COMPARISON OF ORGANIC RANKINE CYCLE SYSTEMS UNDER VARYING CONDITIONS USING TURBINE AND TWIN-SCREW EXPANDERS

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ABSTRACT

A multi-variable optimization program has been developed to investigate the performance of Organic Rankine Cycles (ORCs) for low temperature heat recovery applications. This cycle model contains detailed thermodynamic models of the system components, and the methods used to match the operation of the expander to the requirements of the cycle are described. Two types of ORC system are considered; one containing a turbine to expand dry saturated or superheated vapour, and one with a twin-screw machine allowing expansion of partially evaporated fluid.

Modelling of the ORC system with a twin-screw expander has been described previously (Read et al. 2014a, 2014b). The performance of the turbine in the superheated ORC has been modelled using available operational data for single stage, reaction turbines, where correlations have been used to estimate the efficiency of the turbine at 'off-design' conditions using either fixed or variable nozzle geometries.

The capability of the cycle model has been demonstrated for the case of heat recovery from a source fluid at 120°C. The system parameters are optimised for a typical operating condition, which determines the required size of heat exchangers and the expander characteristics. 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 the system performance, and the seasonal variation in net power output for the turbine and twin-screw ORC systems is estimated.

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.

Maximising the net power output from an ORC 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.

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 1a. However, in most cases, this leads to the working fluid leaving the expander with some superheat, which must be removed before condensation begins. 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 de-superheat the vapour after expansion and, simultaneously to raise the evaporation temperature, as shown in Figure 1b, 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).

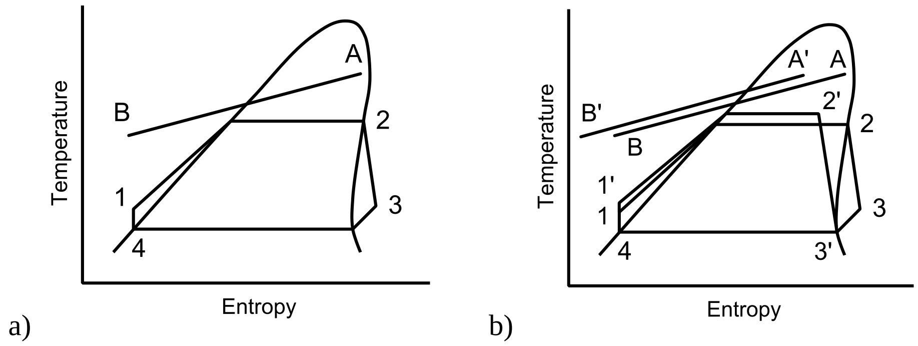

Figure 1: Illustrative T-s diagrams showing a) conventional ORC with dry saturated vapour at the expander inlet, and b) how the expansion of wet vapour can avoid superheated vapour

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. The performance of these systems has previously been studied for operation at both design and off-design conditions (Read et al., 2014a, 2014b). In order to gain more insight into the performance of these systems, they must however be assessed in comparison with equivalent optimised ORC systems using convention turbine expanders. This requires an understanding of how turbine efficiency varies with inlet conditions and required mass flow rate of the working fluid. The aim of this study is to present a comparative analysis of the design and off-design performance of twin-screw WORC and turbine ORC systems.

2. ORC MODEL FOR OPTIMISATION ANALYSIS

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 1, 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 key cycle components are discussed in more detail below.

2.1 Turbine model

Single stage radial inflow reaction turbines are commonly used in ORC applications. These turbines must be sized for specified design point conditions. The flow of working fluid is choked at the throat of the turbine inlet nozzle, and the cross-sectional area at this point must be chosen in order to achieve the required mass flow rate. The operation of the turbine can be characterised by considering the conditions at the throat (denoted by the superscript *) assuming isentropic expansion from turbine inlet conditions (subscript $turb, i$) as described by Wendt and Mines (2013). The pressure, density, enthalpy and velocity of the working fluid at the throat can be calculated using the relationships in equations (1)-(3), where the critical conditions of the working fluid are denoted by the subscript c.

$$p^* = 0.67\, p_{turb,i} \left(\frac{p_{turb,i}}{p_c}\right)^{0.2} \left(\frac{T_c}{T_{turb,i}}\right) \quad (1)$$

$$\rho^*, h^* = f\left(p^*, s_{turb,i}\right) \text{ , where values can be found using an appropriate equation of state} \quad (2)$$

$$u^* = \sqrt{2\left(h_{turb,i} - h^*\right)} \quad (3)$$

For the mass flow rate required at design conditions (denoted by subscript d), the cross-sectional area at the throat of the nozzle can then be calculated using equation (4).

$$\dot{m}_d = (\rho^* A^* u^*)_d \quad (4)$$

The mass flow rate of the working fluid in the cycle will vary at off-design conditions, and two possibilities have therefore been considered for the turbine design:

i. Fixed nozzle geometry, with constant throat area of $(A^*)_d$,
ii. Variable nozzle geometry, allowing the value of A^* to be adjusted.

For the fixed geometry case, an upstream throttle valve is required to reduce the turbine inlet pressure to a value which achieves the required mass flow rate through the fixed design throat area. Using variable geometry, the value of A^* may be varied between zero and a specified maximum value to achieve the required mass flow rate.

To characterise the effect of varying inlet and exhaust conditions on turbine performance, the turbine isentropic efficiency can be related to a velocity ratio, r_u, for the turbine. The velocity ratio is the ratio of the turbine tip speed, u_{tip}, to the spouting velocity, defined as the velocity achieved if the enthalpy change for an isentropic expansion were entirely converted to kinetic energy, as shown in Equation 5 (where $h_{turb,os}$ is the isentropic turbine outlet enthalpy).

$$r_u = u_{tip} / \sqrt{2\left(h_{turb,i} - h_{turb,os}\right)} \quad (5)$$

The correlations proposed by Wendt and Mines (2013) have been used to characterise the change in turbine efficiency as a function of both the change in the nozzle throat area resulting from manipulating the nozzle geometry, and the change in the velocity ratio for the expansion process. In the current study, a constant turbine rotational speed has been used in all cases. As the power output of the turbine is in the region of 100 kW, a representative maximum isentropic efficiency of 75% has been assumed, rather than the 82% proposed for much higher power systems (Wendt and Mines, 2013). The REFPROP database developed by NIST has been used to calculate all thermodynamic properties of the working fluid. The working fluid used in this study is the refrigerant R245fa, which has a critical temperature of 154°C. This is sufficiently high to ensure sub-critical pressure in the evaporator. While using fluids with higher critical temperature may increase the achievable net power output by reducing the pressure difference across the feed pump and expander, the reduced vapour density at condenser pressure would significantly increase the size and cost of cycle components. The cost of the R245fa fluid itself is relatively low, and it is widely used for low temperature ORC applications.

The aim of this study is to understand how changes in operating conditions affect the performance of the conventional ORC system and compare this with the results of previous analysis of a WORC system. Using the method described above, the actual size and operating speed of the turbine are not required to estimate cycle performance, and the detailed design of the turbine has therefore not been considered.

2.2 Heat exchanger models

A discretized approach has been taken to the calculation of the required surface area in the heat exchangers. Once the temperatures of the source, sink and working fluids have been defined, the heat exchangers are split into a number of short sections and the heat transfer and the log-mean temperature difference (LMTD) are calculated. Representative values for the overall heat transfer coefficient in conventional shell and tube heat exchangers with different fluid phases (Roetzel and Spang, 2010) are shown in Table 1, and have been used to calculate the heat transfer surface areas. These are then lumped into two overall heat exchanger areas for 'heat addition' (combined feed-heater, evaporator and, if required, super-heater) and 'heat rejection' (combined de-superheater, condenser and sub-cooler) which can be sized for design-point conditions. The calculation of heat exchanger areas is essential for the analysis of off-design system operation, and while this simple approach is not expected to be highly accurate for design purposes, it can be used to gain some insight into the requirements of the different cycles.

Table 1: Representative values of overall heat transfer coefficient for ORC shell and tube heat exchangers with different states for the heat transfer fluids

State of the heat transfer fluids:		Approximate overall heat transfer coefficient (W/m^{2}K):
Liquid or 2-phase	Liquid	1200
Liquid or 2-phase	Gas	70
Gas	Gas	35

2.3 Integrated cycle model and optimisation analysis

There are two important aspects to applying the component models in an integrated cycle model. Firstly, the mass flow rate identified by consideration of the heat transfer between the source fluid and the working fluid must be matched to the mass flow rate in the expander itself. However, the expander mass flow rate is calculated as a function of the inlet conditions (which may be throttled) and the turbine geometry; an iterative approach is therefore required in order to bring the error between these two calculated mass flow rates below an acceptable value, and identify the required operating conditions for the expander. For turbines with either fixed or variable nozzle geometry, if the required mass flow rate cannot be achieved by the turbine through throttling and/or nozzle area control then the isentropic efficiency is set to zero.

Secondly, although the heat transfer surface area of the heat exchangers can be calculated for the design point optimisation, during off-design operation these values must remain fixed. The varying cycle conditions cause changes in the integrated LMTD and the heat transferred in each heat exchanger. Two separate iterative loops are therefore required to identify the pinch point temperature differences required to achieve the require area of the boiler/evaporator and the de-superheater/condenser/sub-cooler units to within an allowable error. If, for any reason, the required heat exchanger areas cannot be achieved with particular cycle conditions, the expander efficiency is again set to zero.

Applying these iterative subroutines allows the cycle to be completely defined, and the net power output can be calculated. 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, degree of superheat). 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. In this study, the optimisation method was implemented as follows:

i. An initial estimate was made of the optimal system operating conditions.
ii. An initial population of 5000 randomly generated solutions were created, centred on the estimated values.
iii. The combination and mutation algorithm was implemented for 5000 generations, and the best solution identified.
iv. A check was performed by creating a random population of 5000 centred on the best solution, and if a new best solution was identified the procedure was repeated.

3. LOW TEMPERATURE HEAT RECOVERY 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.

The study presented below has investigated the generation of power from this heat source using an ORC with the following characteristics:

- The working fluid is refrigerant R245fa.
- An air cooled condenser is used with 2°C sub-cooling of the working fluid at the exit.
- 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.
- The efficiency of the feed pump has been characterised as a function of volumetric flow and pressure difference rate using data from manufacturers.
- An efficiency of 95% has been assumed for the electrical generator and 90% for pump/fan motors.

For this type of application, where the minimum allowable source temperature is well above the feed pump exit temperature when operating at design conditions, cycle efficiency and hence power output can be improved via recuperation, although the close temperature matching between the superheated vapour and sub-cooled liquid often necessitates a relatively large heat exchanger. In order to simplify the system analysis and the matching of the heat exchanger areas at off-design conditions, this study assumes that no recuperation is used to recover heat from the superheated turbine exit vapour.

Operation of the ORC system has been considered for average climate conditions in Nevada, USA where there are significant geothermal resources of this type. The annual mean temperature is 10.5°C, with monthly variations in the average maximum and minimum temperatures shown in Figure 2. A design-point optimisation has been performed for the annual mean temperature; the fixed parameters for this optimisation are shown in Table 2, and the results are shown in Table 3.

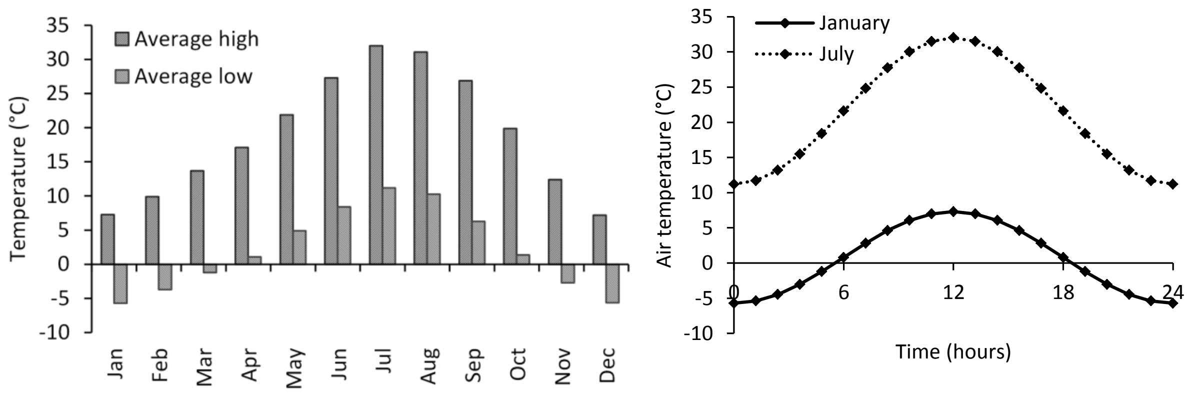

Figure 2: Monthly average maximum and minimum air temperatures in Nevada USA, and examples of assumed daily sinusoidal variation of air temperature

Table 2: Fixed parameters for design point optimisation of ORC system

Working fluid	-	R245fa
Boiler design pinch point	°C	5
Condenser design pinch point	°C	10
$T_{air,design}$	°C	10.5
$T_{source,in}$	°C	120
$T_{source,min}$	°C	70

Table 3: Optimised parameters for ORC with turbine expander operating at the design point conditions stated in Table 2

$p_{evaporator,in}$	bar	8.3
$p_{condenser,in}$	bar	1.7
$\Delta T_{superheat}$ at turbine inlet	°C	0
$\Delta T_{superheat}$ at turbine exit	°C	16
$\dot{m}_{wf}$	kg/s	5.4
$T_{brine,out}$	°C	70
$P_{expander}$ (electrical)	kWe	103
$P_{condenser\ fan}$ (electrical)	kWe	18.1
$P_{feed\ pump}$ (electrical)	kWe	4.6
P_{net} (electrical)	kWe	81.0

3.1 Off-design analysis of ORC with turbine

The off-design performance of the optimised ORC 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 -10 to 40°C. In all cases, the off-design analysis has achieved an error of less than 0.1% between the design-point and off-design values of the heat transfer surface area for the boiler and condenser heat exchangers. Reducing this allowable error was found to have negligible effect on the net power output from the system; with an air temperature of 30°C for example, a maximum error of 1x10^{-6} in the heat exchanger areas was found to change the calculated net power output by less than 0.05%. The resulting system performance for off-design operation is shown in Figures 3 and 4, where the heat recovery efficiency refers to the fraction of available heat that is transferred into the cycle, and the cycle efficiency is the net power output divided by the heat input, as defined in equations (6) and (7).

$$\eta_{heat\,recovery} = \frac{T_{source,i} - T_{source,o}}{T_{source,i} - T_{air}} \quad (6)$$

$$\eta_{cycle} = \frac{P_{net}}{\dot{m}_{source}\left(h_{source,i} - h_{source,o}\right)} \quad (7)$$

3.2 Comparison of results for turbine ORC and twin-screw WORC

In order to assess the effect that the off-design performance has on the operation of the ORC 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 2. 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 5 shows the time-averaged power output from the ORC for each month. The values of net power output calculated for both design-point and time-averaged annual conditions are compared in Figure 6. In Figures 3-6, results are also shown for the WORC using a twin-screw expander, as described by Read et al. (2014b). Finally, Figure 7 shows a comparison between the calculated heat transfer areas for heat addition and reject in both the WORC and ORC systems.

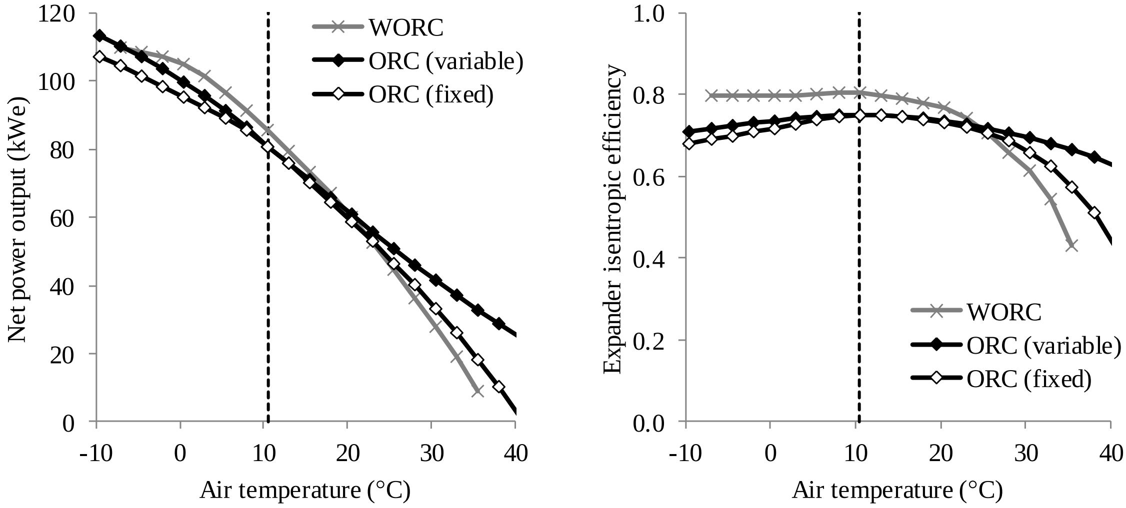

Figure 3: Maximum net power output as a function of air temperature and corresponding expander isentropic efficiency for WORC using twin-screw and ORC using fixed and variable nozzle turbines

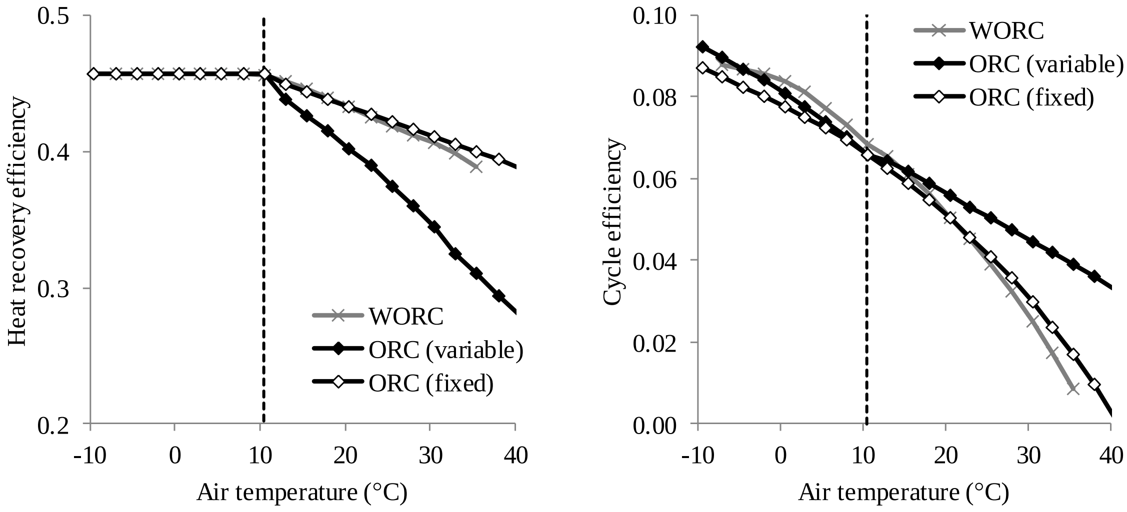

Figure 4: Heat recovery efficiency and cycle efficiency as functions of air temperature for WORC using twin-screw and ORC using turbine with fixed and variable nozzle geometry

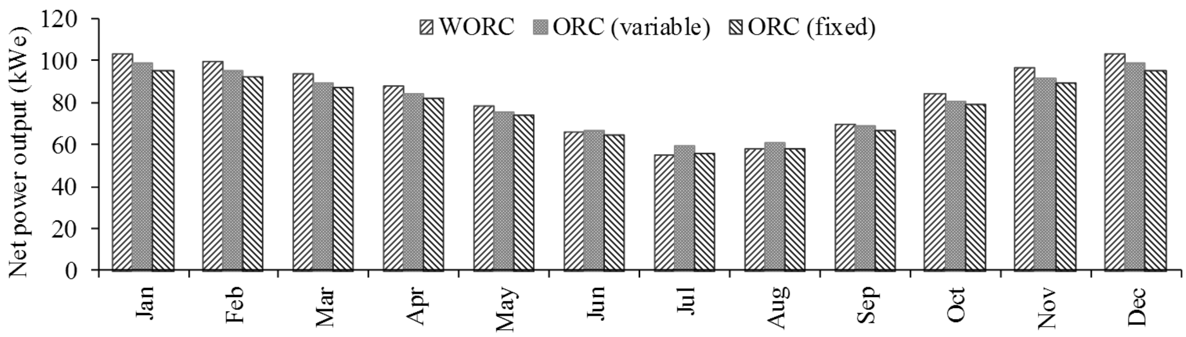

Figure 5: Comparison of monthly time-averaged net power output for the WORC, and ORC using turbine with fixed and variable nozzle geometry

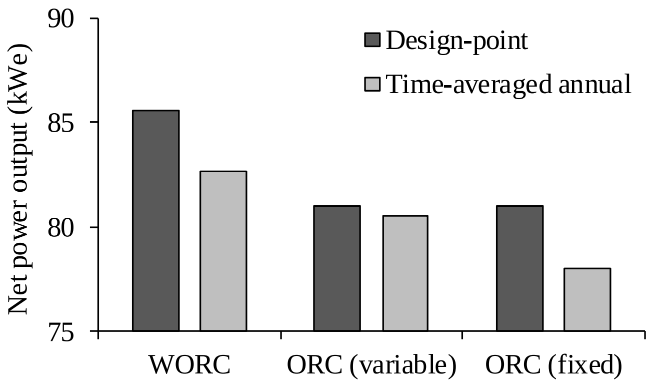

Figure 6: Comparison of net power output from WORC and ORC systems (with fixed and variable turbine nozzle geometries) for design-point and time-averaged annual conditions

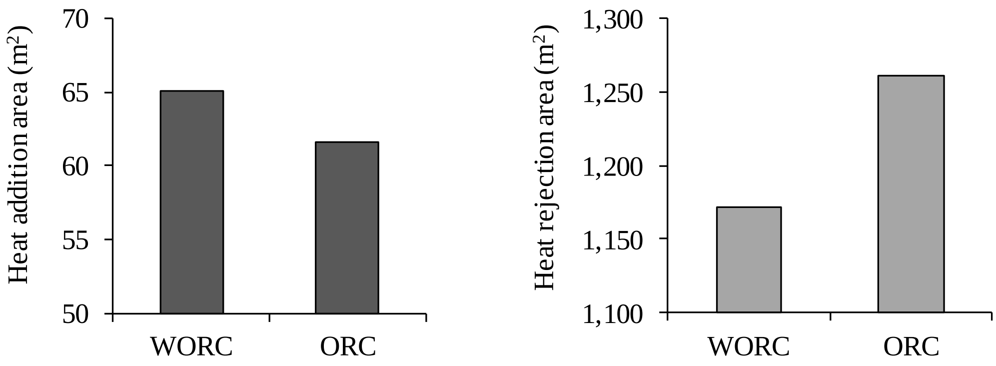

Figure 7: Comparison of predicted heat exchanger areas for WORC and ORC systems sized for design-point conditions

4. DISCUSSION

The results in Figures 3-7 show that similar overall performance is achieved by different ORC systems. The design-point net power output is largest for the WORC, largely due to the slightly higher expander isentropic efficiency predicted for the twin-screw machine, which was also able to maintain higher efficiency at off-design conditions with lower air temperature. The fixed geometry turbine was seen to achieve a lower efficiency than the variable geometry machine at all off-design conditions, but this decrease in efficiency was largely offset by a higher relative recovery of heat from the source fluid at higher air temperatures. The efficiency of the twin-screw machine in the WORC is seen to drop rapidly at higher air temperatures, due to the fixed built-in volume ratio of the machine leading to over-expansion of the working fluid. It may therefore be possible to improve the high temperature performance of the WORC system by allowing optimisation of the expander speed and/or built-in volume ratio in order to better match the volume ratio of the expansion process; the increase in net power output is however expected to be small due to the limited periods of time spent operating at these higher air temperatures. The time-averaged net power output from all cycles is seen to be very similar for all systems, at around 80 kWe, and the use of the average annual temperature to perform the design point calculations is seen to provide a good initial estimate of the real-world system performance.

The results presented above also give some insight into cost implications for the ORC systems. Compared to the optimised WORC, the area required for heat addition in the ORC is reduced by 5%, but the heat rejection area is increased by 8% due to the requirement to cool the superheated vapour at the turbine exit. As the air-cooled condenser is likely to represent a significant proportion of the total system cost due to its large size, this difference could be economically significant. However, more detailed consideration of the design and performance of heat exchangers and the associated heat transfer coefficients would be required to confirm this. The possible economic benefits of using twin-screw machines in WORC systems has already been discussed in detail (Leibowitz et al., 2006), and the results here suggest that performance can match or exceed conventional ORCs for relatively low power applications. It is however worth noting that the isentropic efficiency of turbines increases with power up to a maximum of around 83% for large-scale geothermal applications (Wendt and Mines, 2013). While the efficiency of twin-screw machines also generally improves with size and power output (due to the relative reduction in leakage flows), there is a practical limit of around 0.5 metres for the maximum rotor diameter. Applied to a WORC, isentropic efficiencies of around 84% and a net power output of around 590 kWe are predicted (Read et al., 2014b) and represent an upper limit to WORC operation using twin-screw expanders.

5. CONCLUSIONS

In this paper, the optimisation and part-load simulation of low temperature heat recovery systems has been demonstrated. The case study considered above suggests that similar overall performance can be achieved by ORC systems using both twin-screw and turbine expanders. The results indicate that for the application considered (where the heat source conditions remain constant) there is little benefit, in terms of average power output, in using a variable geometry turbine over the fixed type. While the efficiency of the screw expander is seen to decrease more rapidly than the turbine at higher air temperatures, the WORC is predicted to achieve comparable design-point and time-averaged performance and offers the potential for a low cost and low complexity system. The model described can be used for a wide range of applications, and allows comparative studies of the technical and economic performance of low temperature heat recovery systems and their components.

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