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Proceedings of the 7th International Seminar on ORC Power Systems

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ON AIR-COOLED CONDENSERS FOR ORC SYSTEMS OPERATING WITH ZEOTROPIC MIXTURES

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ABSTRACT

The use of mixtures as working fluids of ORC systems is being intensively investigated because of the better temperature profile matching achievable in the heat exchangers, resulting in lower thermodynamic irreversibilities and increased efficiency. The gains are expected to be higher for low-temperature air-cooled power plants, where the ratio between the auxiliary power consumption associated to the cooling of the working fluid and the net power output of the ORC plant is higher. For instance, a temperature glide in the condenser may enable a reduction in the fan consumption at the cost of an increased heat transfer area, and possibly a decrease in the minimum temperature of the thermodynamic cycle. This solution is expected to be attractive for geothermal applications: since the drilling of the geothermal well is by far the dominant cost, the additional investment for the condenser can be more easily compensated by the increased revenues related to the greater electrical power output.

This study focuses on the modelling and sizing of an air-cooled condenser for geothermal ORC power plants operating with working fluid binary mixtures. A detailed fin and tube air cooled condenser model is developed and integrated with an in-house tool for the simulation of ORC systems. Working fluid thermodynamic properties are computed with the PCP-SAFT equation of state (EoS). The tool is used to investigate the effect of the condenser design assumptions on the geothermal plant maximum power output for an optimal working fluid mixture, whose composition is determined by optimizing the PCP-SAFT parameters. The outcome is a pseudo-fluid mixture that represents the ideal working fluid for the given thermal source.

The results indicate that the adoption of mixtures allows the air-cooler consumption and generally the minimum cycle temperature to be decreased, leading to an increased plant efficiency. In addition, design guidelines for the condenser are derived, based on the tradeoff between component size and plant efficiency. Finally, the optimization results show that if the onset of the mixture condensation occurs in the recuperator, it might be possible to reduce the fan consumption and size of the condenser simultaneously, albeit at the expense of an increased complexity of the regenerator design.

1 INTRODUCTION

Mixtures are a promising option for ORC systems exploiting low temperature thermal sources. In those systems, the impact on the plant power output of the condenser auxiliary power consumption as well as that of the higher condensing temperatures is high and can be reduced by exploiting the temperature glide of mixtures. This improves the matching between the temperature profiles of the working fluid and the power plant thermal sink, thus reducing the heat transfer irreversibilities. The benefit consists of net power gains ranging from 4% (Trapp and Colonna, 2013) to over 10% (Heberle et al, 2011) when compared to ORC systems operating with pure fluids and the same pinch point temperature difference in the condenser. At the same time, the reduced mean temperature difference achievable with the mixture glide leads to a larger condenser heat transfer area, which may further increase due to the degradation of the mixture heat transfer coefficient with respect to pure fluids, as observed in different experimental campaigns (Radermacher and Hwang, 2005). For this reason, the thermodynamic advantages related to the adoption of mixtures as working fluid of ORC systems might not necessarily guarantee an economic benefit.

More recent works evaluated the economic benefit of zeotropic mixtures in geothermal applications by means of thermo-economic optimizations. For instance, Andreasen (Andreasen et al, 2021) compared the performance of mixtures of propane, i-butane, i-pentane and R1234yf to that of their pure components. They found that a R1234yf/i-butane mixture can achieve 5% higher net present values than R1234yf, which was the best pure fluid candidate among the selected ones. A similar result was obtained by Heberle (Heberle and Brüggemann, 2015) who reported a decrease in electricity generation cost of 4-10% compared to the most cost-efficient pure component. However, given the limited set of fluids considered in both works, it is not possible to generalize the results of these studies.

The choice of the mixture components is challenging due to the considerable number of combinations which can be achieved even from a limited set of pure components. This problem can be addressed by using Continuous Molecular Targeting (CoMT) methods, where the EoS parameters that describe the mixture components are relaxed to continuous variables and optimized together with the ORC process parameters. In this way, the computationally demanding mixed integer optimization problem representing the selection of the mixture components is simplified to a continuous one with a larger number of optimization variables. The outcome of the simplified problem is an optimal pseudo-mixture (or pure pseudo-fluid) enabling the ORC system to achieve the maximum possible performance, and whose properties must be subsequently approximated with existing fluids. An alternative method that avoids treating the EoS parameters as optimization variables is that proposed by Schilling (Schilling et al, 2021): by using the Group Contribution PCP-SAFT (Sauer et al, 2014), the authors directly optimized the molecular structure of the working fluids in the mixture, thus taking into account an extremely large number of mixture candidates, albeit with components having a structure limited to certain molecular groups. When considering liquid wastewater at 150 ºC as heat source, Schilling et al. identified a combination of propene/propionaldehyde (94%/6%) as the optimal mixture and reported a limited (3%) decrease in Specific Investment Cost (SIC) compared to the optimal pure fluid, i.e. propene. At the same time, the SIC obtained for the optimal pseudo-mixture and that of the optimal pure pseudo-fluid resulted to be equal. It may be then argued that the better thermo-economic performance observed for the propene/propionaldehyde mixture is only due to the additional degrees of freedom in ORC system design introduced by the possibility of tuning the mixture composition.

The aim of this work is to evaluate the increase in performance achievable by using zeotropic mixtures in geothermal applications with respect to that obtained in the case of pure working fluids as a function of the pinch point temperature difference assumed for the air-cooled condenser design. To this purpose, a detailed fin and tube air-cooled condenser model is developed and integrated with an in-house tool for the simulation of ORC systems. To the authors' knowledge, even though air cooling is by far the most common solution in geothermal binary plants (Di Pippo, 2016), the works documented so far in the literature on ORC mixtures for such applications consider water-cooled condensers. These not only are more compact than the air-cooled ones, but the corresponding power consumption can be neglected as well.

2 METHODOLOGY

2.1 Power Plant

This study considers an air-cooled recuperated ORC system for geothermal applications. Two cases are investigated: in the first one, the geothermal brine temperature at the outlet of the power plant is unconstrained; in the second case, it is limited to 100 ºC, exemplifying a combined heat and power

(CHP) application with a serial plant layout, or equivalently a conventional all-electric power plant configuration exploiting a high salinity brine. Note that there are other possible CHP plant configurations, but the serial one is the most suitable in case of a limitation on the brine leaving temperature.

The thermodynamic cycle calculation procedure is similar to that adopted in (Krempus et al, 2021). In particular:

1. Pinch-point temperature difference is fixed in each heat exchanger.
2. Turbomachinery isentropic efficiencies are fixed.
3. All the pressure drops except the one on the air side of the condenser are prescribed.

Table 1 indicates the general cycle specifications: the thermal source of the plant is geothermal brine at 150 °C. This is modelled as pure water, as it is assumed to be coming out as liquid from the geothermal well. Notice that neither the brine composition (provided that the specific heat capacity remains constant) nor the mass flow rate affect the estimation of the plant thermodynamic performance. Their effect is limited to the sizing of the cycle components and the power plant capacity. The pinch-point temperature difference and pressure drop in the heat exchangers, as well as the turbomachinery thermodynamic efficiencies are fixed in accordance with (Krempus et al, 2021). The only relevant difference with respect to the work documented in (Krempus et al, 2021) is related to the condenser pinch-point temperature difference, which is varied in the present study from a maximum of 10 °C to a minimum theoretical value of 0.5 °C. Moreover, in contrast to (Krempus et al, 2021), cycle maximum temperature and condensation pressure were not constrained: the first one because of the relatively low temperature of the geothermal source, which is unlikely to cause any thermal decomposition of the working fluid, while the second one was left unconstrained to allow for a larger design space exploration. Finally, the condenser is sized as a fin and tube heat exchanger, by using the procedure described in section 2.3.

The power output is chosen as objective function of the optimal design problem. This is reasonable for geothermal applications because, as already mentioned, the exploration and drilling of the geothermal wells are much more expensive than the ORC power block (Astolfi et al, 2013). Optimal cycle parameters and mixture composition are found using the genetic algorithm provided by the optimization suite Pymoo (Blank and Deb, 2020).

Table 1: Thermodynamic cycle specifications

Parameter	Value	Parameter	Value	Parameter	Value	Parameter	Value
$T_{\mathrm{HRVG}_{\mathrm{H_{in}}}}$	150 °C	$\Delta T_{\mathrm{pp_{Reg}}}$	10 °C	$\Delta T_{\mathrm{pp_{HRVG}}}$	10 °C	η_{turbine}	90%
$P_{\mathrm{HRVG}_{\mathrm{H_{in}}}}$	40 bar	$\Delta P_{\mathrm{Reg_H}}$	0.025 bar	$\Delta T_{\mathrm{pp_{Cond}}}$	0.5-10 °C	η_{fan}	65%
$\Delta P_{\mathrm{HRVG_H}}$	0.05 bar	$\Delta P_{\mathrm{Reg_C}}$	$0.025p_{\mathrm{max}}$	$\Delta P_{\mathrm{Cond_H}}$	0.025 bar	T_{amb}	30 °C
$\Delta P_{\mathrm{HRVG_C}}$	$0.025P_{\mathrm{max}}$	η_{pump}	80%				

2.2 Thermodynamic property modeling

The PCP-SAFT EoS (Gross and Sadowski, 2001) as implemented in (Colonna et al, 2019) is adopted to predict the thermodynamic properties of the working fluids, both in the case of pure compounds and mixtures. Only non-polar molecules were considered in this study. These can be modelled using three EoS parameters, namely segment number, diameter and energy parameter. Moreover, because of the lack of experimental data, all mixture binary interaction parameters are set to zero.

The ideal gas specific heat and molecular weight of the pure pseudo-fluids are calculated directly from the PCP-SAFT parameters using a QSPR method (Stavrou, 2017). Additional constraints are applied to the PCP-SAFT parameters to avoid unfeasible pseudo-fluids, see (Stavrou, 2017) for more details. The molar fractions and PCP-SAFT parameters of the working fluid components are optimized together with the cycle parameters. The bounds of the overall optimization space are reported in Table 2.

Table 2: Bounds of the optimization variables.

Cycle	Min	Max	Fluid	Min	Max
T_{max}	90 ºC	140 ºC	M	2 (-)	6.5 (-)
$P_{max,R}$	0.5	0.8	σ	2 (Å)	4.5 (Å)
T_{min}	$30 + \Delta T_{\mathrm{pp_{Cond}}}$	60 ºC	ε_K	150 (K)	300 (K)
			c	0.01	0.5

2.3 Air Cooled Condenser model

The air-cooled condenser is of the fin and tube heat exchanger type, see the example shown in Figure 1. The air flows through a series of finned tube rows, while the condensing fluid flows inside the tubes, following a serpentine path that depends on the tube arrangement. This study exclusively considers arrangements with one row for each pass, as in the example reported in Figure 2.

Table 3 reports the main geometrical features of the tube banks. The tubes feature classical circular fins, whose characteristic dimensions are selected based on (Serth, 2014). The effect of the number of passes was also considered by repeating the optimization study for different values of this variable.

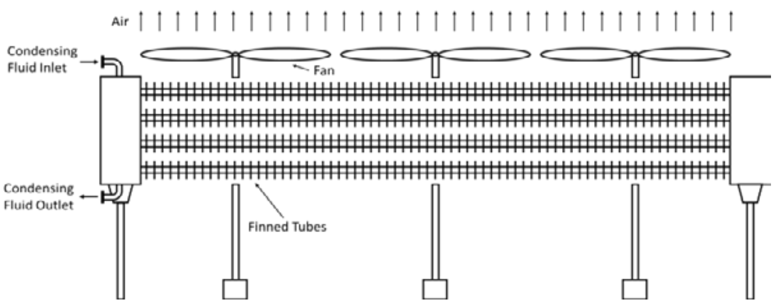

Figure 1: Configuration of an air-cooled fin and tube condenser

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The model handles both superheated and saturated working fluid states. It takes the working fluid mass flow rate, inlet and outlet enthalpies, as well as the heat exchanger design pinch-point as inputs. The outputs are the heat exchange area, frontal area, design air velocity, the air side pressure drop and the fan power.

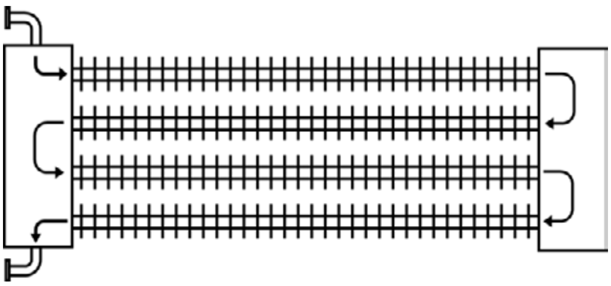

Figure 2: Configuration with one row per pass

Table 3: Tube bank characteristic dimensions

Parameter	Value
D_{Ext}	25.5 mm
D_{Fin}	56 mm
$Tube\ Pitch$	60 mm
$Tube\ Length$	9 m
N_{Passes}	3-6

Table 4 provides a summary of the principal correlations used by the model. Note that the adopted version of the PCP-SAFT is unable to directly compute the transport properties of the pseudo-fluids from the EoS parameters, unlike the molecular weight and ideal gas specific heat. For this reason, the de-superheating and condensing heat transfer coefficients are set to 200 at 1500 W/m^2K, respectively. Based on a preliminary condenser design exploration, these values were found to be characteristic of medium-sized condensers. Significant deviations may be expected for very large or very small condensers due to the very low or high flow velocities involved in these cases. However, such a simplification does not introduce significant inaccuracies in the model for two reasons: the first is that the air heat transfer coefficient was usually found to be 2-3 times lower than that of the condensing fluid, even when considering the effect of the fins. The second is that the working fluid is only slightly superheated at the entrance of the condenser, because of the presence of the regenerator upstream. The rejected sensible heat is then rather small if compared to the total heat load of the heat exchanger. In most of the analyzed test cases, the error on the heat exchanger area was found to be less than 5%. The exceptions are usually in the case of condensers operating with mixtures showing a high glide (on the order of 20ºC or more) and, consequently, exhibiting a considerable degradation of the heat transfer coefficient.

The validation of the model was performed by sizing the condenser for representative working fluids and operating conditions using the model, and then comparing the estimated performance with the predictions generated by the commercial software Aspen Plus. Table 5 summarizes the main process parameters of one of the validation cases.

Table 4: Main correlations of the model.

Parameter	Source
h_{air}	VDI-GVC, 2010
F	VDI-GVC, 2010
ΔP_{air}	$Serth,$ 2014
h_{cond}	Deng et al, 2019
h_{desup}	Dittus-Boelter Corr.

Table 5: Validation case with Aspen Plus.

Parameter	Model	Software
$h_{air}\ (W/m^2K)$	744	730
$\Delta P_{air}\ (Pa)$	91	95
$\dot{Q}\ (MW)$	3.20	3.18
$A\ (m^2)$	1016	1018

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3 RESULTS

3.1 Scenario 1: no constrain on the temperature of the leaving brine

Figure 3 exemplifies the T-s diagram obtained for an optimal pseudo-binary mixture. The glide in the condenser is perfectly linear and matches the air temperature profile. On the contrary, the evaporator operates at a high reduced pressure, with a limited temperature glide and the fluid exiting as saturated vapor, as commonly observed in ORC systems.

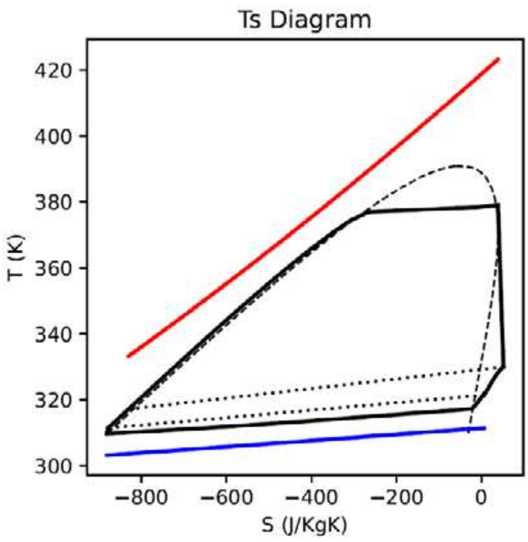
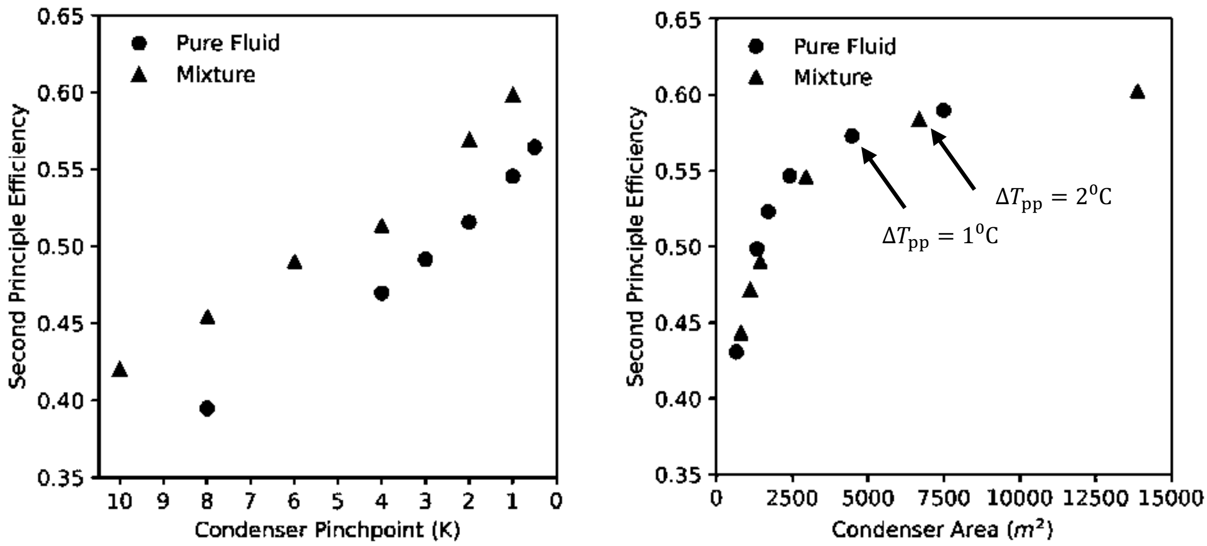


Figure 3: T-s diagram of the optimal pseudo-binary mixture for a condenser with 4 passes and a pinch-point of 8 ºC

Given the large variation of the heat source temperature, the optimal mixture glide in the evaporator is arguably significantly higher than the optimal one in the condenser. Even if the EoS parameters are relaxed to continuous values, it is essentially impossible to find a mixture whose difference in boiling point of the components allows for a glide varying from about 10 degrees during condensation to several tens of degrees in the evaporator Furthermore, large glides during evaporation would correspond to fluids with broader saturation domes and operating at lower reduced evaporation pressures, which may be below the limits imposed on this variable. Finally, it should be noted that such a high glide would also result in a substantial reduction of the mixture heat transfer coefficient, as this phenomenon is driven by the difference in boiling temperature of the mixture components.

Figure 4 compares the second law efficiency of the ORC system operating with the optimal pseudo-mixtures and that found for the optimal pseudo-fluids, as a function of the condenser pinch temperature difference and heat transfer area. For a given pinch-point temperature difference, the optimal pseudo-mixtures show a higher performance, confirming their superior thermodynamic potential. However, if this comparison is performed with respect to the condenser heat exchange area, it can be seen from Figure 4b that pure fluids and mixtures have comparable second-law efficiencies. Similar conclusions might be deduced from the results reported by Schilling et al. for water cooled power plants (Schilling et al, 2021).



a) As a function of the pinch-point **b) As a function of the condenser area**

Figure 4: Comparison of the second law efficiency of the ORC systems operating with the optimal pseudo-mixtures and pseudo-fluids for a condenser with 3 passes and an unconstrained temperature of the leaving brine

At the same time, it should be noted that the pure fluid ORC systems achieving the highest efficiencies exhibit very low condenser pinch-point temperature differences (1 and 0.5 ºC), which may not be achievable in a real component. For instance, effects such as thermal conduction along the tubes, the difference between static and total temperature, or flow maldistribution in the tube bundle may reduce the actual heat transfer rate under low temperature differences. This can limit the efficiency improvement attainable by increasing the condenser area in the case of pure fluids. On the contrary, with mixtures, a higher pinch-point temperature difference is attained also in the case of large heat transfer areas, thus resulting in a feasible and more operable condenser. For this reason, in geothermal applications, where the driving cost is represented by the exploration and drilling of the geothermal wells, it may be profitable to decrease the logarithmic temperature difference both by exploiting working fluid mixtures temperature glide and adopting relatively low pinch points values.

Concerning the number of passes, they do not seem to significantly affect the results. This can be better understood by observing Figure 5: the glides of the resulting optimal mixtures are relatively mild. Increasing the number of passes does not increase the effective logarithmic temperature difference and, thus, does not yield significant reductions in the required heat exchange area. It can also be seen that the glide of the optimal mixtures follows almost a linear trend as a function of the prescribed condenser minimum temperature difference. Given all the complex interaction between the components, fluid and cycle parameters, understanding the reason of this trend requires additional investigations.

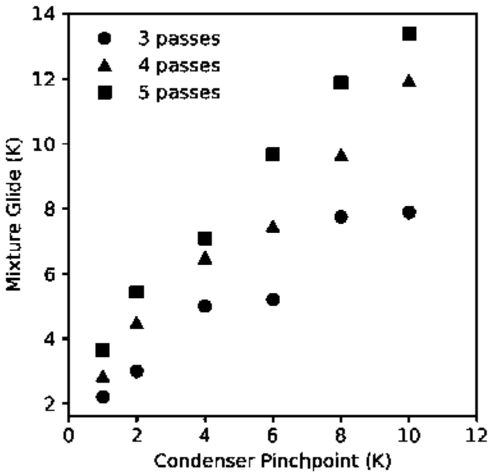
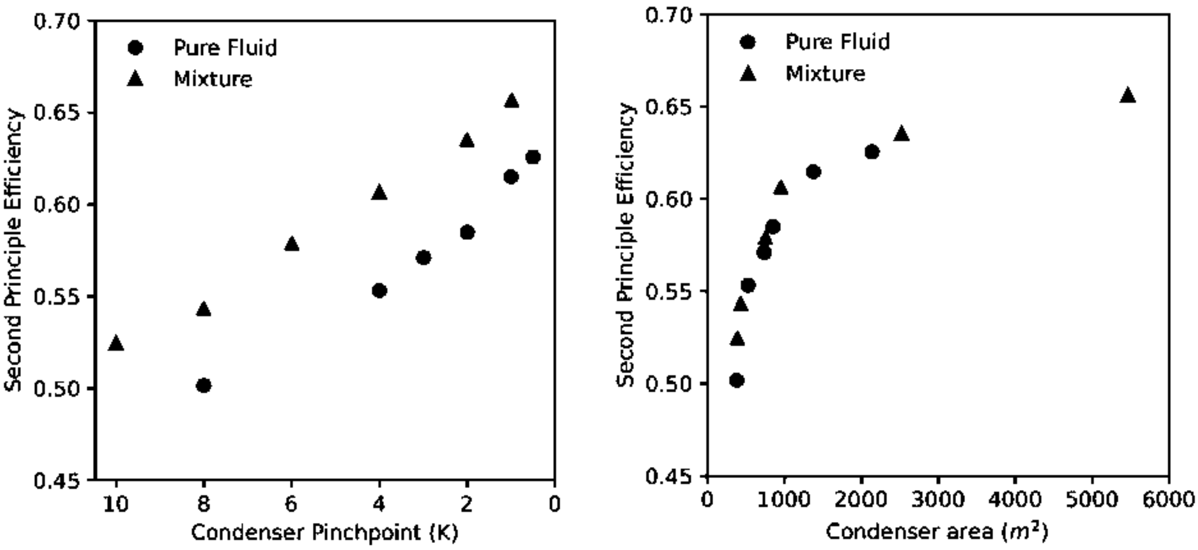


Figure 5: Condensation glide of the pseudo-mixtures as a function of the condenser pinch-point, for an unconstrained temperature of the leaving brine

3.2 Scenario 2: Constrained temperature of the leaving brine

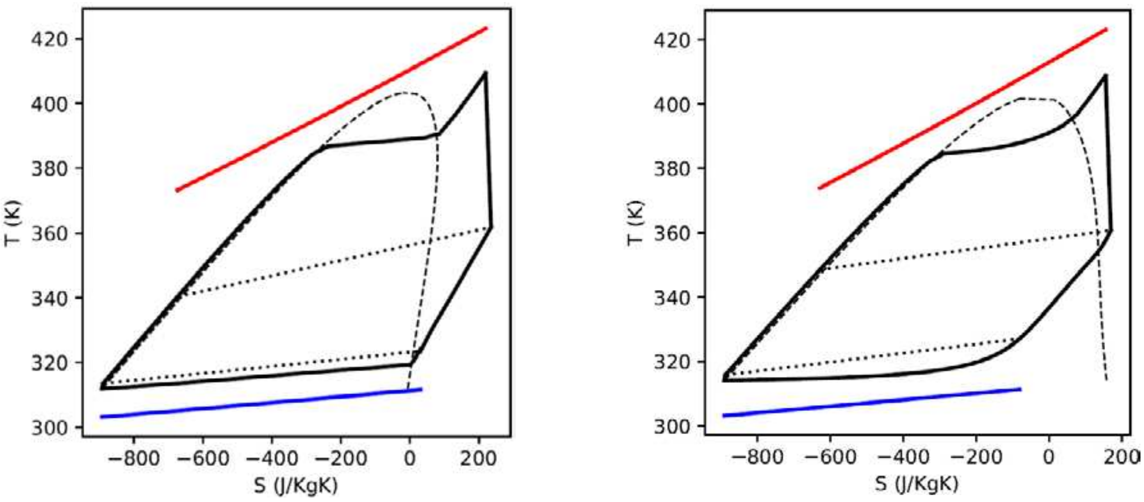
In general, the considerations made in the previous section still apply when the brine outlet temperature is constrained to a minimum value of 100 ºC, as it can be seen in Figure 6.



a) As a function of the pinch-point **b) As a function of the condenser area**

Figure 6: Comparison of second law efficiency of pseudo-pure fluids and pseudo-mixtures for a minimum brine leaving temperature of 100 ºC

As the brine temperature is now constrained to a high value, regeneration plays a much more important role in the thermodynamic cycle. For this reason, the optimization problem may have two different optima, as shown in Figure 7: one features a pseudo-mixture with a linear glide, while the other another pseudo-mixture that partially condenses in the regenerator, so as to maximize the recuperated thermal energy. This is an unexplored option in ORC technology and could prove useful to reduce the total heat exchange area of the power plant by taking advantage of the higher heat transfer coefficient of the liquid working fluid. However, such a gain might be partially offset by the degradation of the mixture heat transfer coefficient, which in this case is expected to be quite significant.



a) Condenser with a linear glide **b) Regenerator with partial condensation**

Figure 7: Possible thermodynamic cycles with mixtures for a minimum brine leaving temperature of 100 ºC

4 CONCLUSIONS

The use of mixtures as working fluids for air-cooled geothermal ORC power plants was investigated by means of an in-house tool for steady-state thermodynamic cycle computations and optimization, integrating a preliminary design model of the condenser. The predictive capabilities of the PCP-SAFT EoS were exploited to identify the optimal working fluids for the application by optimizing the parameters of the EoS. Notably, the identified pseudo-pure compounds or pseudo-mixtures set the maximum possible performance of the ORC system for the prescribed design specifications. The main findings of the study are: 351

- For a prescribed pinch-point temperature difference in the condenser, the ORC systems operating with binary mixtures exhibit better performance than those with pure fluids, confirming the expected thermodynamic advantages.
- The identified pseudo-mixtures and pure fluids exhibit an equivalent performance as a function of the condenser heat transfer area. However, for a prescribed ORC system power output, mixtures allow for condensers with higher pinch-point temperature differences. As temperature differences lower than 1 ºC are arguably not attainable, mixtures are then expected to allow for higher actual efficiencies.
- Because of the relatively low glide of the optimal mixtures, increasing the number of passes does not yield any significant improvement of the effective mean logarithmic temperature difference and, consequently, does not reduce the surface area of the condenser. Thus, a large number of passes is generally not recommended, if not to respect other design constraints.
- Given that generally the cost of exploration and realization of the wells is by far the dominant one, in geothermal applications it could be worthwhile to further increase the plant power output by using mixtures and, at the same time, adopting relatively low pinch-point temperature differences in the condenser.
- When the brine leaving temperature is constrained, such as in CHP applications, allowing a partial condensation of the mixture in the regenerator could be an interesting option to reduce the size of the condenser and improve the efficiency of the power plant.

NOMENCLATURE

m	Segment number (PCP-SAFT)	**Subscript**	
σ	Segment diameter (PCP-SAFT)	pp	Pinch-point
ε_K	Segment energy parameter (PCP-SAFT)	HRVG	Heat recovery vapor generator
c	Molar concentration	Reg	Regenerator
F	Logarithmic temperature difference correction factor	R	Reduced quantity with respect to the critical one
η	Isentropic Efficiency	Ext	External
h	Heat Transfer Coefficient	H	Hot Side
D	Diameter	C	Cold Side
N_{Passes}	Number of tube passes		

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This manuscript reflects only the authors' views and opinions, neither the European Union nor the European Commission can be considered responsible for them.

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This book contains the compilation of works contributed to the *7th International Seminar on Organic Rankine Cycle Power Systems* (ORC 2023), held in Seville between the 4th and 6th of September 2023. The event was hosted by Universidad de Sevilla on behalf of the Knowledge Centre on Organic Rankine Cycle Technology (KCORC), incorporated in The Netherlands.

The ORC conference, organized biennially, stems as the only conference that is specific to ORC technology, therefore gathering a diverse community whose affiliation spans across all the interested stakeholders, not only in this particular technology but also and in a broader context, in the energy transition. Original equipment manufacturers, professional associations, end-users, investors, policy makers, academics, scientists feel at home at ORC 2023.

The almost 100 proceedings in this book cover a wide variety of topics, from fundamentals to system integration through component design, accounting for thermodynamic performance as well as component design. In addition to this, and as a new track in 2023, works on heat pump technology were also accepted in order to raise awareness of the strong ties between both technologies, specifically in energy storage applications.

This book provides an excellent overview of the current maturity of power systems based on Organic Rankine Cycle technology for applications as diverse as geothermal and waste heat recovery in industry or downstream of other prime movers (e.g., marine applications). It is also an excellent source of information to understand the current challenges faced by the technology, stemming from a very competitive market and increasingly stringent environmental regulations.

The organizers of ORC 2023 hope that the reader finds this work as exciting as the attendees to the conference and, maybe, make the decision to join the 8th edition to the conference in 2025.

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