Thermodynamic Optimization of an Organic Rankine Cycle for Power Generation from a Low Temperature Geothermal Heat Source

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Abstract

The increasing concern on environment problems has led to the development of renewable energy sources, being the geothermal energy one of the most promising ones in terms of power generation. Due to the low heat source temperatures this energy provides, the use of Organic Rankine Cycles is necessary to guarantee a good performance of the system. In this paper, the optimization of an Organic Rankine Cycle has been carried out to determine the most suitable working fluid. Different cycle layouts and configurations for 39 different working fluids were simulated by means of a Gradient Based Optimization Algorithm implemented in MATLAB and linked to REFPROP property library. The heat source was hot water from a geothermal reservoir with an inlet temperature of 120ºC and an outlet temperature limit of 75ºC. For each working fluid, an optimal configuration was obtained, based on the optimization of the second law efficiency. In addition, a sensitivity analysis for the polytropic efficiencies of the pump and turbine was carried out. Results show that those working fluids with a critical temperature close to the maximum temperature of the cycle give the highest plant efficiencies (being propylene and R1234yf the best ones). Using a recuperator increases the plant efficiency in all cases with exception of wet working fluids. The cycles experiencing the highest sensitivity on the pump performance are those using working fluids with low critical temperatures. Increasing the number of stages of the turbine increases the overall plant efficiency for all working fluids, but some fluids are more sensitive to the turbine efficiency than others.

Keywords: Process modelling, process simulation, working fluid selection, parametric optimization, second law efficiency.

1   Introduction

Historically, the preferred methods for power generation have been related to Brayton or Rankine power cycles, fueled by natural gas or other fossil fuels (Macchi and Astolfi, 2017). However, during the last years, the increasing concern of the greenhouse effect and climate change has led to an increase of renewable energy, such as wind and solar power. In addition to these listed renewable energies, there is an energy source that shows a promising future due to the advantages it provides when compared to other renewable energies. This developing energy is geothermal energy, and its advantages are related to its availability (Macchi and Astolfi, 2017): it does not depend on the ambient conditions, it is stable, and it offers the possibility of renewable energy base load operation. One of the challenges of geothermal energy is that it does not provide very high temperatures, and this fact has made researches and engineers to focus their studies on how to obtain high thermodynamic efficiencies at low- and medium-temperature heat sources. One of the solutions for this problem are Organic Rankine Cycles (ORCs), which present a simple structure with a high reliability, an easy maintenance and, most importantly, a high potential to produce power from low source temperatures when compared to other power generation cycles (Saleh *et al*, 2007).

The design of the ORC requires a thorough analysis, since there are many parameters affecting its performance, and any change of these parameters will have a major impact on the efficiency of the cycle. With this aim, many different studies have been carried out. (Saleh *et al*, 2007) analyze the performance of 31 different working fluids, finding that those working fluids with the highest boiling temperatures are giving the highest efficiencies, being cyclopentane the best one for the given conditions and assumptions. (Roy *et al*, 2011), by means of a parametric ORC optimization, found that R123 is the best working fluid due to its high efficiency and high power production. (Hung *et al*, 2010) studied the efficiency of an ORC where benzene, ammonia, R11, R12, R1234a and R113, are used as the working fluids, differencing between wet, dry and isentropic working fluids and concluding that isentropic

fluids are preferred for ORC systems, since they are giving the best efficiencies. (Shu *et al*, 2014) studied the thermal efficiency and exergy losses in an ORC, using pure hydrocarbons and mixtures; zeotropic mixtures seem to present a better thermodynamic performance than pure working fluids. These conclusions were also found by (Haberle *et al*, 2012), who use a mixture of i-butane and i-pentane for studying the second law efficiency.

Even though many studies have been carried out, none of them shows the same results regarding which is the best working fluid for an ORC with low heat source temperature, and all of them agree on the fact that the thermodynamic boundaries of the cycle, the working fluid selection, the constraints, and choice of the degrees of freedom are key parameters for analyzing and designing the cycle. In this work, a thermodynamic analysis has been carried out, focused on the optimization of the cycle (being the second law efficiency the objective function), analyzing more than 35 different pure working fluids from different classes, and investigating the pump and expander efficiencies impact on the cycle performance.

2 Organic Rankine Cycle

2.1 Cycle design

The ORC is the main technology used to transform low temperature thermal energy into electrical power. The heat is transferred to the working fluid in an evaporator (or primary heat exchanger), which evaporates and may superheat. Then, the working fluid reaches the expander, normally a turbine, in which it is expanded and experiences an enthalpy drop between the inlet and outlet, transforming the energy from the fluid into mechanical work. This mechanical work is transformed into electrical energy in a generator. After the expander, the working fluid reaches the condenser, where it rejects the surplus heat and changes its phase again. Once all the vapour has been condensed, the remaining liquid is pumped back to the evaporator. This is the simplest

ORC, which can be seen in Figure 1, together with a typical T-s diagram for the same cycle layout.

This simple configuration can be modified to obtain higher efficiencies (by adding more pressure levels, different types of regeneration, reheating, etc). Multiple pressure levels are normally used when the available heat to be absorbed from the heat source is high; otherwise, the expensive and complex cycle layout would not be justified. The reheating unit is installed in the cycle when the amount of liquid that is formed during the expansion process is so high that it could damage the turbine (Macchi and Astolfi, 2017; Agromayor, 2017). For the case study of this work, the heat source does not provide a high temperature; therefore, multiple pressure levels and reheating configuration were not considered.

Regeneration is a common practice in ORC layouts. It is beneficial from a thermodynamic point of view when the fluid at the outlet of the expander is superheated vapour and there exists a limitation for the lower temperature of the heat source (Agromayor, 2017). Figure 2 shows the regenerative cycle layout. The regeneration allows to preheat the liquid that enters the primary heat exchanger, and, at the same time, to cool the superheated vapour coming out of the turbine. This results into lower cooling loads and smaller condensers. Both cycle layouts (the simple and the regenerated one) allow to work in different cycle configurations: saturated, superheated and transcritical. For this project, all the configurations were considered for each working fluid, to determine which one gives the best results in terms of efficiency.

2.2 Working fluid selection

The fluid selection for an ORC is the key parameter to design the cycle, meaning that it is the most important degree of freedom during its analysis. The complexity of the process is not just a consequence of the large amount of working fluids that can be found in the market, but also of the infinite number of mixtures that
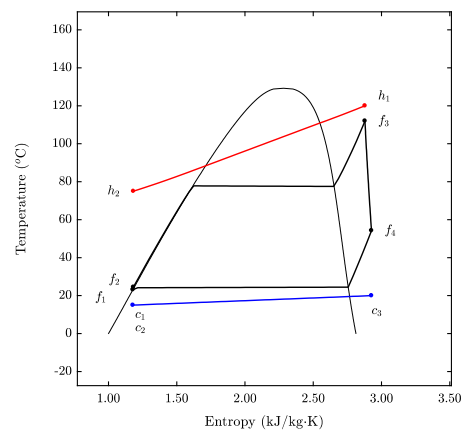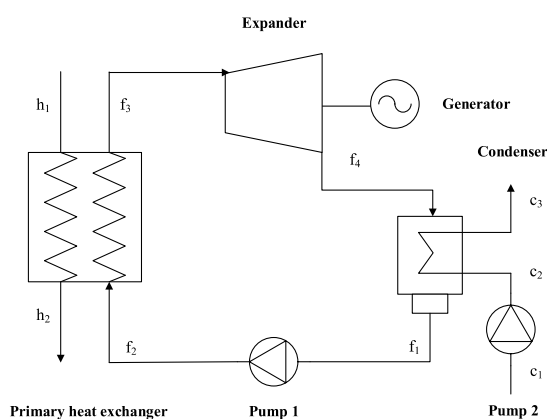

Figure 1. **Simple ORC layout and typical T-s diagram for a random pure working fluid**
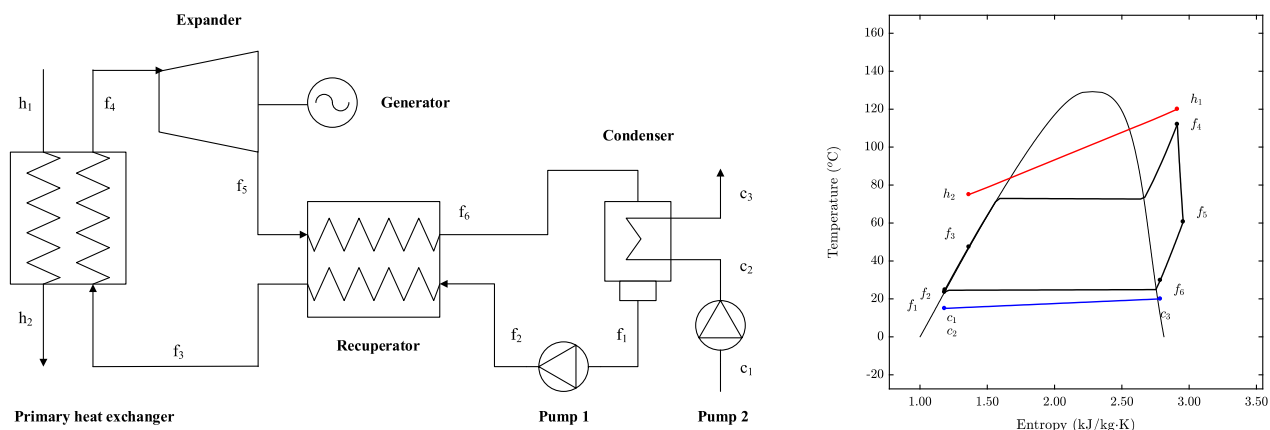
Figure 2. **Recuperated ORC layout and typical T-s diagram for a random pure working fluid**

can be obtained from these pure substances. Taking a decision when choosing the working fluid for the ORC has a major impact on the cost, the components and cycle performance and on the safety requirements (Macchi and Astolfi, 2017).

(Feng *et al,* 2015) propose the use of mixtures in the ORC, due to the non-isothermal change of phase in the evaporator, which allows to obtain better gliding temperature matches between the heat source and the working fluid. However, the analysis reflects that mixtures are not always giving better cycle performances than pure working fluids. Due to the high complexity that this problem has, the study of different mixtures was not included in this work.

The uncertainty fluid properties can influence the reliability of the optimized cycles, and, for this reason J. Frutiger et al. propose in (Frutiger *et al*, 2016) an analysis of the impact of uncertainty of the fluid properties in the behavior of the ORC, including the Molecular Weight (MW), critical temperature, critical pressure and acentric factor. This gives an idea of how important the working fluid properties are when choosing the working fluid. There are different factors or properties to consider when taking this decision, and we can find, among others (Macchi and Astolfi, 2017):

- The molecular complexity. It influences the T-s diagram shape (differencing between dry, isentropic and wet fluids) which is essential for the performance of the turbine, since it determines the amount of liquid that may be formed during the expansion. (Zhai *et* al, 2014) demonstrate that complex molecules that include double bonds or cycle bonds are much more efficient than molecules with only single bonds, with a lower molecular complexity
- The molecular mass. This parameter has a major influence on the design of the turbine. When the molecular weight increases, the enthalpy drop in the turbine is lower, and this is translated into a need of less stages for the expansion, lower peripheral

velocities and mechanical stresses, although the heat transfer coefficient diminishes and heat exchangers with larger heat transfer areas are needed (higher costs).
- The critical temperature. It is recommended to use working fluids with critical temperatures as closest as possible to the highest temperature of the cycle, since they guarantee an efficient heat exchange process.

The impact of the working fluid on the environment is important. Fluids with a high Ozone Depletion Potential (ODP) are undesirable, while organic fluorinated fluids (HFCs) are to be avoided because of their high Global Warming Potential (GWP). From (Zhai *et al*, 2014) it was found that the GWP is not only important for the environment concern, but also for the efficiency of the cycle (results show that those working fluids with a GWP lower than 1500 can absorb more energy from the heat source, improving the cycle efficiency).

The most common working fluid for conventional Rankine cycles is H_2O. However, when the heat source temperature is low, H_2O shows poor performances. As H_2O is a wet working fluid, it condensates during the expansion, and this may damage the turbine. To solve this problem, H_2O requires a too high degree of superheating and a low enthalpy drop across the expander, meaning that the power produced is low (low plant efficiencies). It was found to be of interest to investigate how H_2O behaves in our case of study even if the performance was expected to be poor.

3 Methodology

The Rankine cycle model was implemented in MATLAB and optimized with a Gradient Based Algorithm (Sequential Quadratic Programming, SQP). The advantage of Gradient Based Algorithms against the popular direct search algorithms, such as the Genetic Algorithm, is that the former has a faster convergence rate. The code can simulate and optimize both the simple

and the recuperated Rankine cycles and it computes all the thermodynamic states of the cycle, using the REFPROP library.

3.1 Fluid screening selection

Most of the manufacturers opt to work with organic fluids that have already been tried, to reduce the uncertainty and simplify the fluid selection process. However, this does not necessarily mean that they are the best choice. In fact, (Colonna *et al,* 2015) insist on the need of carrying out research on new substances. In this work, some of the most used working fluids were analyzed, although others fluids that are not so well known were also included. After a literature search, we found that the most commonly used organic working fluids for ORC applications are: R134a, R245fa, n-pentane, octamethultrisiloxane, toluene (Maraver *et al,* 2014), and, among the hydrocarbons (HCs), butane (R600), iso-butane (R600a), pentane (R601), iso-pentane (R601a) and hexane (Liu *et al*, 2013).

To select the working fluid, we took as a starting point a list of more than 130 different pure substances. Since not all of these pure substances were suitable for the given conditions, a first screening selection based on the need of meeting four main requirements was carried out to discard unsuitable working fluids. The requirements were:

1. The ODP had to be zero or close to zero
2. The GWP must be lower than 2000
3. The fluid critical temperature needed to be higher than the ambient temperature to make condensation possible
4. The saturation pressure at ambient temperature had to be higher than 1 kPa to limit vacuum in the condenser

From these requirements, a revised list of 44 working fluids was obtained. 39 working fluids were simulated and optimized by means of the Gradient Based Algorithm in MATLAB.

3.2 Assumptions and boundary conditions

The main assumptions and boundary conditions can be found in Table 1. Some data such as the pump and turbine efficiencies were assumed based on the information found during the literature review process, although the impact of these parameters was studied by means of a sensitivity analysis.

Values for the pump polytropic efficiency between 10% and 100% were simulated for the most optimal working fluids under subcritical and transcritical conditions. For the expander, the correlation from (Astolfi and Macchi, 2015) for axial turbines of one, two, and three stages to compute the polytropic efficiency as a function of the volume ratio and the size parameter was also analyzed for some of the best working fluids. This correlation states:

$$\eta_{turb} = f(V_r, SP) \qquad (1)$$

$$V_r = \frac{\rho_{in}}{\rho_{out}} \qquad (2)$$

$$SP = \frac{(\dot{m}_{out}/\rho_{out,s})^{1/2}}{(\Delta h_s)^{1/4}} \qquad (3)$$

Table 1. Assumptions and boundary conditions.

Hot source		
$T_{h,in}$	[ºC]	120
$T_{hmin,out}$	[ºC]	75
$T_{hmax,out}$	[ºC]	85
$\dot{m}$	[kg/s]	13.23
P_{out}	[bar]	3
Working fluid	[-]	H_2O
Cold source		
T_{in}	[ºC]	10
Working fluid	[-]	H_2O
$\Delta T_{c,max}$	[ºC]	10
$\Delta T_{c,min}$	[ºC]	5
Ambient conditions		
P_{amb}	[bar]	1.013
T_{amb}	[ºC]	15
Pinch points		
ΔT_{max}	[ºC]	8
ΔT_{min}	[ºC]	5
Pressure drops		
ΔP (evap,cond,rec)	[bar]	$0.01 \cdot P_{in}$
Pump		
Polytropic efficiency	[%]	70
Turbine		
Polytropic efficiency	[%]	80

Regarding the minimum outlet temperature of the hot source, the limitation was given to allow for use of the remaining heat in a low-temperature district heating system (this system is not considered in this work).

3.3 Objective function

Defining the objective function is crucial when it comes to optimizing the cycle. For this work, the variable to be optimized was the second law efficiency of the plant. A natural choice would have been optimizing the net power output or the first law efficiency (these three objective functions would give the same thermodynamic optimum), but optimizing the second law efficiency gives insight about how much potential for improvement is left.

The parameter to be optimized was the second law efficiency of the whole power plant, considering the recovery efficiency in the main heat exchanger. This is because not all the heat which was available from the heat source could be used (due to the limitation of the

hot source outlet temperature). The plant second law efficiency, cycle second law efficiency and recovery second law efficiency are related according to Equations. 4-5-6.

$$\eta_{II.cycle} = \frac{\dot{W}_{net}}{\dot{E}_{in}} \qquad (4)$$

$$\eta_{II.rec} = \frac{\dot{E}_{in}}{\dot{E}_{max}} \qquad (5)$$

$$\eta_{II.plant} = \frac{\dot{W}_{net}}{\dot{E}_{max}} = \eta_{II.cycle} \cdot \eta_{II.rec} \qquad (6)$$

Where $\dot{E}_{max}$ is the maximum exergy flow rate that can be extracted from the hot source (if the hot source stream is cooled down to ambient conditions, $T_0=15$ ºC and $p_0=1.013$ bar). All these equations were applied to the list of working fluids which have been studied (see Table 2).

The first law efficiency can be defined as:

$$\eta_{I.plant} = \frac{\dot{W}_{net}}{\dot{Q}_{in,max}} \qquad (7)$$

Where $\dot{Q}_{in,max}$ is the maximum heat flow rate that can be extracted from the hot source

Table 2. List of studied and simulated working fluids

	Chemical name	Alternative name	Class	$T_{crit}/T_{hot,in}$
1	Methyl alcohol	Methanol	Alcohol	1.3038
2	Ethyl alcohol	Ethanol	Alcohol	1.3092
3	Ethane	R170	Alkane	0.7766
4	Propane	R290	Alkane	0.9408
5	2-Methylpropane	Isobutane – R600a	Alkane	1.0373
6	Butane	R600	Alkane	1.0813
7	2,2-Dimethylpropane	Neopentane	Alkane	1.1032
8	2-Methylbutane	Isopentane – R601a	Alkane	1.1709
9	Pentane	R601	Alkane	1.1947
10	2-Metylpentane	Isohexane	Alkane	1.2659
11	Hexane	-	Alkane	1.2917
12	Heptane	-	Alkane	1.3739
13	Propene	Propylene – R1270	Alkene	0.9283
14	2-Methyl-1-propene	Isobutene	Alkene	1.0634
15	Ethylethylene	1-Butene	Alkene	1.0665
16	Cis-2-butene	Cis-butene	Alkene	1.1084
17	Propyne	-	Alkyne	1.0235
18	Benzene	Benzol	Aromatic	1.4295
19	Methylbenzene	Toluene	Aromatic	1.5052
20	Dimethyl carbonate	DMC	Carbonate ester	1.4168
21	Cyclopropane	-	Cycloalkane	1.0131
22	Cyclopentane	-	Cycloalkane	1.3016
23	Cyclohexane	-	Cycloalkane	1.4081
24	Dimethylether	DME	Ether	1.0184
25	Fluoromethane	R41	HFC	0.8070
26	Difluoromethane	R32	HFC	0.8934
27	1,1,1,2-Tetrafluoroethane	R134a	HFC	0.9518
28	Fluoroethane	R161	HFC	0.9545
29	1,1-Difluoroethane	R152a	HFC	0.9829
30	1,1,1,3,3-Pentafluoropropane	R245fa	HFC	1.0865
31	1,1,2,2,3-Pentafluoropropane	R245ca	HFC	1.1384
32	1,1,1,3,3-Pentafluorobutane	R365mfc	HFC	1.1700
33	2,3,3,3-Tetrafluoroprop-1-ene	R1234yf	HFO	0.9356
34	Trans-1,3,3,3-tetrafluoropropene	R1234ze	HFO	0.9729
35	Propanone	Acetone	Ketone	1.2924
36	Hexamethyldisiloxane	MM	Linear Siloxane	1.3193
37	Carbon Dioxide	CO_2 – R744	Inorganic	0.7736
38	Ammonia	R717	Inorganic	1.0312
39	H_2O	R718	Inorganic	1.6459

3.4 Degrees of freedom and constraints

When it comes to optimize an ORC, choosing the constraints and defining the degrees of freedom is of special importance. For this work, the simple cycle had 6 degrees of freedom, while the recuperated one had 7. These degrees of freedom were defined as dimensionless parameters related to the cycle variables which have maximum and minimum values, and take values between 0 and 1.

x_1 Heat source outlet temperature
x_2 Condenser temperature jump
x_3 Pressure at the inlet of the turbine
x_4 Pressure at the outlet of the turbine
x_5 Enthalpy at the inlet of the turbine
x_6 Enthalpy at the inlet of the primary heat exchanger
x_7 Enthalpy at the outlet of the condenser (only for the recuperated cycle

Regarding the constraints, they are the same for the simple and the recuperated cycles. These are:

c_1 The working fluid has to be subcooled at the inlet of the pump to avoid cavitation
c_2 If the pressure is subcritical, the working fluid has to be subcooled at the inlet of the evaporator to avoid the phase change in the recuperator
c_3 If the pressure is subcritical, the working fluid has to be saturated or superheated at the outlet of the evaporator (avoid trilateral and partial evaporation cycles)
c_4 The working fluid has to be saturated or superheated at the outlet of the expander (avoid vapour qualities lower than 1)
c_5 The pinch point in the evaporator has to be higher than the minimum temperature difference specified to avoid temperature crossing
c_6 The pinch point in the condenser has to be higher than the minimum temperature difference specified to avoid temperature crossing
c_7 The pinch point in the recuperator has to be higher than the minimum temperature difference specified to avoid temperature crossing

4 Results and discussion

4.1 Thermodynamic analysis

Some of the obtained results can be found in Table 3, which includes the working fluid mass flow, turbine and pump powers, net power output and first and second law efficiencies for the plant. The best obtained results for each different working fluid family have been included.

Figure 3 shows the results of the thermodynamic optimization as a function of the ratio of the critical temperature to the highest temperature of the cycle (hot source inlet temperature). Results show that those cycles with $T_{crit}/T_{hot,in}$ between 0.93 and 1.02 are giving the best plant second law efficiencies (between 31.33% and

33.46%). For values of $T_{crit}/T_{hot,in}$ lower than 0.9, the second law efficiency considerably drops, reaching values down to 21%, which correspond to fluids such as R41, CO_2 or ethane, whose best cycle configuration is the transcritical one. The reason for this is that, when the $T_{hot,in}$ is above the critical temperature, the best match between the hot source and the working fluid in the primary heat exchanger is found when the working fluid operates above its critical pressure. This is consistent with the conclusions from (Hærvig, 2016; Xu and Liu, 2013). The rest of the working fluids show their best performance at subcritical conditions, except for propylene and R1234yf. For $T_{crit}/T_{hot,in}$ values higher than 1.02, the performance of the cycle is slightly worse than the one found for the most optimal working fluids, although the difference found for the second law efficiency is not large. The working fluids showing the worst performance for the analyzed range were ammonia (recuperated) and methanol. The reason for this is that they are wet fluids and, in case of the ammonia, it has no potential for heat recovery (what is more, the recuperator would decrease the temperature at the inlet of the evaporator and the cycle efficiency). For the rest of the working fluids, the efficiencies are all above 31%, showing a better performance when the recuperator unit is installed.
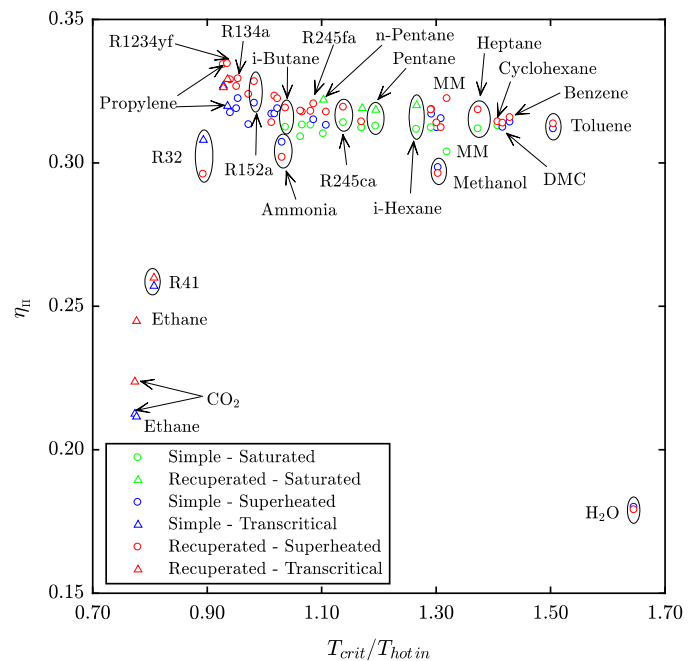

Figure 3. **Influence of the relationship between the working fluid critical temperature and the highest cycle temperature on the plant second law efficiency**

As it happens when the H_2O is used as the working fluid, some wet fluids require superheating to prevent condensation during the expansion (Bao and Zhao, 2013). For example, ammonia, H_2O, and ethanol (wet

fluids) require a high degree of superheating, while some isentropic fluids such as the R245fa or R365mfc require a low one. Most of the dry working fluids such as the butane heptane or n-pentane are operating at saturated conditions due to the positive slope of their diagram, which avoids the need of superheating the working fluid before its expansion.

As Figure 3 and Table 3 show, the working fluids giving the highest potential of improvement are alkanes, followed by the HFOs and HFCs, while the cycloalkanes do not show an appreciable efficiency increase when installing a recuperator unit. Even though ethane is the working fluid showing the highest potential of improvement when installing a recuperator unit, its performance is still one of the worst among all the analyzed working fluids. The highest efficiency gains are observed for the dry working fluids, because they have higher recuperation potentials than wet working fluids (ethanol and methanol experience an efficiency drop of 0.22% and 0.32% respectively when a regenerator unit is used). For the simple cycle layout, propylene shows the best results, operating at transcritical conditions and reaching an efficiency of 32.69%. For the recuperated cycle, R1234yf becomes the best working fluid, reaching a second law plant efficiency of 33.46%. It can also be seen that, while for the simple cycle layout many fluids are working at saturated conditions, for the recuperated configuration they show a better performance operating at superheated conditions. The reason is that a higher power production can be achieved when superheating, but this makes the cooling load to be higher (Bao and Zhao, 2013) unless a recuperator is used to decrease the cooling load and avoid wasted heat.

Regarding inorganic working fluids, the results for ammonia deviate from the tendency of Figure 3, while the H_2O is giving the worst performance. For both H_2O and ammonia, the recuperator is cooling down the working fluid at the inlet of the evaporator. Also, H_2O is giving the lowest power output. This is a consequence of the too low enthalpy drop taking place across the turbine for avoiding wet expansion. Regarding CO_2, it is also giving a low second law efficiency that improves with the use of a recuperator but it is still lower than the one for the rest of organic fluids. The results of this work indicate that the studied inorganic working fluids are not recommended for power generation in Rankine Cycles with low-heat source temperatures and low power capacities.

The Backwork Ratio (BWR) was plotted as a function of T_{crit} in Figure 4. It can be seen that those working fluids with the lowest critical temperatures (from 0ºC to 92ºC) are giving the highest BWRs. For subcritical cycles with a critical temperature between 100ºC and 160ºC, the simple cycles have higher BWRs than the recuperated cycles, due to the lower power production and higher power consumption they have.

For those working fluids with critical temperatures higher than 160ºC, the BWRs of the recuperated and simple cycles are very similar, since they can operate at lower pressures, guaranteeing a low pump power that does not appreciably change when moving from one cycle layout to another.

Finally, for toluene, the working fluid with the highest critical temperature, the maximum pressure to guarantee an efficient heat transfer process is the lowest one (0.33 bar), and this results into the lowest power consumption among all the results (only 0.39 kW), giving the lowest BWR.

The choice of the working fluid depends on the application at hand. If the target is power generation, the working fluids giving the highest second law efficiencies (those ones with a critical temperature close to the highest temperature of the cycle) should be chosen. Among these fluids, an economic analysis should be carried out to determine whether the use of a recuperator is justified or not. For those working fluids with a low second law efficiency, the power production is lower when comparing to the rest, and, since the amount of heat into the cycle is fixed, the heat rejected in the condenser is higher. This means that those working fluids with a low second law efficiency may have a promising potential for co-generation if the rejected heat in the condenser is used. For example, the ethane, with a second law efficiency of 21.15% rejects 2326.23 kW in the condenser at an inlet temperature of almost 56 ºC, while the recuperated propylene rejects 2225.36 kW with a plant efficiency of 33.42% and an inlet condenser temperature lower than 32 ºC.
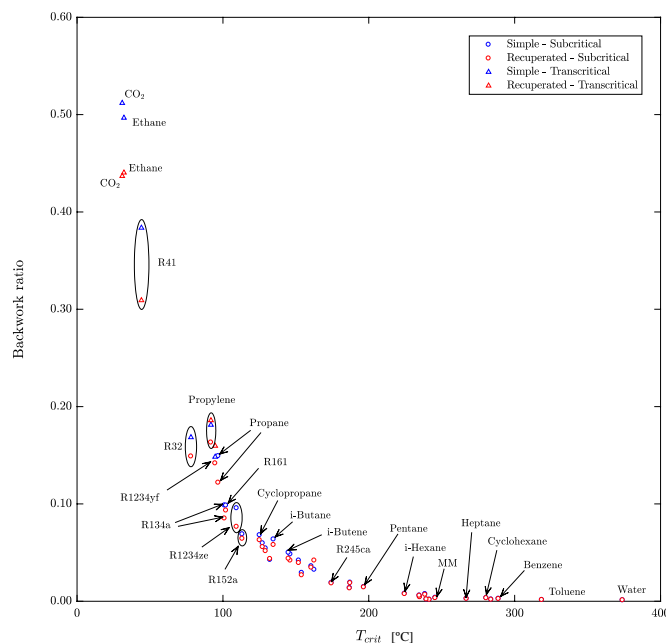

Figure 4. **Influence of the critical temperature on the BWR for all the simulated working fluids**

Table 3. Main obtained results

Fluid	Cycle layout	Cycle configuration	$\dot{m}_{w.f}$	$\dot{W}_{turb}$	$\dot{W}_{pump}$	$\dot{W}_{net}$	$\eta_{I.plant}$	$\eta_{II.plant}$
[-]	[-]	[-]	[kg/s]	[kW]	[kW]	[kW]	[%]	[%]
Acetone	Simple	Subcritical	3.83	274.76	1.23	273.53	4.69	31.69
Ammonia	Simple	Subcritical	1.91	276.88	11.80	265.08	4.55	30.71
Benzene	Simple	Subcritical	4.64	271.89	0.71	271.17	4.66	31.41
Butane	Simple	Subcritical	5.64	282.08	11.84	270.24	4.64	31.31
Cis-butene	Simple	Subcritical	5.36	279.29	9.06	270.23	4.64	31.30
CO_2	Simple	Transcritical	12.00	375.77	192.37	183.40	3.15	21.25
Cyclopropane	Simple	Subcritical	4.87	293.58	19.95	273.62	4.70	31.70
DMC	Simple	Subcritical	4.73	270.10	0.50	269.60	4.63	31.23
DME	Simple	Subcritical	4.87	291.00	17.36	273.65	4.70	31.70
Ethanol	Simple	Subcritical	2.38	272.65	0.47	272.18	4.67	31.54
i-Hexane	Simple	Subcritical	5.64	271.14	2.14	269.00	4.62	31.16
MM	Simple	Subcritical	7.75	263.19	0.97	262.22	4.50	30.38
Propane	Simple	Subcritical	5.91	322.06	48.05	274.00	4.70	31.74
Propylene	Simple	Transcritical	6.15	344.63	62.44	282.19	4.84	32.69
Propyne	Simple	Subcritical	4.27	291.07	15.85	275.22	4.72	31.88
R134a	Simple	Subcritical	10.75	305.29	30.1	275.19	4.72	31.88
R152a	Simple	Subcritical	7.05	297.40	20.50	276.91	4.75	32.08
R161	Simple	Subcritical	5.91	308.65	30.38	278.27	4.78	32.24
R1234yf	Simple	Transcritical	13	324.11	48.09	276.02	4.74	31.98
R1234ze	Simple	Subcritical	12.16	299.04	28.62	270.42	4.64	31.33
Toluene	Simple	Subcritical	0.96	269.53	0.40	269.13	4.62	31.18
Water	Simple	Subcritical	0.96	155.44	0.18	155.26	2.67	17.99
Acetone	Recuperated	Subcritical	4.13	276.81	1.19	275.63	4.72	31.86
Ammonia	Recuperated	Subcritical	1.89	273.08	11.91	261.18	4.47	30.19
Benzene	Recuperated	Subcritical	5.09	273.85	0.68	273.17	4.68	31.57
Butane	Recuperated	Subcritical	6.01	286.31	11.27	275.03	4.71	31.79
CO_2	Recuperated	Transcritical	12.59	343.64	150.12	193.53	3.32	22.37
Cyclohexane	Recuperated	Subcritical	5.75	272.76	0.90	271.86	4.66	31.43
DMC	Recuperated	Subcritical	5.26	271.96	0.47	271.49	4.65	31.38
DME	Recuperated	Subcritical	5.43	296.11	16.51	279.61	4.79	32.32
Heptane	Recuperated	Subcritical	5.98	276.23	0.72	275.51	4.72	31.85
i-Hexane	Recuperated	Subcritical	6.48	278.96	2.08	276.88	4.74	32.01
i-butane	Recuperated	Subcritical	6.58	293.03	16.97	276.06	4.73	31.91
Methanol	Recuperated	Subcritical	1.91	256.80	0.52	256.27	4.39	29.62
MM	Recuperated	Subcritical	9.40	279.85	0.94	278.91	4.78	32.24
n-Pentane	Recuperated	Subcritical	7.40	288.55	10.00	278.55	4.77	32.20
Propane	Recuperated	Subcritical	6.34	324.08	39.52	284.56	4.87	32.89
Propylene	Recuperated	Subcritical	6.31	345.57	56.41	289.17	4.95	33.42
Propylene	Recuperated	Transcritical	6.31	346.72	64.43	282.29	4.84	32.63
Propyne	Recuperated	Subcritical	4.68	294.02	15.17	278.85	4.78	32.23
R134a	Recuperated	Subcritical	13.16	308.82	26.31	282.50	4.84	32.66
R152a	Recuperated	Subcritical	7.74	303.48	19.53	283.96	4.86	32.82
R161	Recuperated	Subcritical	6.34	314.20	29.34	284.86	4.88	32.93
R245fa	Recuperated	Subcritical	11.42	284.94	7.63	277.31	4.75	32.05
R1234yf	Recuperated	Transcritical	14.41	337.61	53.89	283.73	4.88	32.92
R1234yf	Recuperated	Subcritical	14.48	337.26	47.84	289.42	4.96	33.46
R1234ze	Recuperated	Subcritical	13.90	303.39	23.24	280.16	4.80	32.38
Toluene	Recuperated	Subcritical	5.35	271.63	0.38	271.25	4.65	31.36
H_2O	Recuperated	Subcritical	0.96	154.92	0.18	154.74	2.65	17.90

4.2 Pump sensitivity analysis

Geothermal energy allows to work at part-load conditions when desired. (Landelle *et al*, 2017) stated efficiency losses of more than a 50% when operating below the nominal power plant conditions, and this, joint to the fact that each working fluid has a single and limited operating range in which it is optimal (Hærvig, 2016) has led to the conclusion that a pump sensitivity analysis should be carried out in this work. In this sensitivity analysis, one of the goals is to determine which working fluid is the optimal and for which range of pump polytropic efficiency, setting the nominal pump efficiency as 70% based on the literature review (Brosukiewicz-Gozdur, 2013). The analysis was then performed for 5 different working fluids. These were:

- Toluene: Working fluid operating under subcritical conditions with the lowest power consumption
- R161: Working fluid operating under subcritical conditions with the highest power consumption
- R1234yf: Working fluid operating under transcritical conditions with the lowest power consumption
- CO_2: Working fluid operating under transcritical conditions with the highest power consumption
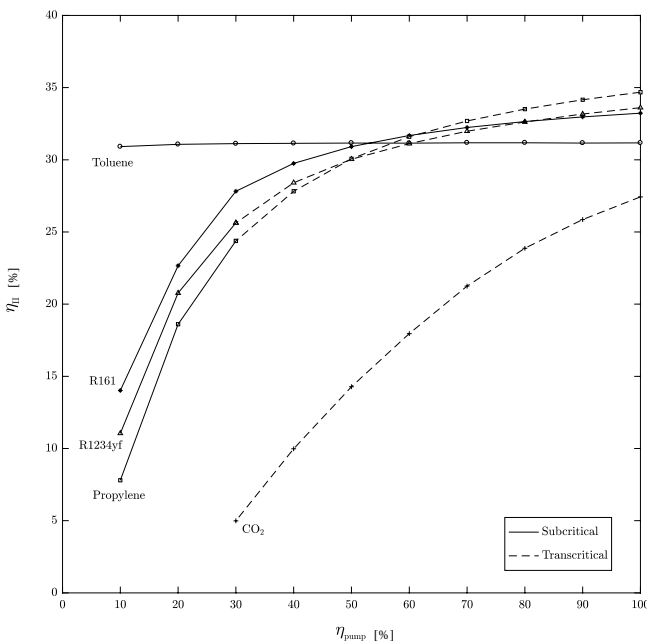- Propylene: Best working fluid for the non-recuperated layout.

Figure 5. **Effect of the pump polytropic efficiency on the overall plant second law efficiency for different working fluids**

Efficiencies from 10% to 100% were analyzed for all these fluids. Results for the second law plant efficiency can be found in Figure 5. It can be seen that the second

law efficiency suffers a great variation for all the fluids except for toluene. CO_2 shows the greatest variation, because it is the working fluid that requires the highest power consumption in the pump (high operating pressures), and any small change in the pump polytropic efficiency leads to important changes on the net power output and plant efficiency. For pump efficiencies reduced below 30%, the efficiency of the plant turns out to be negative (the power consumption is higher than the power produced in the expander). Even at a 100% pump efficiency, the CO_2 would still be the worst working fluid because of other irreversibilities in the cycle. The pump efficiency determines which working fluid is the best one at each operating point. Toluene is found to be the working fluid giving the second worst efficiency for efficiencies higher than a 70%, but, for pump efficiencies lower than 50%, it becomes the best choice. The reason for this is that the BWR for this fluid is low, and this means that any change of the pump polytropic efficiency has a minor impact on the efficiency. Propylene, which was the working fluid giving the best efficiency for the base case is still giving the best results for efficiencies higher than 70%. However, when the pump efficiency drops below this value, it is the second worst working fluid. This gives insight about how important the pump efficiency is for the performance of the cycle. Despite transcritical cycles might appear as the most sensible to pump efficiency changes, some subcritical cycles can be more sensitive. The BWR can be used as a good indicator of the sensitivity of the cycle to the pump efficiency. At the same time, the BWR is a strong function of the critical temperature of the working fluid. Figure 5 also shows that R1234yf and propylene need to move from transcritical to subcritical conditions to reach the optimal performance when the pump efficiency is below 30%, due to the high power consumption that operating at transcritical conditions with a low pump efficiency implies.

When comparing the obtained results with the literature, it was found that similar results were obtained in (Brosukiewicz-Gozdur, 2013). Even though this work includes the simulation of working fluids that were not considered by (Brosukiewicz-Gozdur, 2013), the same tendencies were obtained for the same studied working fluids, meaning that propylene is the working fluid with the highest specific power consumption, followed by the propane, while toluene presents the lowest one.

4.3 Turbine stages analysis

Since the specific work produced in the turbine is lower in the ORC than for gas and steam turbines, the expansion process can be handled in few stages. Assuming a fixed efficiency for the turbine (independent of factors such as the working fluid) leads to unrealistic results (Astolfi and Macchi, 2015). In this work, 1, 2 and 3 stage axial turbines have been studied in order to determine how this may affect the

performance of the whole plant when different working fluids are used. To do so, the turbine efficiency was studied as a function of the Volume Ratio (which accounts for compressibility effects (Macchi, 2013)) and the Size Parameter (which accounts for the dimensions of the expander). Each pair of SP and V_R gives a specific turbine efficiency, which increases with lower V_R and higher SP (Astolfi and Macchi, 2015). For the low density at the outlet of the turbine results into a high SP, which benefits the turbine efficiency. shows how the expander efficiency influences the overall power plant second law efficiency. Three different efficiencies were obtained for each studied working fluid, corresponding to a 1-, 2- or 3-stage expansion.

Results from Figure 6 show that increasing the number of stages leads to an increase of the expander efficiency for all the simulated working fluids, although some of them are more sensitive to this change than others. The best and worst working fluids are still the same (propylene and CO_2 respectively) when moving from 1 to 3 expansion stages. Regarding the rest of the working fluids, some as acetone, butane, and MM are experiencing the greater improvements when increasing the turbine number of stages, while the R32 and the propyne are experiencing the lowest ones.
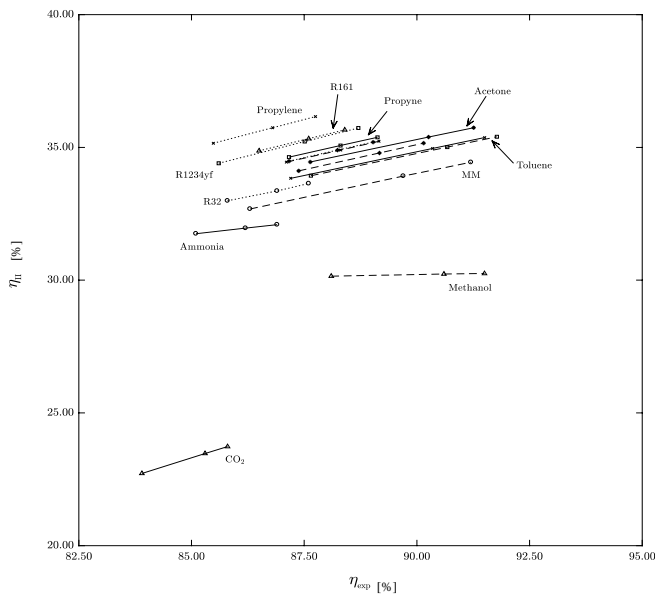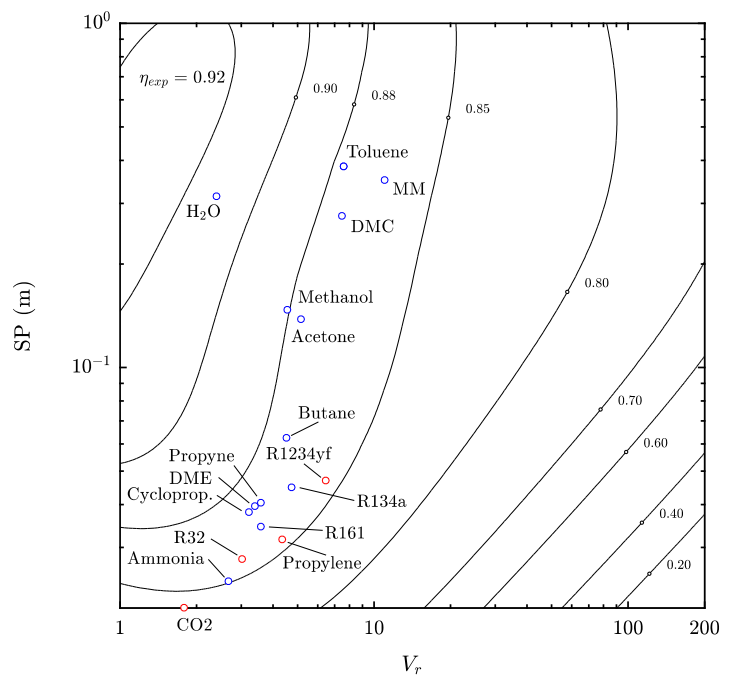

Figure 6. **Influence of the expander efficiency on the second law efficiency of the whole power plant for different working fluids**

Increasing the number of expansion stages for MM results into an increase of the turbine efficiency close to a 5%, which implies a second law efficiency improvement of 1.76%. Acetone also shows a great turbine efficiency improvement when increasing the

number of stages, a 3.63%, which results into a second law efficiency rise of 1.29%.

Fluids such as R32 experience only an improvement of 1.83% on the turbine efficiency when moving from 1 to 3 stages of expansion, and this means a plant efficiency increase of only 0.65%. On the other side, for CO_2, even though the turbine efficiency experiences one of the highest improvements (3.72%), the second law efficiency does not experience a great change (only 1%).

Figure 7 shows how the SP and V_R influence the efficiency of the turbine with only one stage for the different simulated working fluids. Working fluids with the highest efficiencies are the H_2O, methanol, acetone and toluene, while those ones with the lowest turbine efficiencies are CO_2, ammonia and propylene. For the low density at the outlet of the turbine results into a high

SP, which benefits the turbine efficiency.

Figure 7. **Volume ratio and Size Parameter influence on the expander efficiency with one stage of expansion and for the different simulated working fluids**.

5 Conclusions

After having carried out an analysis of the ORC for 39 different working fluids and under different cycle layouts and configurations, some conclusions have been reached:

1. Those working fluids with a critical temperature and maximum temperature of the cycle relationship between 0.93 and 1.02 (such as propylene and R1234yf) are showing the best results. Fluids with too low critical temperatures are giving the lowest efficiencies.

2. Regarding the use of a recuperator, the dry and isentropic fluids are experiencing the highest efficiency improvements, while some wet fluids do not have potential to operate under this cycle layout (the second law efficiency for the whole power plant drops).

3. Fluids with a low critical temperature are much more dependent on the pump performance than those ones with a high critical temperature, due to the high Backwork Ratios of the former.

4. Increasing the number of stages for the turbine results into an efficiency improvement for all the studied cases. The degree of improvement is a function of the variation of the V_R and SP.

5. It was demonstrated that H_2O and CO_2 are not suitable working fluids for an ORC with a low-temperature heat source for low-capacity power plants. In case of co-generation for the cycle or higher power productions, their potential may increase, but this should be studied.

Nomenclature

Acronyms and symbols

BWR	Backwork Ratio	[kW/kW]
ρ	Density	[kg/m^3]
η	Efficiency	[%]
h	Enthalpy	[J/kg]
$\dot{E}$	Exergy flow	[kW]
$\dot{Q}$	Heat flow	[kW]
$\dot{m}$	Mass flow	[kg/s]
MW	Molecular Weight	[kg/mole]
$\dot{W}$	Power	[kW]
p	Pressure	[bar]
SP	Size Parameter	[m]
T	Temperature	[K]
V_r	Volume ratio	[-]
GWP	Global Warming Potential	
HC	Hydrocarbon	
HFC	Hydrofluorocarbon	
HFO	Hydrofluoroolefin	
ODP	Ozone Depletion Potential	
ORC	Organic Rankine Cycle	

Subscripts

amb	ambient
c	condenser
crit	critical
hot	hot-source
in	turbine inlet
s	isentropic
out	turbine outlet
rec	recuperator
turb	turbine
w.f	working fluid

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