plotted against inlet air temperature. A logarithmic curve was then fitted to these results. This curve gives the relationship between revenue per annum and inlet air temperature.

Air temperature data for the Ngawha plant was used to determine the frequency at which each air temperature is present. The average of this distribution is 14.2 °C and the standard deviation is 3.7 °C. The results are shown in Figure 1.

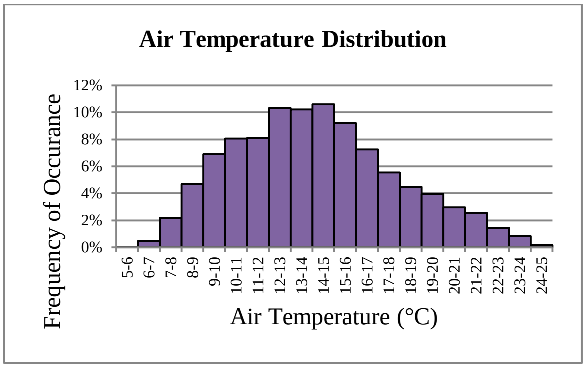

Figure 3. Histogram of the air temperature range at the plant.

The revenue per annum at each air temperature in the distribution (calculated from the logarithmic fitted curve) was multiplied by the relative frequency to calculate overall revenue per annum for each given number of fans.

5. RESULTS AND ANALYSIS

Table 1. The results of the air temperature analysis.

Variable	Unit	Case 1	Case 2	Case 3
Air Inlet Temperature	°C	15	15	15
Air Outlet Temperature	°C	20	25	30
Condenser Pressure	kPa	100	100	100
Pentane Temperature (assumed constant, due to phase change)	°C	35.5	35.5	35.5
LMTD	°C	17.9	14.9	11.4
Heat duty	W	7.26E+07	7.26E+07	7.26E+07
UA	W/°C	4.06E+06	4.86E+06	6.37E+06
Total area	m^2	1.02E+05	1.02E+05	1.02E+05
U	$W/m^2 \cdot °C$	40	48	63
Air density @ 15 °C	kg/m^3	1.209	1.209	1.209
Air inlet volume flow	m^3/s	4.28E+07	2.14E+07	1.43E+07
Inlet area	m^2	738.8	738.8	738.8
Inlet velocity	m/s	16.1	8.1	5.4
ρv^2	J/m^3	313.8	78.4	34.8
Kinetic power (for all fans)	kW	3734	466	138
Assuming 70 % Efficiency	kW	5335	666	197

Three possible air outlet temperatures were examined: 20 °C, 25 °C and 30 °C. Depending on which outlet temperature was chosen, the U for the condenser varies. It is known that the design power for the condenser is 820 kW. By calculating an estimate of the fan power needed to provide the necessary air flow rate an air outlet temperature of 25 °C is selected as this gives a result (666 kW) closest to the real design power. A value of 70 % efficiency was assumed due to similar fan efficiencies being used in the GPSA handbook for fin fan design (Gas Processors, Gas 1972).This means an air temperature rise of 10 °C appears to best fit the available data out of the three possibilities. It is possible that an air temperature rise of 8 or 9 °C is a better fit; however there will be additional losses in the fans as not all the mechanical work is used to increase the axial component of the kinetic energy of the air, and thus the fan power is likely to be underestimated. Therefore a rise of 10 °C is assumed for the rest of this analysis.

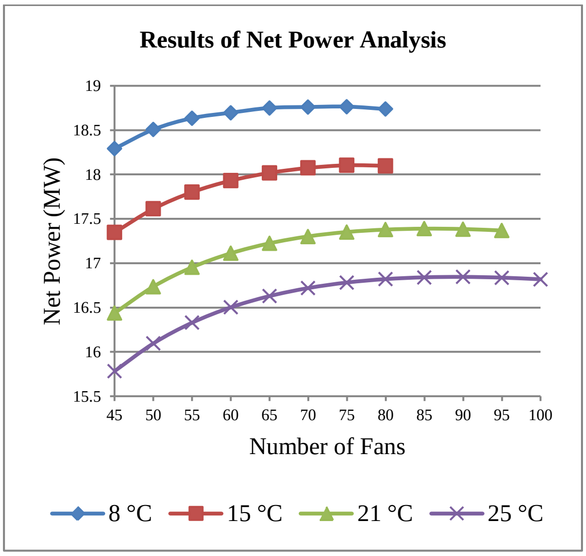

Figure 4. Model results showing net power for a range of inlet air temperatures and number of fans.

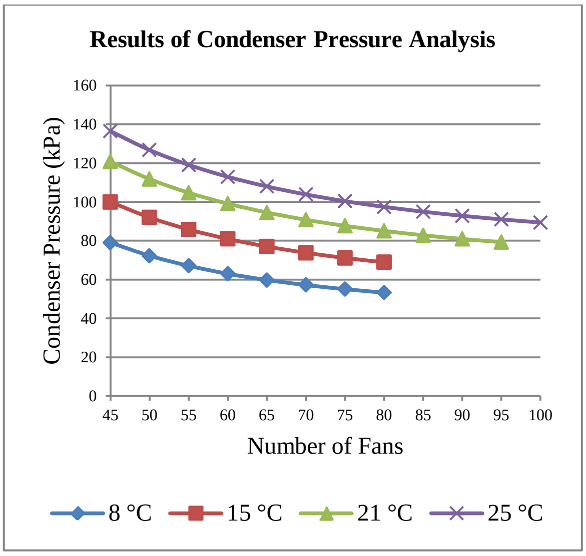

Figure 5. Model results showing condenser pressure for a range of inlet air temperatures and number of fans.

The results from the model showing the net power and condenser pressure for a range of inlet air temperatures and number of fans are shown in Figure 2 and Figure 3.

As can be seen the maximum net power is achieved at around 75 fans for 8 °C and 15 °C, at 85 fans for 21 °C and at 90 fans for 25 °C.

Based on the air temperature data available it was determined that 95 % of data points were below 21 °C. By using the heuristic that the design should have an atmospheric condenser pressure at this temperature the number of fans selected for further analysis was set at 60.

Table 2. Economic results.

# Fans	50	60	70
Additional Fans	5	15	25
Capital Cost (000s NZD)	250	750	1250
Increased Revenue (000s NZD p.a.)	113	340	422
Payback (years)	2.22	2.21	2.96

Economic analysis was done for 50, 60 and 70 fans (an increase of 5, 15 and 25 respectively). The results are presented in Table 2. It can be seen that the heuristic has chosen a reasonable number of fans that balance capital cost with revenue. The difference between 50 and 60 fans is minimal, however increasing the number of fans to 70 results in an increase in the payback period by ¾ of one year.

Another consideration is operating the condenser at vacuum conditions. Depending on the air temperature the condenser pressure varies between 80 and 140 kPa at the current condenser size with 45 fans. If the number of fans were increased the condenser would operate at vacuum for a larger air temperature range. This could cause safety issues because the working fluid, pentane, is flammable. When operating at vacuum air would leak into the system which poses an explosion risk. It would be necessary to include equipment to purge the air that leaked into the working fluid to mitigate this risk. By using the heuristic the minimum condenser pressure at cold temperatures is reduced to around 60 kPa. Further investigation is needed to determine the effect this would have on the existing systems used to purge air from the working fluid, however selecting 60 fans instead of attempting to maximise the net power output is a more conservative improvement to the condenser considering the lack of experience in operating the plant at these low pressures.

6. CONCLUSIONS

It is important to note that this work is a paper exercise and no changes are intended for the real plant.

The conclusions of this study are as follows:

- Improved performance of the plant could be achieved by increasing the number of fans on the condenser (including increasing the amount of surface area) from 45 to 60.
- This would require an investment that is estimated to be approximately $750k, with a payback period of 2.21 years.
- An even higher number of fans may produce more net power, but this requires further investigation of air leakage at vacuum pressure and will increase the payback period due to the diminishing return on net power output for each additional fan, while the capital cost is expected to remain relatively constant.

7. FUTURE WORK

One area where it may be possible to improve the accuracy of the air temperature rise prediction is in predicting the fan power. Currently this was done by calculating the amount of power needed to increase the kinetic energy of air to the inlet velocity from rest. If the static pressure drop across the fans can be calculated, then a more reliable estimate of the fan power can be found, which will improve the accuracy of the air temperature rise (and subsequently the UA of the condenser).

Another area where there may be the potential for improvement is implementing variable speed drives on the fans. This would allow the speed of the fans to be varied in response to the air temperature. This could allow improvements in the net power produced by the plant, but would come at the cost of increased up-front investment in equipment. Related to this is looking at the economics of increasing the amount of fans in the condenser, but turning off fans at low air temperatures to prevent very low condenser pressures. This will require more investigation on the effect of these low pressures on air leaking into the working fluid to decide when it would be worthwhile to turn fans off.

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