TECHNO-ECONOMIC ANALYSIS OF PARTIAL EVAPORATION ORGANIC RANKINE CYCLE SYSTEMS FOR GEOTHERMAL APPLICATIONS

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ABSTRACT

The organic Rankine cycle technology is an efficient and cost-effective way to utilise low-grade heat for power generation. Especially when the investment costs for the extraction of the heat source are high, it is of crucial importance to increase the power production in order to maximize the revenues and thus ensure a reasonable pay-back time. Besides the classical sub-critical cycles, various cycle architectures have been investigated to improve the performance of organic Rankine cycle systems. Among these, the partial evaporation organic Rankine cycle architecture is considered particularly promising because it provides a better match with the heat source temperature profile during heat exchange, resulting in a greater utilization of the available energy of the heat source. Previous studies have reported an increase in exergy efficiency of the system by up to 60 % when considering partial evaporation instead of complete evaporation of the organic fluid, with the greatest advantages occurring for low temperature heat sources. The use of partial evaporation also results in a more voluminous and costly organic Rankine cycle system. However, there are no previous analyses that address the economic feasibility of partial evaporation organic Rankine cycle systems. This paper presents a techno-economic analysis of partial evaporation cycles, with particular focus on low-grade geothermal heat sources. Partial evaporation and sub-critical organic Rankine cycle systems were optimized for different heat source conditions and their techno-economic performances were compared. The results suggest that partial evaporation cycle systems are able to provide an increase in net power output between 20 % and 60 % compared with sub-critical cycles for the considered heat source temperatures, but also require higher specific investment costs (up to 75 %) due to the increase in heat exchange area and number of stages of the expander. Nevertheless, due to the small contribution of the power system to the cost of the whole plant and due to the high equivalent full load hours, the results of this paper indicate that adopting partial evaporation can reduce the levelized cost of electricity up to 12 % for geothermal plants.

1 INTRODUCTION

Thermodynamic cycles with partial or no evaporation of the working fluid for power production from low-grade heat sources have been proposed by various authors. The primary reason for adopting partial evaporation cycles is that they resemble the trilateral cycle, which is the thermodynamic cycle that theoretically achieves the highest efficiency when extracting power from heat sources of finite heat capacity (Smith, 1993). The trilateral (also known as triangular) cycle is characterized by the expansion phase starting at saturated liquid conditions. In fact, removing the constant temperature vaporization process allows for a better match between heat source and working fluid temperature profiles during the heat exchange, thus reducing irreversibilities and increasing the amount of exchanged heat and of produced power. However, the applicability of trilateral cycles in power production systems is strictly dependent on techno-economic feasibility considerations. The main concerns are related to the degradation of performance and the reduction of lifetime of the expander due to two-phase expansion.

The increase in net power production achieved by trilateral cycles compared with classical Rankine cycles has been estimated by various authors. Smith (1993) evaluated the power produced by the recovery of a hot water stream at different initial temperatures (between 100 °C and 200 °C) using both water and organic fluids. An increase in power output up to 80 % was found for trilateral cycles for a two-phase turbine isentropic efficiency above 75 %, with the highest increase occurring for the lowest temperatures of the heat source and with water as working fluid. Fischer (2011) found an increase in net power output for trilateral cycles using water between 14 % and 29 % compared to sub-critical ORCs for a temperature of the heat source ranging from 350 °C to 150 °C. Based on considerations about the costs and expander design, both authors suggest to adopt organic fluids instead of water as working fluid for triangular cycles. Lai and Fischer (2012) further investigated the performance of trilateral cycles using organic fluids such as aromates, siloxanes and alkanes. Despite ensuring the highest exergy efficiency, high critical temperature working fluids such as water and siloxanes resulted in very high volumetric flow rates across the expander, making its design very challenging and costly. Moreover, such fluids have condensation pressures well below atmospheric pressure, leading to expensive condensers. On the contrary, alkanes and light hydrocarbons showed slightly lower exergy efficiencies but also lower volume flow rates and condensation pressures close to the atmospheric one.

Some of the drawbacks of trilateral cycle systems can be reduced by introducing a partial evaporation of the working fluid, resulting in partial evaporation organic Rankine cycle (PEORC) systems. First, using part of the enthalpy of vaporization of the working fluid decreases the required mass flow rate of working fluid for the same heat transfer rate with the heat source and thus the power consumption of the pump drops. Moreover, heat transfer coefficients are enhanced by the vaporization process and, hence, heat transfer surfaces are reduced. Furthermore, the increase in volumetric flow rate in the expansion is lower, thus facilitating the expander design, increasing its efficiency and reducing its cost. The thermodynamic performance of partial evaporation cycles was investigated by Lecompte *et al.* (2013) and Lecompte *et al.* (2015) with fixed efficiency of the expander. In both papers, the results suggest that the trilateral cycle outperforms the partial evaporation cycle for heat source temperatures below 250 °C when the pump isentropic efficiency is above 60 %. However, since the results were obtained for a fixed isentropic efficiency of the expander, the effect of the expander specific work and volumetric flow rates was not taken into account. Moreover, the analysis was limited to a thermodynamic optimization of the cycle and the effect on the components' design and cost was not considered.

An estimation of the system cost is proposed by Lecompte *et al.* (2014) for a low temperature heat source (water at 100 °C). Here the trilateral cycle was compared with the trans-critical and sub-critical ORC, with the expander efficiency fixed at 60 %. The adoption of trilateral cycle systems was found to be associated with an increase of 19 % in specific investment cost for the trilateral cycle power system. A similar analysis was performed by Yari *et al.* (2015) for a heat source consisting of hot water at 120 °C. Unlike Lecompte *et al.* (2014), the authors found no relevant increase in specific investment cost for the optimized trilateral cycle over the sub-critical ORC. A sensitivity analysis on the expander isentropic efficiency was also performed, showing a decrease in net power output by 40 % and an increase in specific investment cost by 25 % for a decrease in efficiency from 85 % to 65 %, demonstrating the great impact of the expander efficiency on the system performance and cost.

Based on the results from previous studies, there is clear evidence of the potential of trilateral and partial evaporation cycles in increasing the power production from low-temperature heat sources. However, a thorough evaluation of the performance and costs of such systems that takes into account the influence of the design conditions on the expander efficiency has not been carried out so far and it is therefore illustrated in this paper, as it is expected to have a strong influence on the optimum cycle architecture. Since the increase in net power output achieved by partial evaporation cycles is associated with an increase in specific investment cost of the power system compared to sub-critical organic Rankine cycles, the actual economic feasibility of PEORC systems results from a trade-off between a high initial investment cost and high revenues from power production. The levelized cost of electricity ($LCOE$) parameter accounts for these two competing objectives, since it depends both on the system cost and on the total energy production during the plant lifetime. Previous studies on partial evaporation cycles did not address the trade-off between revenues and investment cost, so this aspect is addressed in the present work, where the $LCOE$ is estimated and it is included among the optimization objectives.

Partial evaporation cycles are expected to be more beneficial for power plants with high equivalent full load hours of operation and high overall investment costs, like geothermal plants, where the power production unit cost is small compared to the extraction cost of the heat source and it is likely to have a smaller effect on the $LCOE$. The objective of this paper is therefore to evaluate the techno-economic feasibility of PEORC systems for geothermal applications by optimizing the thermodynamic cycle and by estimating relevant economic parameters as the specific investment cost (SIC) of the system and the levelized cost of electricity ($LCOE$). As research on the design and optimization of two-phase turbines is still at an early stage, there are only very few commercial solutions available and there is a lack of efficiency and cost correlations tailored for two-phase turbines. For these reasons, assumptions on their efficiency were made in order to compare partial evaporation cycles with sub-critical cycles. As heat source, a low-grade geothermal brine with a 100 kg/s mass flow rate and a supply temperature of 100 °C (cases 1 and 2) and 150 °C (cases 3 and 4) is considered. In order to evaluate how the reinjection temperature constrains the maximum power extraction, cases 1 and 3 limit the reinjection temperature of the heat source to a low minimum of 30 °C, whereas cases 2 and 4 consider a higher limit of 70 °C.

2 METHODS

The techno-economic analysis was performed using an optimization routine implemented in MATLAB® that was originally developed for the optimal design of sub-critical ORC systems for waste heat recovery applications by Pili *et al.* (2019). The purpose of the optimization routine is to calculate and minimize (or maximize) the chosen optimization objective based on the design of the thermodynamic cycle and the estimation of the investment cost for each component, as a function of a set of decision variables. The optimization uses a genetic algorithm and was carried out for three optimization objectives: the net power output of the system (to be maximized) and two economic parameters characterizing the investment (to be minimized). When the optimization is completed, the optimum set of decision variables and the corresponding optimum cycle are found.

2.1 Thermodynamic model

The key components that realize the thermodynamic cycle are the working fluid pump, the primary heat exchanger, where heat is transferred between the heat source and the working fluid, the expander and the condenser. A recuperator is also included when the working fluid leaves the expander in superheated vapour conditions. In addition, auxiliary components such as the geothermal brine circulation pump and the condenser fans are considered in the model. The thermodynamic properties of the fluids are calculated only at the inlet and outlet of each component for each set of optimization variables. The most relevant parameter characterizing the performance of the system is the net power output, P_{net}, which is directly related to the exergy efficiency for power production, η_{ex}, through the inlet exergy flux of the heat source ($\dot{E}_{in,hs}$):

$$P_{net} = P_e - P_{p,wf} - P_{p,hs} - P_{f,cs} \tag{2.1}$$

$$\eta_{ex} = \frac{P_{net}}{\dot{E}_{in,hs}} = \frac{P_{net}}{\dot{m}_{hs} \cdot \left[\left(h_{in,hs} - h_{ref}\right) - T_0\left(s_{in,hs} - s_{ref}\right)\right]}, \tag{2.2}$$

where P_e is the power produced by the expander, and $P_{p,wf}$, $P_{p,hs}$ and $P_{f,cs}$ are the power consumption of the working fluid pump, the heat source pump, and the condenser fans, respectively.

The system is assumed to operate in steady state conditions. Heat losses and pressure drops in heat exchangers are neglected. Losses occurring in the other components, as pumps, fans, electric motors, inverters and electric generator, are modelled through fixed isentropic or conversion efficiencies, except for the expander. In fact, the expander performance is strongly dependent on the design conditions and therefore its isentropic efficiency is evaluated through numerical correlations. For the working fluid pump an isentropic efficiency of 70 % is considered (Astolfi *et al.*, 2014a), whereas for the geothermal brine circulation pump a 75 % isentropic efficiency and a 300 kPa pressure drop are assumed. Air cooler fans are considered to cover a 200 Pa pressure drop with a 65 % isentropic efficiency (AspenTech Inc., 2020). Electric motors driving pumps and fans have a 95 % efficiency, while inverters and the electric turbine generator have a 98 % efficiency (Astolfi *et al.*, 2014a).

The decision variables of the optimization consist of the minimum set that allows to fully determine the thermodynamic cycle. The following optimization variables are adopted: the evaporation pressure P_{eva}, the condensation temperature T_{cond}, the equivalent expander inlet quality $x_{in,e}$, the primary heat exchanger pinch point $\Delta T_{pp,PHE}$, the condenser pinch point $\Delta T_{pp,cond}$ and the recuperator effectiveness ε_{rec}. Details regarding the range of variation of the optimization variables are given in Table 1. The equivalent expander inlet quality is defined in equation (2.3) and it is used to normalize the expander inlet enthalpy with the latent heat of vaporization, given by the difference between the enthalpy of the saturated liquid $h_{sat,L}(P_{in,e})$ and the saturated vapour $h_{sat,V}(P_{in,e})$ at the expander inlet pressure. The equivalent expander inlet quality substitutes the degree of superheating, which is commonly used as optimization variable for sub-critical organic Rankine cycle (SCORC) systems. Together with the evaporation pressure, the equivalent quality univocally defines the expander inlet conditions and allows a smooth transition between partial evaporation and superheated cycles during the optimization.

$$x_{in,e} = \frac{h_{in,e} - h_{sat,L}(P_{in,e})}{h_{sat,V}(P_{in,e}) - h_{sat,L}(P_{in,e})} \tag{2.3}$$

The choice of the working fluid plays a fundamental role in the performance of the thermodynamic cycle (Astolfi *et al.*, 2017). Based on the conclusions from previous works that investigated trilateral and partial evaporation cycles, such as Yari *et al.* (2015) and Lecompte *et al.* (2015), 8 fluids were chosen for the analysis. Among them, 4 fluids were selected because they perform well in SCORC systems for the considered range of heat source temperatures (R227ea, R143a, R114 and R218) and 4 fluids were chosen because they were considered to be suitable for low-temperature PEORC systems (cyclopentane, toluene, pentane and R245fa).

The inlet conditions of the heat source are given as input to the optimization routine. The cooling medium in the condenser is ambient air with a temperature of 25 °C and its mass flow rate is a result of the optimization. In addition, the geothermal brine is modelled as pure water and the properties of the working fluids are retrieved from the REFPROP database (Lemmon *et al.*, 2018).

Table 1: Range of variation of the optimization variables

	P_{eva} [kPa]	$x_{in,e}$ [-] (PEORC; SCORC)	T_{cond} [°C]	$\Delta T_{pp,PHE}$ [°C]	$\Delta T_{pp,cond}$ [°C]	ε_{rec} [-]
Lower bound	$P_{sat}(T_{out,hs,lim})$	$(0; 1)$	$T_{in,cs}$	10	10	0
Upper bound	$0.8 \cdot P_{cr}$	$(1; 3)$	$T_{in,hs}$	30	30	0.8

2.2 Choice and preliminary sizing of the components

The sizing of the main components of the power plant is limited to the calculation of the parameters needed to estimate the cost of the components through available cost correlations. Both the primary heat exchanger (PHE) and the recuperator are shell and tube heat exchangers. In the PHE the geothermal brine flows on the tube side and the working fluid on the shell side, while in the recuperator the organic fluid flows on the tube side as high-pressure liquid and on the shell side as low-pressure vapour. The condenser is an air cooler with radial fins and forced circulation powered by axial ducted fans. To estimate the cost of the heat exchangers, the calculation of the heat transfer surface area was carried out with the logarithmic mean temperature difference ($LMTD$) approach, following the procedure presented by Pili *et al.* (2019). The heat exchangers were not discretized and the global heat transfer coefficient

U includes the contribution of the convective heat transfer for both fluids through the heat transfer coefficients for the external and internal flowing fluid h_{ext} and h_{int}, and the fouling resistances through the fouling factors f_{ext} and f_{int}, as shown by equation (2.4). The reference surface (A_{ref}) for the calculation of the global heat transfer rate is the bare external surface of the tubes and each contribution to U is thus referred to this surface through "area ratios". The convective heat transfer coefficients were estimated by designing the heat exchangers in Aspen® Exchanger Design and Rating (AspenTech Inc., 2020) for typical design conditions, while the fouling factors and the area ratios suggested by Astolfi (2014) were adopted. These parameters are summarized in Table 2.

$$U = \left[\frac{1}{h_{ext} \frac{A_{ext}}{A_{ref}}} + \frac{f_{ext}}{\frac{A_{ext}}{A_{ref}}} + \frac{1}{h_{int} \frac{A_{int}}{A_{ref}}} + \frac{f_{int}}{\frac{A_{int}}{A_{ref}}} \right]^{-1} \tag{2.4}$$

Table 2: Heat exchange coefficients, area ratios and fouling factors for heat exchanger sizing

	PHE				**Recuperator**		**Condenser**		
	Organic fluid				Organic fluid		Organic fluid		
Fluid conditions	Liquid	Sat. vapour	Super-heated vapour	Brine	Liquid	Super-heated vapour	Super-heated vapour	Sat. vapour	Air
Heat exchange coefficient [W/m²K]	1000	2400	700	3000	600	200	400	1600	50
Area ratio [-]	1			0.78	1	0.87	1	1	23.5
Fouling factor [m²K/W]	0.00018				0.00018		0.00018		0.00035

Centrifugal pumps are utilized for the geothermal brine and the working fluid. The expander is a multistage axial turbine, whose number of stages is calculated based both on efficiency and technical considerations. A first constraint is set on the maximum specific enthalpy drop across each turbine stage, which is set to 65 kJ/kg as indicated by Astolfi *et al.* (2014a). In addition, the efficiency for a single-stage, a two-stage and a three-stage turbine is calculated based on the efficiency correlation proposed by Macchi and Astolfi (2017) as a function of the size parameter and the volume ratio VR, defined in equation (2.5). The latter is the ratio between volumetric flow rates at the outlet ($\dot{V}_{out,is}$) and inlet ($\dot{V}_{in}$) of the expander and it takes into account the effect of increasing velocities and blade height on the expander performance. When the volume ratio is so high that adding an extra stage would lead to an increase in isentropic efficiency above 3%, an additional stage is added to the turbine. For two-phase turbines used in PEORC systems, the same correlation is used in order to take into account the effect of volume ratio and size parameter on the efficiency and number of stages, which is assumed to be the same as that of single-phase turbines. Due to the uncertainties on two-phase turbines design and efficiency, a reduction factor f_e on the efficiency with respect to the value given by the correlation for single-phase turbines is introduced to allow for a sensitivity analysis.

$$VR = \frac{\dot{V}_{out,is}}{\dot{V}_{in}} \tag{2.5}$$

2.3 Estimation of the components cost

The investment cost of all the major components of the system is estimated. The cost correlations adopted for heat exchangers, pumps and turbine are the same as those selected by Pili *et al.* (2019). A cost correlation based on Turton (2012) is used for axial ducted fans. The indicators chosen to evaluate the economic feasibility of the systems are the power system specific investment cost (SIC) and the levelized cost of electricity ($LCOE$), respectively defined in equations (2.6) and (2.7). To characterize the investment over time, the $LCOE$ was chosen instead of the pay-back time because it is not dependent on the local price of electricity and it is therefore a more general indicator. For the same reason, the financial incentive for the produced energy was not considered when calculating the $LCOE$.

$$SIC = \frac{C_{p,wf} + C_{PHE} + C_e + C_{rec} + C_{cond} + C_{p,hs} + C_{f,cs}}{P_{net}} \tag{2.6}$$

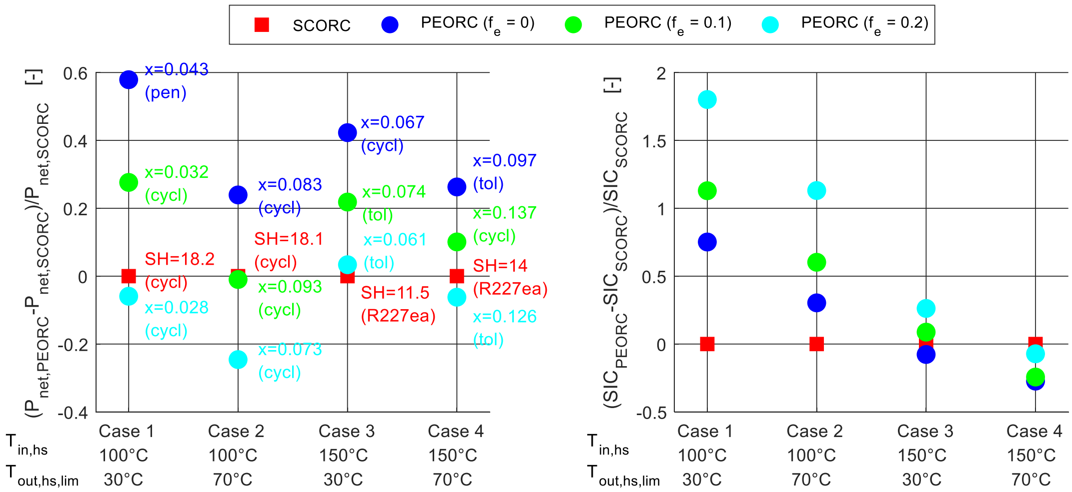
$$LCOE = \frac{C_0 + \sum_{t=1}^{N} \frac{C_t}{(1+i)^t}}{\sum_{t=1}^{N} \frac{P_{net} \cdot h_{op}}{(1+i)^t}} \tag{2.7}$$

To evaluate the $LCOE$, the following values (Walraven *et al.*, 2015) were assigned to the parameters: a number of operating years N equal to 30, an interest rate i of 4 %, a number of equivalent full load operating hours per year h_{op} equal to 8000 and yearly operating and maintenance costs, C_t, equal to 2 % of the total investment cost C_0. The latter includes both the investment cost for the power block and for the extraction of the heat source, which is considered to be dependent on the geothermal brine temperature only. Based on Astolfi *et al.* (2014b), fixed extraction costs of the heat source C_{hs} equal to $6 \cdot 10^6$ € and $4 \cdot 10^6$ € were assumed for a geothermal brine temperature of 150 °C and 100 °C respectively. Due to the uncertainties in predicting the extraction costs, a sensitivity analysis on this parameter was carried out.

3 RESULTS AND DISCUSSION

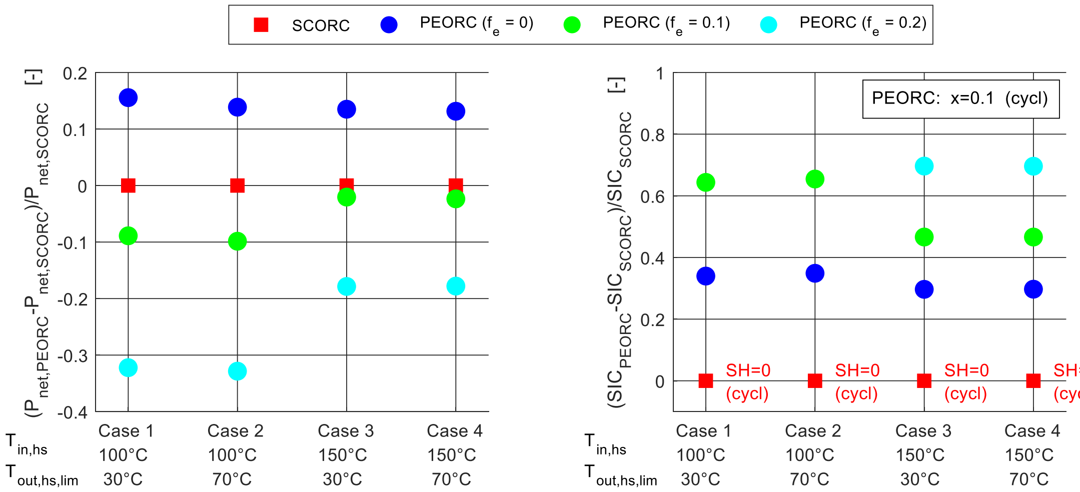
The results when using the net power output P_{net} of the system as optimization objective are shown in Figure 1a. The abscissa shows the different case studies while the ordinate shows the relative gain in net power output for the PEORC compared with the SCORC. Only results for the best performing working fluids are reported, but differences in P_{net} smaller than 3 % were found among cyclopentane (‘cycl’), toluene (‘tol’) and pentane (‘pen’) for the PEORC and among cyclopentane, pentane, R227ea and R114 for the SCORC. The effect of adopting turbine efficiency correlations results in expander inlet qualities ($x = x_{in,e}$ in the two-phase region) higher than 0 in contrast to what Lecompte *et al.* (2015) found for a fixed efficiency. The results suggest that the optimum value of $x_{in,e}$ increases with the heat source temperature, and the constraint on the minimum reinjection temperature ($T_{out,hs,lim}$) is found to have a great influence on the maximum P_{net} only for the PEORC. Regarding the efficiency reduction caused by the two-phase expansion, the results indicate that the PEORC cannot outperform the SCORC if the efficiency reduction factor f_e is higher than 0.2 (except for case 3). Figure 1b depicts the specific investment cost (SIC) of the optimal solutions. The SIC is found always to be higher for the PEORC with a 100 °C heat source, while it is slightly lower for $T_{in,hs}$ = 150 °C, due to the larger increase in P_{net} with respect to the increase in component costs.

Figure 2a and 2b show the relative gain in P_{net} and SIC for the PEORC compared to SCORC using SIC as optimization objective. As expected, in this case the SIC of the PEORC is always higher than the SCORC due to the higher cost of the heat exchangers. This means that, for the PEORC, the optimal x would be 1 for all cases. To allow for a comparison, the results shown in Figure 2a and 2b are obtained with an expander inlet quality x constrained to 0.1. The optimal SCORC for minimum SIC is found for a degree of superheating equal to 0 K, whereas the optimal SCORC for maximum P_{net} has a degree of superheating between 10 K and 20 K (Figure 1a). This is consistent with what Astolfi *et al.* (2014a) and Astolfi *et al.* (2014b) found for thermodynamic and economic optimization of ORC systems for geothermal applications.



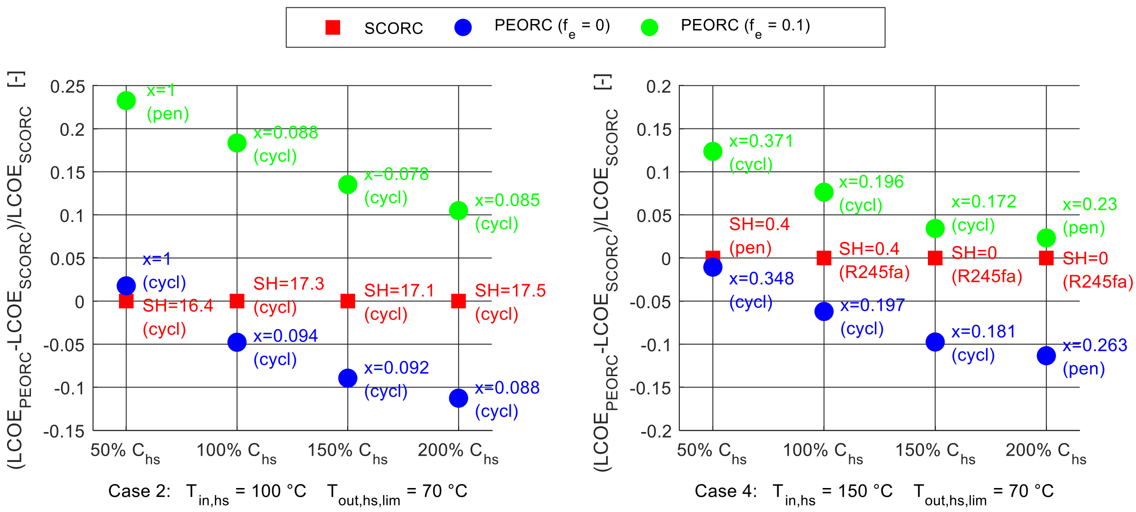
(a) **(b)**

Figure 1: Relative gain in P_{net} (a) and SIC (b) with P_{net} as optimization objective



(a) **(b)**

Figure 2: Relative gain in P_{net} (a) and SIC (b) with SIC as optimization objective



(a) **(b)**

Figure 3: Relative gain in $LCOE$ for case 2 (a) and case 4 (b) with $LCOE$ as optimization objective

The absolute figures for the P_{net} and SIC obtained for the reference cycle (SCORC) are listed in Table 3 for both P_{net} and SIC as optimization goals, and they are found to be very close to the specific costs found by Astolfi *et al.* (2014b). In general, the results are affected by the power consumption of the air cooler fans, which is a relatively large share of the produced power, especially for cases 1 and 2.

A comparison for $LCOE$ as optimization objective is presented in Figure 3a for case 2 and in Figure 3b for case 4 in terms of relative gain in the $LCOE$. The absolute values of the $LCOE$ for the reference case (SCORC) are reported in Table 4. Only the two cases where the reinjection temperature is limited to 70 ºC are shown, since they are more representative of actual geothermal applications. The optimization of $LCOE$ was not performed for a reduction in expander efficiency f_e = 0.2, since in this case adopting the PEORC does not improve neither the P_{net} nor the SIC of the system (Figures 1a and 1b) and thus the SCORC is always the optimum. The results suggest that if no expander efficiency degradation occurs (f_e = 0), the $LCOE$ of PEORC systems is significantly lower (in the range 2 % to 12 %) than that of the SCORC. The sensitivity analysis on the extraction cost of the heat source C_{hs} indicates that the optimal cycle gets closer to the one found when optimizing the net power output (Figure 1) when C_{hs} is increased, due to the necessity of maximizing the revenues from power production when the cost of the whole plant increases. On the contrary, when the cost of the heat source extraction is lower, the optimum cycle gets closer to the sub-critical ORC, that allows minimizing the SIC of the power system. The $LCOE$ values obtained for both cases 2 and 4 for the SCORC system are overall aligned with what was previously found by Walraven *et al.* (2015), namely, a $LCOE$ between 0.15 €/kWh and 0.20 €/kWh for a 100 °C geothermal brine and between 0.05 €/kWh and 0.08 €/kWh for a 150 °C brine.

Table 3: Results for P_{net} (row 1) and SIC (row 2) as optimization objectives for the SCORC system

	P_{net} [kW]				SIC [€/kW]			
Objective	**Case 1**	**Case 2**	**Case 3**	**Case 4**	**Case 1**	**Case 2**	**Case 3**	**Case 4**
P_{net}	304	305	2225	2210	4682	4784	2739	2931
SIC	270	274	1518	1529	4313	4278	1308	1306

Table 4: Results for $LCOE$ as optimization objective for the SCORC system

	Case 2				**Case 4**			
	50 % C_{hs}	100 % C_{hs}	150 % C_{hs}	200 % C_{hs}	50 % C_{hs}	100 % C_{hs}	150 % C_{hs}	200 % C_{hs}
$LCOE$ [€/kWh]	0.108	0.172	0.236	0.300	0.028	0.042	0.055	0.069

4 CONCLUSIONS

In this paper a techno-economic analysis of partial evaporation organic Rankine cycle systems was performed for low-temperature geothermal sources, taking into account the heat source extraction costs and the dependence of the expander efficiency on the parameters of the thermodynamic cycle. The results suggest that adopting the partial evaporation provides a decrease of the levelized cost of electricity up to 12 % when the performance of the expander is not significantly affected by the two-phase expansion. However, the two-phase expander remains the most critical component of the system and research efforts addressing its design would enable the effective adoption of partial evaporation systems, thus allowing for a more efficient utilisation of geothermal heat sources.

NOMENCLATURE

SIC	Specific investment cost	(€/kW)	x	Quality	(-)
$LCOE$	Levelized cost of electricity	(€/kWh)	f	Reduction factor *or* Fouling factor	(-) *or* (m^2K/W)

P	Power	(kW)	A	Heat exchange area	(m^2)
η	Efficiency	(-)	VR	Volume ratio	(-)
$\dot{E}$	Exergy flux	(kW)	$\dot{V}$	Volumetric flow rate	(m^3/s)
$\dot{m}$	Mass flow rate	(kg/s)	C	Cost	(€)
h	Specific enthalpy *or* Heat exchange coefficient	(kJ/kg) *or* (W/m^2K)	t	Time	(years)
T	Temperature	(°C)	i	Interest rate	(-)
s	Specific entropy	(kJ/kgK)	N	Number of operating years	(yr)
P	Absolute pressure	(kPa)	h_{op}	Operating hours per year	(hr/yr)
SH	Degree of superheating	(°C)			

Subscripts

net	net	pp	pinch point
ex	exergy	sat	saturation
e	expander	cr	critical
wf	working fluid	L	liquid
hs	heat source	V	vapour
f	fan	out	outlet
cs	cold source	lim	limit
in, 0	inlet/initial	is	isentropic
ref	reference state	rec	recuperator
eva	evaporation	int	internal
$cond$	condensation	ext	external

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