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Effect of Air Temperature Variation on the Performance of Wet Vapour Organic Rankine Cycle Systems

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Keywords: Screw expander, two-phase expansion, power generation, optimisation, organic Rankine cycle

Abstract

A multi-variable optimization program has been developed to investigate the performance of Wet Organic Rankine Cycles (WORC) for generating power from low temperature liquid dominated brines. This cycle model contains a detailed thermodynamic model of a twin-screw expander, and the methods used to match the operation of the expander to the requirements of the cycle are described. Optimum operating conditions are calculated for a particular design point, which specifies the required size of heat exchangers and the port geometry and operating speed of the expander. Performance at off-design conditions can then be optimized within these constraints. This allows a rigorous investigation of the effect of air temperature variation on performance of WORC systems. The capability of the cycle model has been demonstrated for the case of power generation from a brine heat source at 120°C, assuming typical air temperature conditions for Nevada, USA. There are two main findings from the paper. Firstly, optimization of the WORC system using the annual average air temperature of 10.5°C achieves maximum power output with 75% dry working fluid at the inlet to the expander. Secondly, analysis of the off-design performance of the system shows that positive net power output is possible for air temperatures up to around 40°C. The estimated average power output over the course of a year was only 3.4% smaller than the power generated at the average annual temperature of 10.5°C. This confirms that a single calculation of WORC system performance using the average temperature for the region gives a good estimate of the expected average annual power output of the system. For the resource conditions assumed, screw driven WORC systems can be built with net power outputs of the order of 600kW, using standard size machines.

1. Introduction

The Organic Rankine Cycle (ORC) provides a means of recovering useful energy from low temperature heat sources. In comparison with conventional high temperature steam Rankine cycles, the low temperature of these heat sources means that the attainable cycle efficiency is much lower, while the required surface area of the heat exchangers per unit power output is much higher. The lower latent heat of evaporation of organic fluids relative to steam also means that the feed pump work required in ORCs is a significantly higher proportion of the gross power output.

Especially at lower source temperatures, up to approximately 120°C inlet, the only cycle normally considered is that where the working fluid enters the expander as dry saturated vapour, as shown in Figure 1. However, in most cases, this leads to the working fluid leaving the expander with some superheat, which must be removed before condensation begins.

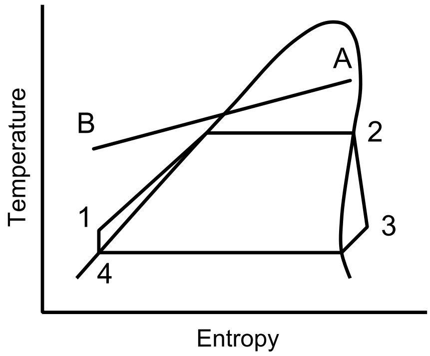

Figure 1: Illustrative T-s diagram showing conventional ORC with dry saturated vapour at the expander inlet

Maximising net power output is a compromise between increasing the mean temperature of heat addition (which, in accordance with Carnot's principle, can increase cycle efficiency) and increasing the amount of heat extracted from the source, which requires a lower evaporation temperature.

By the use of a screw expander, instead of the more conventional turbine, it is possible to admit the working fluid to the expander as wet vapour and thereby eliminate both the need to desuperheat the vapour after expansion and, simultaneously to raise the evaporation temperature, as shown in Figure 2, thus improving the cycle efficiency. The potential cost and performance benefits of using screw expanders in ORC systems have been extensively studied for geothermal applications by Smith et al. (2001, 2004, 2005).

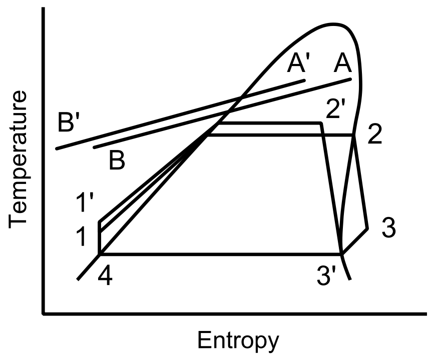

Figure 2: Illustrative T-s diagram showing how the need to cool superheated vapour in a conventional ORC system can be avoided by expanding wet vapour

However, screw expander efficiencies are more sensitive to expansion pressure ratio than turbines and the expansion ratio increases as the expander inlet vapour dryness fraction decreases. To determine the value of inlet dryness fraction that leads to the maximum system power output, it is therefore necessary to include estimates of how both the screw expander and the feed pump performance vary as the inlet dryness fraction of the working fluid is changed in such a wet ORC (WORC) system.

A further consideration that must be included, is the pump or fan power required to drive the coolant across the condenser coils. This increases with the coolant flow rate, but a higher coolant flow rate permits a lower condensing temperature for the same condenser pinch point temperature difference and, hence, a higher cycle efficiency and gross power output. The best combination of these conflicting factors to obtain the condensing temperature that yields the maximum net power output needs also to be estimated.

In reality there is some deviation from the idealised Rankine cycles shown in Figure 2 due to pressure drops in the heat exchangers and the requirement for sub-cooling of the condensed working fluid to avoid cavitation in the feed pump. While these effects do not generally have a large influence on the performance of individual components, the sensitivity of the cycle performance to the operating conditions means that they should also be considered in the analysis of low temperature heat recovery systems.

A further consideration is the size of the heat exchangers, which will depend on the heat transfer rate, the heat transfer coefficients of the fluids and the temperature difference between the fluids. This affects the overall system cost and must therefore be taken into consideration in a full evaluation of the system choice.

In practical applications of low temperature heat recovery there is usually a minimum allowable discharge temperature for the source fluid, especially in geothermal power generation, where solid precipitates can form. If required, this lower limit must be included as the cut-off point in evaluating the whole system.

The potential of the WORC as a cost effective system for power recovery from low temperature heat sources was investigated by Leibowitz et al. (2006), but only limited cycle optimisation was performed using a simple expander model with constant efficiency. The overall requirement for evaluating the potential of WORC systems for geothermal applications is therefore for a computer program which simulates the performance of each of the components and applies a multi-variable optimisation procedure.

1.1 Screw Expanders for ORC Systems

The efficiency of the twin-screw expander depends on a range of factors including the expander dimensions, rotor profiles, inlet and discharge pressures, inlet dryness fraction and rotational speed. Another important factor for screw machines is the built-in volume ratio (BIVR), which is the ratio between the volume of the working chamber at the points where the inlet port closes and the discharge port opens. All of these variables must be chosen in order to match the mass flow rate through the machine to that required in the cycle. It is therefore essential that the ORC system analysis should capture the operating characteristics of the expander to allow optimisation for a particular application; this can only be achieved through numerical modelling of the complete system using detailed thermodynamic models for key components.

2. Analysis of Wet ORC Systems

A computational ORC model has been developed using a well-established quasi one dimensional model of twin-screw machines and with thermo-fluid properties obtained from the ‘Reference Fluid Thermodynamic and Transport Properties Database’ (REFPROP) program produced by the National Institute of Standards and Technology (NIST). Other cycle components such as the feed pump and motor have been characterised using performance data from manufacturers, while pressure loss in the heat exchanger components has been estimated. Multi-variable optimisation has been implemented using an evolutionary algorithm and defining a target function for the optimum values of all the cycle variables identified. This optimisation program has been used to identify the cycle conditions that result in maximum net power output for a range of applications. This paper illustrates how the optimum cycle conditions can be identified for a given temperature and flow rate of the heat source fluid, and the resulting net power output, expander operation and required heat exchanger surface areas for the WORC system.

2.1 Thermodynamic Cycle Model

The performance of ORC systems has been assessed using a computational model of the cycle. This has been written as an object-oriented program in the C# language, which provides a convenient structure as it allows a generic description of heat sources, heat sinks and cycle components. Each of these cycle elements contains definitions for all the necessary input and output parameters along with the required calculations. Both simple cycles such as those shown in Figure 2, and more complex cases (including multiple heat source streams, multiple paths for the working fluid or varying working fluid composition) can be analysed by creating models of the required components and providing the necessary input parameter values.

The current study investigates a WORC system as illustrated in Figure 2. Simplified heat exchanger models have been used with specified pressure loss factors. The minimum allowable temperature differences in the boiler and condenser heat exchangers have also been specified as an input to the model. The efficiency of the feed pump has been characterised as a function of volumetric flow rate using data from manufacturers.

An important output from the cycle model is the required area of the heat exchangers, as this is necessary for the analysis of system performance at off-design conditions described in Section 4.2. This has been estimated by assuming constant heat transfer coefficients for the boiler and condenser. The heat exchanger surface areas required for the cycle can therefore be considered proportional to the integrated values of Q/LMTD for the heat exchangers, where Q is the rate of heat transfer and LMTD is the log mean temperature difference between the fluids.

2.2 Twin-Screw Expander Model

A full thermodynamic model of the expander has been created for integration with the cycle model. This is based on the quasi one dimensional analysis of twin-screw machines as described by Stosic and Hanjalic (1997a, 1997b), which has been extensively validated for compressors (and, to a lesser extent, expanders (Smith et al., 1996)) for a wide range of working fluids and operating conditions. Using this procedure, machine geometry and rotor profiles have been optimised for a particular set of operating conditions representative of those considered in this paper, and have been fixed for the purposes of the current study. The City University 'N' rotor profile described in Figure 3 has been used in the current analysis as this geometry is known to be superior to other well-known types.

In principle, the geometry optimisation could be integrated with the cycle analysis described here to ensure the best profile is used for the required operating conditions, but this would be very computationally intensive and is not expected to significantly affect results. Furthermore, from a manufacturing perspective it would be prohibitively expensive to produce an optimised machine for every different application. For a specified geometry, the characteristics of the twin-screw machine such as the curve of working chamber volume against angular position, sealing line lengths, blowhole area and axial/radial clearances between the rotors and the casing are therefore defined as fixed inputs for the expander model. The variable input parameters required for the expander model within the cycle analysis program are then limited to the main rotor speed, BIVR, inlet dryness fraction, and inlet and discharge pressures. In order to integrate this expander model with the main cycle model, two approaches can be taken to match the machine operation to the cycle conditions:

i. The BIVR is specified and iterations are performed to find the rotor speed required to match the mass flow rate of the expander to that of the working fluid calculated in the cycle model – no limits are imposed on rotor speed, which is some cases can become impractically high.

ii. The rotor speed is fixed and iterations are performed to find the value of BIVR required to match the mass flow rate – if the BIVR is greater than the limit for the chosen screw machine geometry then the expander cannot meet the requirements of the cycle conditions, and the expander efficiency is set to zero.

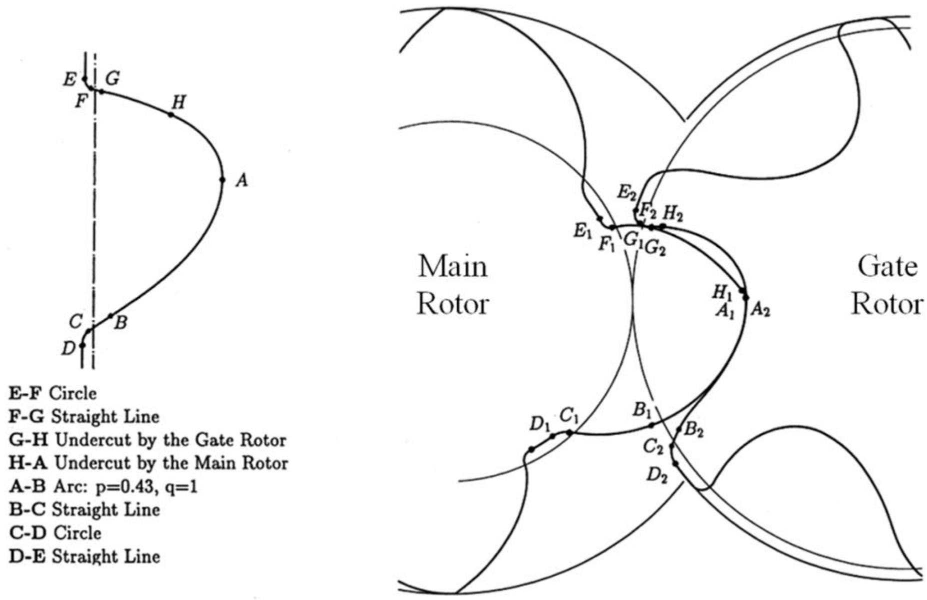

Figure 3: Description of the City University 'N' rotor profile for positive displacement screw machines

In general, the expander efficiency can be calculated using either of these approaches. For the analysis presented in this paper, the expander operating speed has been fixed at a constant value of 50 m/s (known to be a sensible maximum value for the application) and method (ii) has therefore been used. The expander efficiency resulting from either method is used in the thermodynamic cycle model to calculate overall cycle performance for specific operating conditions. An iterative numerical procedure can then be used to identify the optimum operating conditions for the cycle.

2.3 Optimisation Procedure

An evolutionary algorithm has been used to identify the optimum operating conditions for the cycle model. This is a flexible and stable numerical approach which allows for optimisation with any number of variables and is particularly good for distinguishing global from local maxima and coping with discontinuities in the target function. A population of solutions is defined in which each individual solution has a unique 'gene' consisting of a 'chromosome' for each of the cycle optimisation variables under consideration (e.g. boiler pressure, condenser pressure, dryness fraction at expander inlet and BIVR). The values of the chromosomes are initially randomly generated, and a function (in this study, the net power output of the cycle) is defined in order to calculate the 'fitness' of a particular solution. Over successive generations of the calculation procedure, 'fitter' genes are used to create new solutions through both combination and random mutation of the chromosomes. After a specified number of generations, the single best solution is identified as the optimum.

2.4 Off-Design Analysis

The optimisation performed for the operation of the WORC at the design-point gives the following system parameters:

i. Expander BIVR and rotor speed,
ii. The total values of Q/LMTD for the boiler and condenser heat exchangers.

While it is possible for screw machines to have variable BIVR, this adds considerable cost and complexity to the machine, and a fixed BIVR is preferable. If a synchronous generator is used with the expander then the rotor speed must also remain constant under all operating conditions. Likewise, the values of Q/LMTD are assumed to define the required size of the heat exchangers, which cannot easily be changed during operation. These parameters represent physical system components, and must therefore remain constant when considering the performance at off-design conditions.

The WORC cycle model with evolutionary optimisation can again be used for the off-design analysis, but the fitness function must be changed in order to reflect the new constraints on the system. The aim of the off-design optimisation is to maximise power while ensuring that the heat exchangers and expander are the same in all cases. The variable parameters that can be optimised in order to maximise net power output are limited to the following:

i. Pinch point temperature differences in the boiler and condenser heat exchangers,
ii. Boiler and condenser inlet pressures,
iii. Expander inlet dryness fraction.

Values for these input parameters are specified by the evolutionary optimisation program, and the performance of the WORC is calculated in terms of the net power output, the values of Q/LMTD for the heat exchangers and the ratio of mass flow rate required in the cycle to that calculated by the expander model for the specified operating conditions. Error values for the heat exchangers and expander are then calculated according to the following definitions;

The total error, e_{hx}, between the calculated and design-point (DP) heat exchanger areas, where;

$$e_{hx} = \left|\frac{\left(\frac{Q}{LMTD}\right)_{boiler,calc}}{\left(\frac{Q}{LMTD}\right)_{boiler,DP}} - 1\right| + \left|\frac{\left(\frac{Q}{LMTD}\right)_{cond,calc}}{\left(\frac{Q}{LMTD}\right)_{cond,DP}} - 1\right|$$

The error, $e_{\dot{m}}$, between the calculated mass flow rates of working fluid in the cycle and the expander model (for the fixed values of BIVR and rotor speed), where;

$$e_{\dot{m}} = \left|\frac{\dot{m}_{cycle}}{\dot{m}_{expander}} - 1\right|$$

By specifying a maximum allowable tolerance, e_{tol}, for the total error in heat exchanger areas and mass flow rate, the fitness function to be maximised, F, can be defined as follows;

$$If\ (e_{hx} + e_{\dot{m}} < e_{tol}),\ \ F = \ P_{net}$$

$$Otherwise,\ \ F = P_{net}\left(\frac{e_{tol}}{e_{hx} + e_{\dot{m}}}\right)$$

This fitness function results in a continuous decrease in the fitness value as the total error in the heat exchanger areas and mass flow rate increases beyond the allowed tolerance, and has been found to allow accurate optimisation of the WORC system.

The optimisation of the system requires many calculations to be performed using the full thermodynamic expander model discussed in Section 2.2. This is very computationally intensive, and in order to speed up the calculation procedure an initial optimisation was performed using an approximate relationship between expander efficiency and the 'expansion ratio', defined as the BIVR of the expander divided by the overall specific volume ratio for the expansion process. An example of this is shown in Figure 4, where a second order polynomial is fitted to a dataset calculated for a particular expander size, BIVR and rotor speed, with a range of inlet dryness fractions, inlet and discharge pressures.

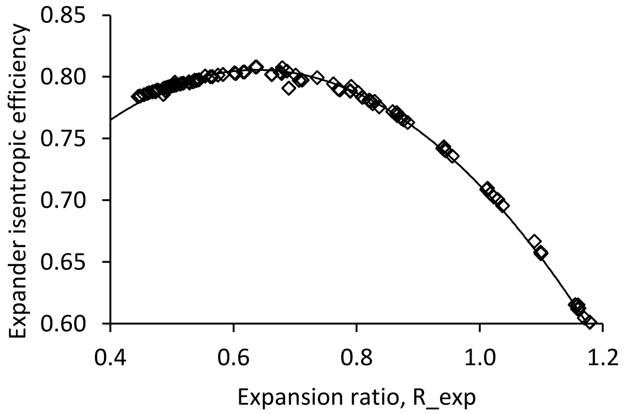

Figure 4: Fitted curve showing an example of the approximate relationship between expander isentropic efficiency and the expansion ratio (R_{exp} = overall specific volume ratio of expansion / BIVR)

The approximation of expander efficiency shown in Figure 4 can be used in an initial optimisation to identify the operating condition required to achieve maximum power output for a specified boiler pressure. A single calculation of the system performance at these conditions, using the full expander model, then allows the estimate of efficiency to be improved and the error in mass flow rate to be calculated. The boiler pressure is then adjusted and the process repeated until the mass flow rate error is less than the specified tolerance. This initial optimisation significantly reduces the number of times the computationally intensive thermodynamic expander calculation is performed, and provides the initial conditions for a full final optimisation to identify maximum power output.

3. Results of Case Study

In order to demonstrate the cycle analysis described in Section 2, a simple case study has been performed for the recovery of heat from a geothermal brine source fluid. This liquid stream has an inlet temperature of 120°C and contains a recoverable heat content of 2.7 MW if cooled to an ambient temperature of 10°C; however, a minimum allowable brine temperature of 70°C has been imposed as this represents a typical limit for controlling the formation of precipitates.

This study has investigated the generation of power from this heat source using a WORC with the following characteristics:

i. The working fluid is refrigerant R245fa.
ii. The expander is a twin-screw machine with a main rotor diameter of 204 mm and a maximum allowable built-in volume ratio of 4.5. A fixed rotor tip speed of 50m/s has been used in all calculations.
iii. An air cooled condenser is used with 2°C sub-cooling of the working fluid at the exit.
iv. Minimum pinch point temperature differences of 5°C and 10°C respectively have been applied for the boiler and condenser for the design-point optimisation
v. An efficiency of 95% has been assumed for the electrical generator and 90% for pump and fan motors.

While this is a relatively low power system, the thermodynamic model of the 204 mm expander has been validated using experimental results from testing of this size of machine. The results of the design point and off-design optimization of the WORC system using this expander are described below, while predicted results for a larger scale system are presented in the discussion section.

3.1 Design point optimization

Operation of the WORC system has been considered for average climate conditions in Nevada, USA, where the annual mean temperature is 10.5°C, with monthly variations in the average maximum and minimum temperatures shown in Figure 5. A design-point optimisation has been performed for the annual mean temperature; the fixed parameters for this optimisation are shown in Table 1, and the results are shown in Table 2. The maximum net power output from the geothermal brine was found to be 85.6kW, and was achieved with an expander inlet dryness of 75% and a BIVR of 3.54.

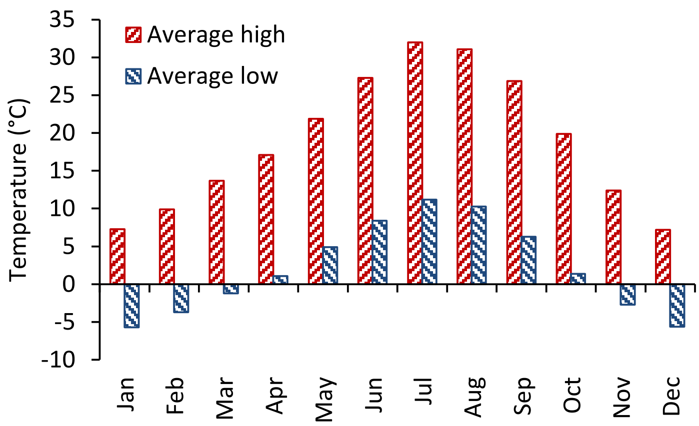

Figure 5: Monthly average maximum and minimum air temperatures in Nevada, USA

Working fluid	-	R245fa
Main rotor tip speed	m/s	50
Main rotor diameter	mm	204
Boiler pinch point	°C	5
Condenser pinch point	°C	10
Condenser air inlet temperature	°C	10.5
Brine inlet temperature	°C	120
Minimum allowable brine temperature	°C	70

Table 1: Fixed parameters for design point optimisation of WORC system

Boiler saturation temperature	°C	86.6
Condenser saturation temperature	°C	28.5
Expander isentropic efficiency	%	80.4
Expander inlet dryness	%	75
Expander exit dryness	%	91
Working fluid mass flow rate	kg/s	6.44
Expander power (electrical)	kWe	109.4
Condenser fan power (electrical)	kWe	17.8
Feed pump power (electrical)	kWe	6.0
Net power output (electrical)	kWe	85.6
Brine exit temperature	°C	70.0
BIVR	-	3.94
Feed pump efficiency	%	72.0
Feed pump motor efficiency	%	89.6
Q/LMTD boiler	kW/K	79.1
Q/LMTD condenser	kW/K	82.8

Table 2: Optimised parameters for WORC operating at the design point conditions stated in Table 1

3.2 Off-design optimization

The off-design performance of the optimised WORC has been investigated by identifying the conditions required to achieve maximum net power output from the system defined in Table 2 for a range of air temperatures from -7.5 to 35°C. In all cases, the off-design analysis has achieved an error of less than 0.1% between the off-design and the design-point values of Q/LMTD for the boiler and condenser heat exchangers, and an error of less than 0.1% between the mass flow rate of working fluid calculated in the cycle model and that predicted by the expander model using the specified values of BIVR and rotor speed. The required input parameters and the resulting system performance for off-design operation are shown in Figures 6-12.

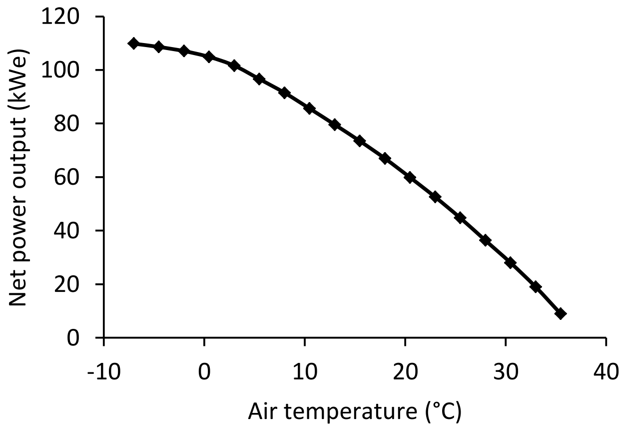

Figure 6: Maximum net power output as a function of air temperature at off-design conditions using WORC optimised for 10.5°C air temperature

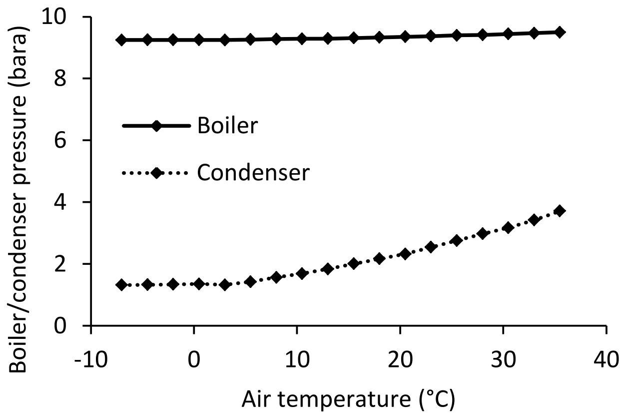

Figure 7: Boiler and condenser pressure required for maximum net power output at off-design conditions using WORC optimised for 10.5°C air temperature

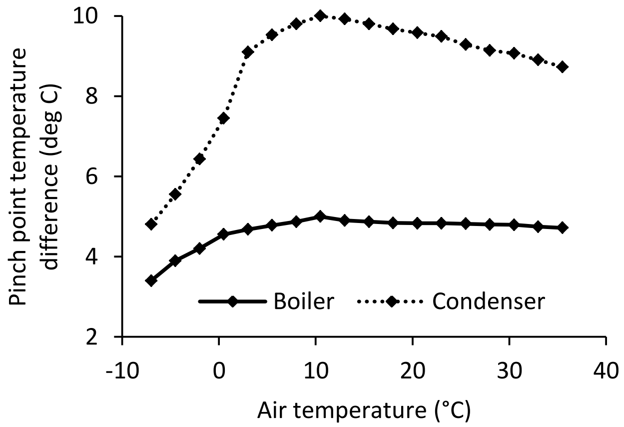

Figure 8: Boiler and condenser pinch point temperature differences required for maximum net power output at off-design conditions using WORC optimised for 10.5°C air temperature

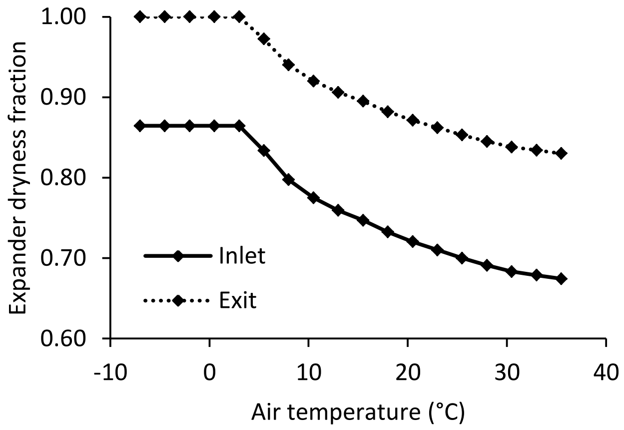

Figure 9: Expander inlet dryness fraction required for maximum net power output at off-design conditions using WORC optimised for 10.5°C air temperature, and the resultant exit dryness fraction

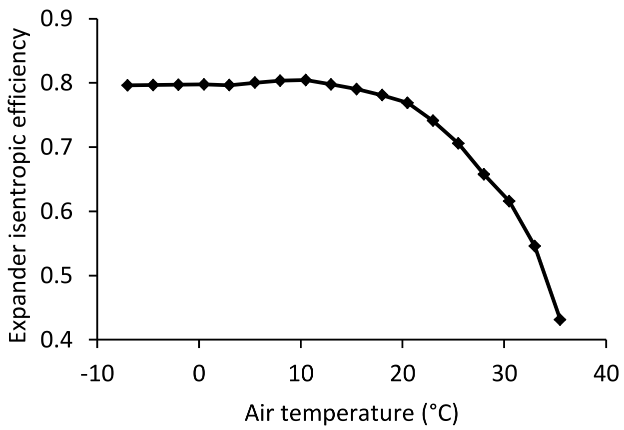

Figure 10: Expander isentropic efficiency achieved when net power output is maximised at off-design conditions using WORC optimised for 10.5°C air temperature

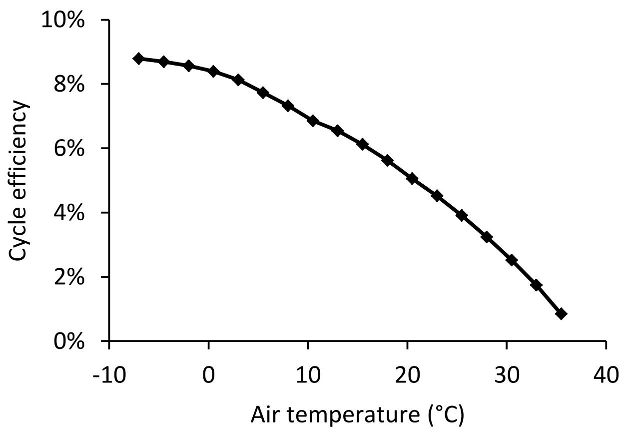

Figure 11: Cycle efficiency achieved when net power output is maximised at off-design conditions using WORC optimised for 10.5°C air temperature

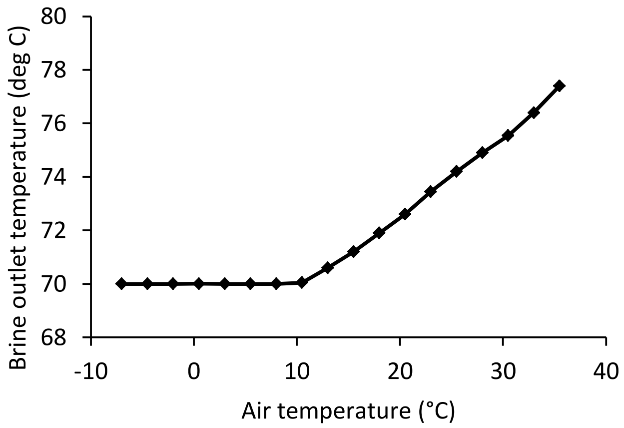

Figure 12: Geothermal brine outlet temperature when net power output is maximised at off-design conditions using WORC optimised for 10.5°C air temperature

In order to assess the effect that the off-design performance has on the operation of the WORC system throughout the year, it has been assumed that for a typical day, the temperature has a sinusoidal variation between the average monthly maximum and minimum temperatures as shown in Figure 13. The variation in power with temperature through the course of a typical day in each month can then be calculated, and the mean power output for each month can be found. Figure 14 shows the maximum and minimum net power output from the WORC at the average monthly minimum and maximum temperatures respectively, and the average power output for the month.

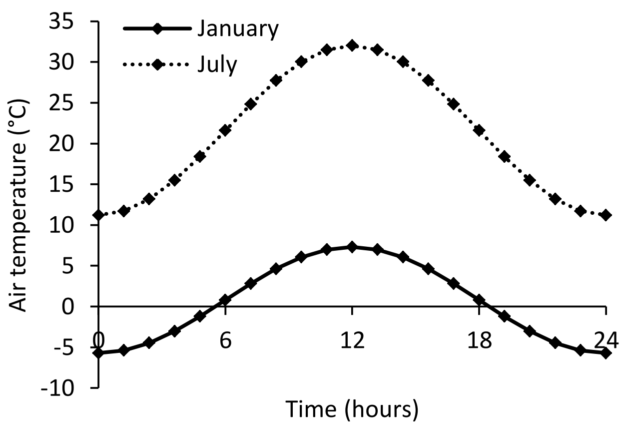

Figure 13: Assumed sinusoidal variation of air temperature between average monthly maximum and minimum temperatures for a typical day in January and July in Nevada, USA

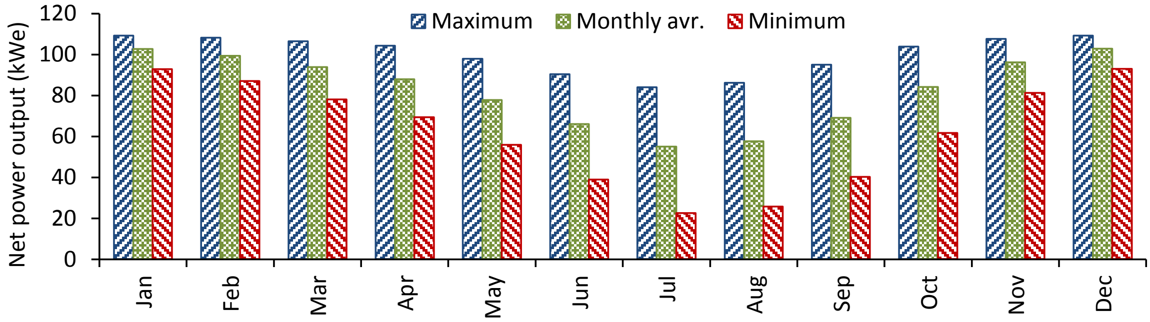

Figure 14: Maximum, minimum and time-averaged monthly net power output from WORC system optimised for 10.5°C air temperature

4. Discussion

The results presented in Section 3 show that variation in air temperature has a large effect on the net power output from the WORC system. Off-design operation can be optimised by allowing variation in the inlet pressure and pinch point conditions in the boiler and condenser, and by varying the dryness fraction at the inlet to the expander. In all cases the BIVR and rotor speed of the expander is kept constant.

The main effect of increasing air temperature on the cycle conditions is the increase in condenser pressure. This decreases the overall volume ratio for the expansion process. The BIVR of the expander is however fixed, which at higher temperatures (over 25°C) leads to over-expansion of the working fluid and a decrease in expander efficiency (as shown in Figure 10), reducing the net power output.

As air temperature decreases the required expander inlet dryness fraction tends to increase. The optimisation has been performed with the constraint of wet or dry-saturated vapour at the expander outlet in order to avoid the need to cool superheated vapour. Maximum net power output is achieved at this limiting case of dry-saturated expander exit vapour when the air temperature falls below 5°C. The boiler and condenser pressures are then approximately constant, and the required values of Q/LMTD for the heat exchangers are then achieved by decreasing the pinch point temperature differences.

The results in Figure 14 show that the WORC system optimised for the average air temperature of 10.5°C has a positive net power output under all typical temperature conditions. Power output is greatest during the winter months, where there is little daily variation between maximum and minimum values. During the summer months, power generation in the hottest part of the day can however be as little as 27% of the daily maximum. When averaged over the year, the power output is calculated as 82.6kWe. This is 3.4% smaller than the power output of 85.6kWe calculated at the design point using the average annual temperature of 10.5°C.

While these results are for a 204mm screw expander, which has already been built and tested, the performance of a larger scale geothermal power plant utilising brine with the same inlet and minimum temperatures can be estimated for larger machines of this type. Standard screw machine sizes up to around 500mm are currently produced. Using the model presented in this paper, the performance of a 512mm screw expander at the optimum operating conditions identified for the WORC is predicted to achieve expander adiabatic efficiencies of around 80% and a net power output of around 590kWe.

5. Conclusions

For the conditions considered in the case study, this paper demonstrates that the optimization of the WORC system using the annual average air temperature of 10.5°C can achieve a maximum net power output of 85.6kWe with 75% dry working fluid at the inlet to the expander. At these conditions, the isentropic efficiency of the expander is over 80% and an overall cycle efficiency of 7% is achieved.

The analysis of the off-design performance of the system shows that positive net power output is possible for air temperatures up to around 40°C. By calculating the variation in power output with temperature and averaging over the course of a year, it was found that the annual average power was only 3.4% smaller than the power generated by the system optimized for the average annual temperature of 10.5°C. This is the key finding of the study, as it suggests that a single calculation of WORC system performance using the average temperature for the region gives a good estimate of the expected average annual power output of the system.

For the resource conditions assumed, screw driven WORC systems can be built with net power outputs of the order of 600kW, using standard size machines.

While the results of the case study do not present a definitive design for a geothermal application, they demonstrate the capability of the current cycle analysis and optimisation method. In reality, additional factors need to be considered. These include the choice of working fluid, available standard machine sizes and operating limits, the sizing and design of the heat exchangers and the cost of components. These issues will be addressed in future developments of the ORC model.

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