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SIMULATION OF A SMALL SCALE ORGANIC RANKINE CYCLE

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Abstract. *Due to environmental issues and energetic concerns, specially, increasing energetic demand in the coming years and fossil fuels sources depletion, different technologies have been studied in the attempt to make energetic sector more sustainable and efficient. In this context, organic Rankine cycle (ORC) – modification of the traditional steam Rankine cycle (SRC) that replaces water vapor as working fluid by an organic fluid, which has lower boiling temperature and higher vapor pressure, when compared to water – rises as a promising technology for electricity generation from low temperature heat sources, that could be supplied from renewable energy sources and waste heat from industrial processes or energy conversion. This paper aims to model a low scale ORC (10 kW), considering different fluids and operational conditions to investigate the cycle responses. The results are compared to some experimentally obtained values from the literature. Among the main tested working fluid candidates, isobutene and isopentane (hydrocarbons), R123 (HCFC), R11 (CFC) and R245fa (HFC) provided the higher cycle efficiency. R123 was considered as working fluid to compare cycles with and without heat regenerator. Also, the influence of heat source temperatures was investigated.*

Keywords: *Organic Rankine cycle, Renewable energy, Energy efficiency*

1. INTRODUCTION

The gases present in the atmosphere retain the heat from the incident solar radiation on Earth, preventing this heat from being emitted from the planet. This phenomenon is called the greenhouse effect, and it is vital to the existence of life on Earth; however, the massive emission of these gases into the atmosphere intensifies this effect, being responsible for the increase of the global temperature.

Concern about global warming and levels of greenhouse gas emissions is not a recent issue; in the nineteenth century, the study of climate change and ice ages culminated with the identification of greenhouse gases, and theories that explain human activity as responsible for the increase of global temperatures. Scientists such as Samuel P. Langley, John Tyndall, Svante Arrhenius and Arvid Högbom carried out work related to the measurement of carbon dioxide (CO_2) and water vapor in the atmosphere with increasing temperature. Since carbon emissions in this period were still relatively low, Arrheius believed that the effects of CO_2 on the climate would take thousands of years to occur (ANDERSON, HAWKINS, JONES, 2016).

There is a strong relation between of an exponential increase since the 1800s in the concentration of CO_2 in the atmosphere and the burning of fossil fuels - oil and its byproducts, coal and gas - especially coal, accompanied by the intense deforestation caused by human being.

According to data published by the IEA (2016), the sectors responsible for the largest amounts of CO_2 emitted in the world are, in this order, power generation, industrial and transportation. In Brazil, the transportation sector leads emissions, followed by industrial and power generation.

Nevertheless, a 28% increase in global energy consumption between 2015 and 2040 is estimated, possibly reaching up 700 billion BTU. Thus, both the decrease in the use of fossil energy sources and the increase of renewable energy sources - solar, wind, geothermal, biomass, biofuels, hydropower and hydrogen -, as well as improvements in energy efficiency, since its generation to its consumption are extremely desirable as they mitigate both the problems of energy supply and global warming. For this purpose, strategies, policies and technologies developments are fundamental.

Rankine cycle is one of the methods used to convert thermal energy into electrical energy and traditionally uses water as working fluid (SRC). In this cycle, water is pressurized and pumped to the evaporator, where heat is supplied from an external source until vapor state is obtained. Then, vapor is delivered to an expansion device (typically a turbine) coupled to an electric generator; during the expansion, vapor produces shaft rotation witch powers the generator. After the expansion to a lower pressure, water is led to the condenser, rejecting heat to a heat sink until liquid state is achieved and pumped again to restart the cycle. However, the use of SRC to generate electricity from heat sources below 370 ° C becomes inefficient and economically infeasible (HUNG, SHAI and WANG, 1997); but organic Rankine cycle (ORC) – modification of the traditional Rankine cycle, in which water is replaced by an organic working fluid, which has a lower boiling point and a higher vapor pressure than water – needs less heat during boiling, that occurs in lower pressures and temperatures than Rankine Cycle (TCHANCHE *et al.*, 2011), making it possible to use low and medium temperature heats (between 60 and 250°C) for electric power generation.

In this context, the ORC is a promising technology to increase electricity generation and/or to improve energy efficiency due to the possibility of using heats from renewable energy sources and unused heats by industrial processes or power generation (combustion exhaust gases, cooling waters or vapors at lower temperatures) and which are rejected for the environment, and for their simplicity, low maintenance and easy operation.

This paper aims to model a low scale ORC (input of 10 kW in the evaporator), considering different fluids and operational conditions to investigate the cycle responses. The results are compared to some experimentally obtained values from the literature. Further analysis is carried out on cycles efficiencies and features configurations.

1.1 Previous works

Table 1 below shows a non-extensive list of previous works.

Table 1 – Comparison between some experimental works found in literature.

Author	Wang, Zhao, Wang (2010)			Jung, Taylor, Krumdieck (2015)	Bracco *et al.* (2013)	
Recuperator	No			No	No	
Expander	Valve			Scroll	Scroll	
Fluid	R245fa	R245fa/R152a (0,9/0,1)	R245fa/R152a (0,7/0,3)	R245fa/R365mfc (0,485/0,515)	R245fa	
Mass flow [kg/s]	1,73	1,69	1,51	0,07	-	
Pressure at the evaporator outlet [MPa]	0,23-0,37	0,28-0,48	0,38-0,8	1,41	0,9-1,3	
Pressure at the condenser inlet [MPa]	0,14-0,18	0,14-0,20	0,26-0,35	0,13	0,14-0,18	
Temperature at the evaporator outlet [°C]	40-105	25-100	30-101	136,4	98-195	89-147
Temperature at the condenser inlet [°C]	30-60	21-63	20-60	88,1	-	
Temperature at the condenser outlet [°C]	20-34	11-22	15-25	15,9	-	
Generated power [W]	4-7 5,98 (average)	1,8-7,8 6,03 (average)	4,8-9 7,72 (average)	700	1050-1320	1340-1500
Pump power consumption [W]	1,04 (average)	0,88 (average)	1,07 (average)	-	-	
Cycle efficiency [%]	4,16 (average)	4,29 (average)	5,59 (average)	3,9	7,2-8,8	7,2-8,2

Table 1 – Comparison between some experimental works found in literature (continuation).

Author	Chaiyat and Kiatsiriroat (2015)	Peris *et al.* (2015)	Dickes *et al.* (2017)		Shao *et al.* (2017)
Recuperator	No	Yes	Yes		No
Expander	Screw	Vane	Scroll		Turbine
Fluid	R245fa	R245fa	R245fa		R123
Mass flow [kg/s]	-	-	0,015-0,068	0,277-0,619	0,197
Pressure at the evaporator outlet [MPa]	0,81-1,12	2,72-2,94	0,65-1,43	0,98-2,05	0,31-0,35
Pressure at the condenser inlet [MPa]	0,23-0,24	0,3-0,94	0,16-0,66	0,27-0,43	0,14-0,21
Temperature at the evaporator outlet [°C]	85,7-94,6	148,6-151,2	82,8-117,9	102,9-135,8	100
Temperature at the condenser inlet [°C]	59,5-75	44,5-84,7	32,9-69,6	61,6-83,7	78-83
Temperature at the condenser outlet [°C]	37	37,4-81	17,9-52,1	35-47,6	26-51
Generated power [W]	9000-21500	3890-6900	63-1405	1134-6889	889,5-1242,7
Pump power consumption [W]	1240-1900	1060-1300	39-150	251-939	0,06-0,43
Cycle efficiency [%]	4,71-8,73	4,9-8,8	0,31-8,5	1,48-4,91	3,05-5,2

It is important to check on efficiency calculation method to compare different systems; Peris *et al.* (2017) and Dickes *et al.* (2017) consider generator efficiency into the system thermal efficiency, Chaiyat and Kiatsiriroat (2015) divide expander work by the sum of input heat and pumps power.

Despite different experimental settings and different efficiency calculation methods are adopted, the efficiency ranges are within the expected for an ORC system, which is limited by Carnot´s efficiency, as discussed in the next topics.

2. METHODOLOGY

This paper aims to model a low scale ORC, considering different fluids and operational conditions, to investigate fluid candidates and cycle responses, comparing it to some experimentally obtained values from the literature.

The following hypotheses have been adopted:

- Input heat = 10 kW
- Pump efficiency = 0.65
- Expander efficiency = 0.7
- Heat recuperator effectiveness = 0.9
- Constant pump, expander and recuperator efficiencies
- Saturated vapor in the evaporator outlet
- Saturated liquid in the condenser outlet
- Negligible pressure drop

Varying the boiling temperature, the system responses evaluated were: cycle efficiency, mass flow, net power and evaporation pressure.

Then, despite being a HCFC gas, R123 was chosen as working fluid for cycle analysis and comparison of the effect of a recuperator as feature, due to its good performance but this fluid should not be adopter in a real system because its impact on environment.

To model an organic Rankine cycle, the following equations were considered. These equations can be obtained from mass and energy balance, considering each cycle component as a control volume. In all cases, kinetic and potential energies changes are neglected. Pump and expander are considered adiabatic control volumes.

The work rate introduced into the cycle by the pump can be determined by:

$$\dot{W}_p = \dot{m}_{wf} \cdot (i_2 - i_1) = \frac{\dot{m}_{wf} \cdot (i_{2s} - i_1)}{\eta_p} \tag{2.1}$$

When there is a heat recuperator in the cycle, the heat exchange rate in this component is given by:

$$\dot{Q}_r = \dot{m}_{wf} \cdot (i_{2r} - i_2) = \dot{m}_{wf} \cdot (i_4 - i_{4r}) \tag{2.2}$$

$$\varepsilon_r = \frac{Q_r}{Q_{máx}} = \frac{i_{2r} - i_2}{i_4 - i_{mín}};\ i_{mín} = enthalpy(P = P_4, T = T_2) \tag{2.3}$$

In the evaporator, the thermal exchange between the working fluid and the heat source can be determined by:

$$\dot{Q}_e = \dot{m}_{wf} \cdot (i_3 - i_{2r}) \tag{2.4}$$

If there is no heat recuperator,

$$i_{2r} = i_2 \rightarrow \dot{Q}_r = 0\ ,\ \dot{Q}_e = \dot{m}_{wf} \cdot (i_3 - i_2) \tag{2.5}$$

And, to compare cycles with and without heat recuperator, input heat is fixed, respecting the relation:

$$\dot{Q}_{in} = \dot{Q}_r + \dot{Q}_e \tag{2.6}$$

In the expander, the shaft work produced during the expansion can be determined by:

$$\dot{W}_t = \dot{m}_{wf} \cdot (i_3 - i_4) = \dot{m}_{wf} \cdot (i_3 - i_{4s}) \cdot \eta_t \tag{2.7}$$

In the condenser, the thermal exchange between the working fluid and the cooling fluid can be given by:

$$\dot{Q}_c = \dot{m}_{wf} \cdot (i_{4r} - i_1) \tag{2.8}$$

If there is no heat recuperator,

$$i_{4r} = i_4 \rightarrow \dot{Q}_r = 0\ ,\ \dot{Q}_c = \dot{m}_{wf} \cdot (i_4 - i_1) \tag{2.9}$$

2.1 Cycle performance check

Two of the most used parameters in performance evaluation of thermodynamic cycles, including ORCs, are thermal efficiency (first law) and exergy efficiency (second law). (BRAIMAKIS *et al.*, 2015)

Thermal efficiency is calculated by the ratio between net power generated and thermal power supplied to the cycle:

$$\eta_{th} = \frac{\dot{W}_t - \sum \dot{W}_p}{\dot{Q}_e} \tag{2.10}$$

$$\eta_{Carnot} = 1 - \frac{T_{cold}}{T_{hot}} \tag{2.11}$$

Equation 2.11 represents the efficiency of Carnot for one cycle operating enter two reservoirs, hot temperature T_{hot} and cold temperature T_{cold}; however, for heat sources with limited heat capacity, this metric is not representative. (LECOMPTE *et al.*, 2015). Thus, for a reversible Carnot cycle coupled to a source of finite heat capacity, represented in Figure 2.1, the heat fraction recovered from the source can be given by:

$$\eta_{frac,Carnot} = \frac{T_{hot} - T_2}{T_{hot} - T_{cold}} \tag{2.12}$$

The thermal efficiency of this reversible Carnot cycle is then:

$$\eta^*_{Carnot} = \eta_{frac,Carnot} \cdot \eta_{Carnot} = \frac{T_{hot} - T_2}{T_{hot} - T_{cold}} \cdot \left(1 - \frac{T_{cold}}{T_2}\right) \tag{2.13}$$

Deriving Eq. (2.13) with respect to T_2 and equaling to zero, we obtain the maximum theoretical efficiency of the cycle:

$$\frac{d}{dT_2}\left(\eta^*_{Carnot}\right) = 0 \rightarrow T_2 = \sqrt{T_{hot} \cdot T_{cold}} \tag{2.14}$$

$$\eta^*_{Carnot,máx} = 1 - \sqrt{\frac{T_{cold}}{T_{hot}}} \tag{2.15}$$

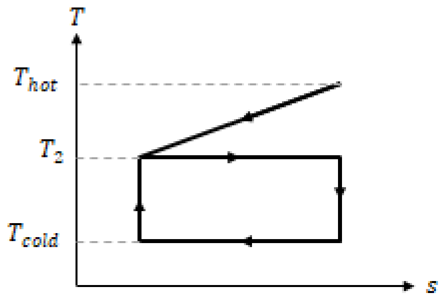

Figure 2.1 - Carnot's external reversible cycle coupled to a source of finite heat capacity. Source: adapted from Lecompte *et al.* (2015).

However, thermal efficiency considers only the first law of thermodynamics. It takes on account only energy balance but ignore the quality of energy and its conversion potential to work. So, efficiency based on exergy, or second law efficiency, is adopted. According to DiPippo (2008), exergetic efficiency is the most adequate parameter to compare thermodynamic performances of power plants.

The exergectic cycle efficiency may be written as:

$$\epsilon_{ex} = \frac{recovered\ energy}{supplied\ energy} = \frac{\dot{W}_t - \sum \dot{W}_p}{\dot{E}_s} \tag{2.16}$$

$$\dot{E}_s = \left(1 - \frac{T_0}{T_s}\right) \cdot \dot{Q}_{in};\ \dot{Q}_{in} = \dot{Q}_e \tag{2.17}$$

$$\epsilon_{ex} = \frac{\dot{W}_t - \sum \dot{W}_p}{\left(1 - \frac{T_0}{T_s}\right) \cdot \dot{Q}_{in}} = \frac{\eta_{th}}{\eta_{Carnot}} \tag{2.18}$$

Equation (2.18) combines Eq. (2.16) and Eq. (2.17), and shows that exergectic efficiency can be seeing as a measure of proximity to Carnot´s efficiency. In other words, it measures how close the efficiency is to the theoretical maximum efficiency for such heat source. Using the same logical, Eq. (2.15) exhibits a modified Carnot´s efficiency; then, the measure of how close cycle efficiency is to this modified efficiency can be written as:

$$\epsilon^*_{ex} = \frac{\eta_{th}}{\eta^*_{Carnot,max}} \tag{2.19}$$

3. RESULTS AND DISCUSSION

In order to analyze the applicability of fluid candidates to be used in the ORC, computer simulations were carried out. Varying boiling and condensing temperatures, the following system responses were evaluated: cycle efficiency, mass flow, net power, boiling pressure and condensing pressure. This procedure was run for various working fluid candidates, and the results of this analysis were plotted in the graphs of Fig. 1-4 for a condensing temperature of 40 °C. In Tab. 2, we present a brief comparison between cycle efficiency obtained in the present study and that obtained by other authors.

Some fluids are more suitable for ORC applications. Among the hydrocarbons, isobutane and isopentane are outstanding, since, besides presenting good thermal efficiencies, they have condensation pressures higher than atmospheric pressure and moderate evaporation pressures. Cyclohexane, n-hexane, n-heptane also provide good cycle performance, but have sub-atmospheric condensation pressures, which requires a de-aeration system. Ethanol is another fluid which provides good performance but has bub atmospheric pressure and besides is a “wet fluid”, that means that part of the fluid may condensate during expansion, leading to problems on the expander. To avoid this problem that may occur to wet fluids, superheated vapor is used instead of saturated vapor in the expander inlet. Also noteworthy are R123 (HCFC), R11 (CFC) and R245fa (HFC).

The proposed model used constant efficiency values for the components of the cycle, namely pump, turbo-generator and heat recuperator, and ignored pressure drops and consumption of auxiliary pumps (hot and cold sources). This modeling approach, although simpler, may provide greater errors in relation to the value measured experimentally. Dickes *et al.* (2017) propose the semi-empirical method or the polynomial regression method to obtain greater precision efficiency curves and better simulation results in relation to the experimental values.

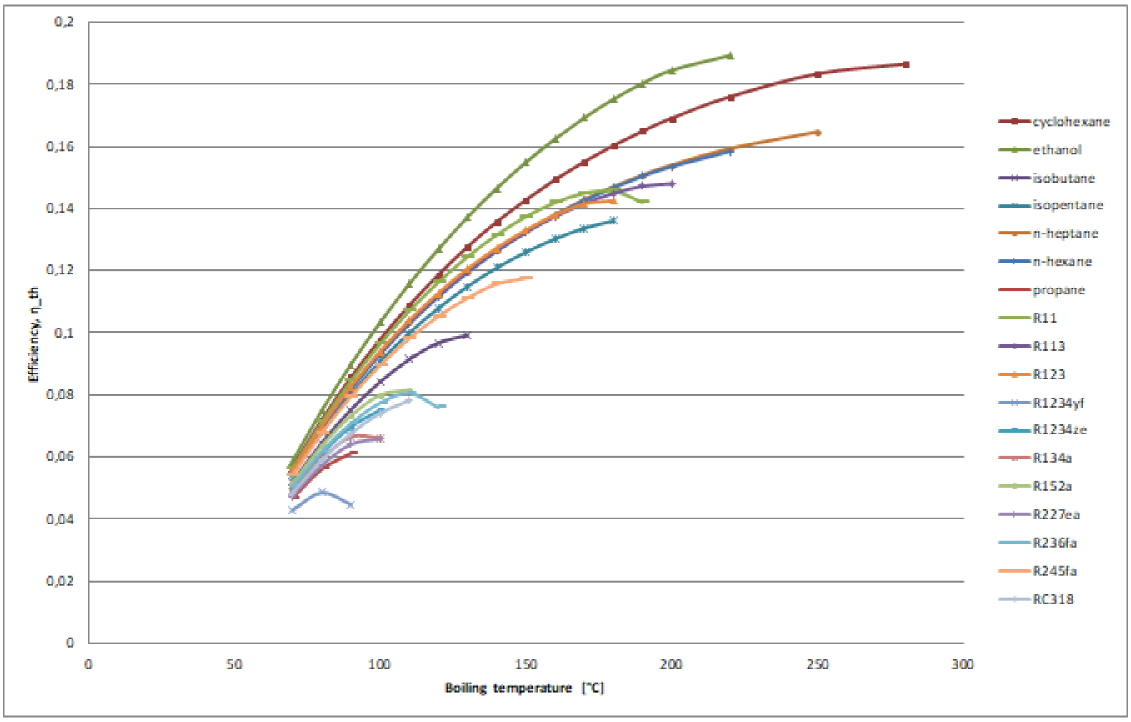

Figure 1. Efficiency versus boiling temperature at condersing temperature equal to 40 °C.

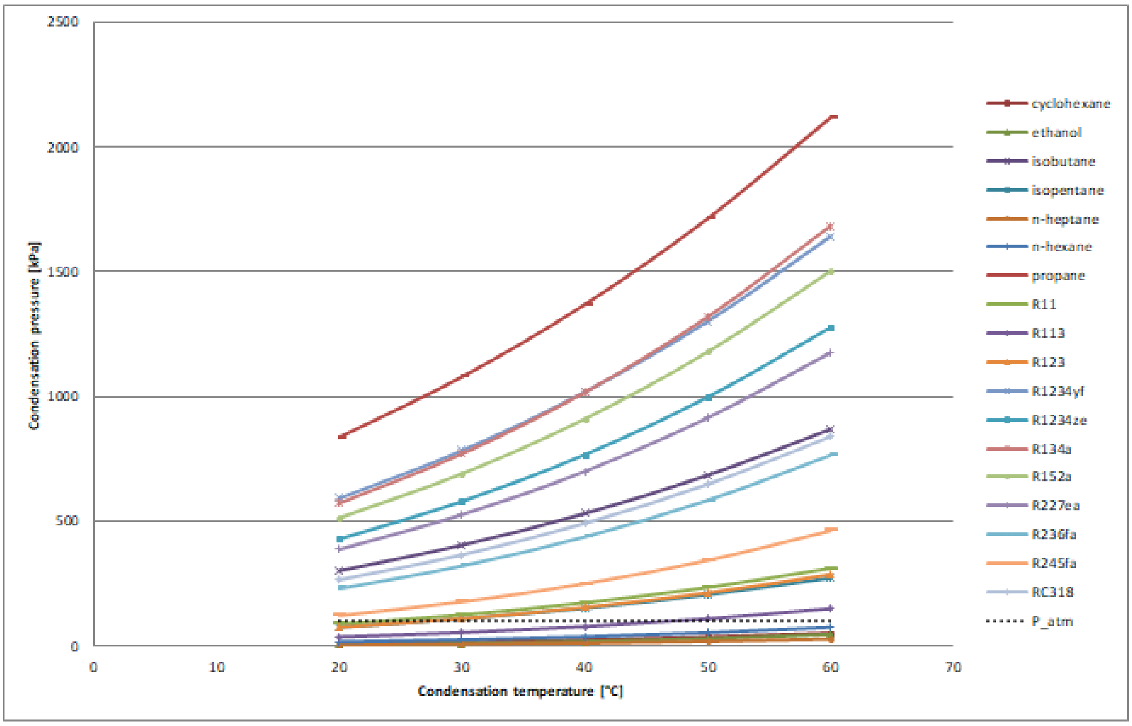

Figure 2. Condensation pressure versus condensation temperature.

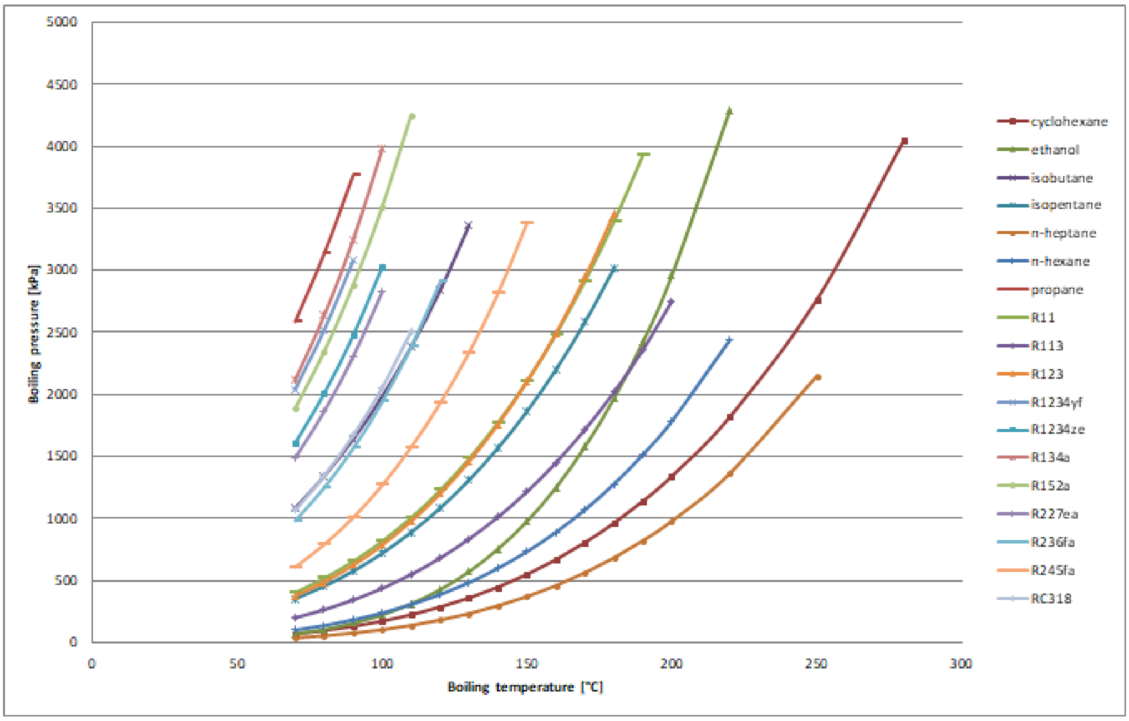

Figure 3. Boiling pressure versus boiling temperature.

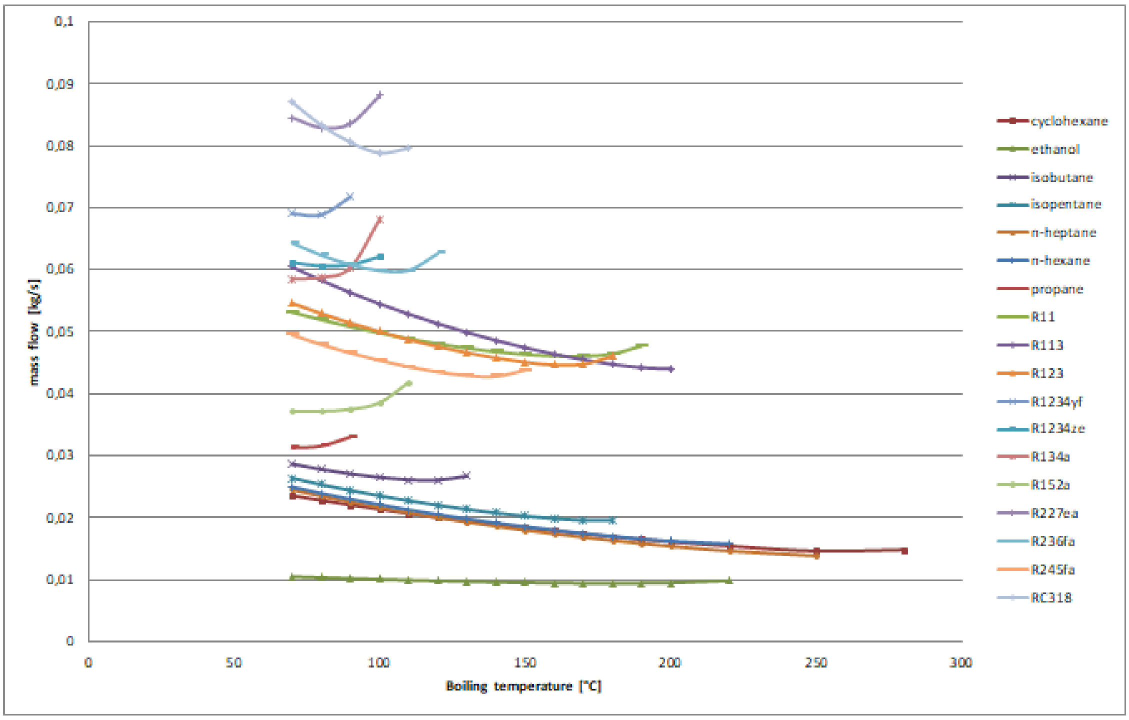

Figure 4. Mass flow rate versus boiling temperature at condersing temperature equal to 40 °C.

Table 2. Comparison between efficiency obtained in the present study and efficiency obtained by the other authors.

	Fluid	Data reported by the authors			Data of the present study		
Author		$T_{saí,e}$ [°C]	$T_{saí,c}$ [°C]	Efficiency [%]	$T_{saí,e}$ [°C]	$T_{saí,c}$ [°C]	Efficiency [%]
Wang, Zhao, Wang (2010)	R245fa	40-105	20-34	4,16-5,59	70-110	40	4,9-8,8
Bracco *et al.* (2013)	R245fa	98-195 89-147	- -	7,2-8,8 7,2-8,2	90-150	40	7,1-10,4
Chaiyat and Kiatsiriroat (2015)	R245fa	85,7-94,6	37	4,71-8,73	85-95	40	6,6-7,6
Peris *et al.* (2015)	R245fa	148,6-151,2	37,4-81	4,9-8,8	150	40-80	6,3-10,4
Dickes *et al.* (2017)	R245fa	82,8-117,9 102,9-135,8	17,9-52,1 35-47,6	0,31-8,5 1,48-4,91	80-120 100-140	20-50 35-50	4,6-11,7 6,8-10,8
Shao *et al.* (2017)	R123	100	26-51	3,05-5,02	100	25-50	7,1-10,3

In Tab. 2, it is compared some reported experimental measures with simulated points of the proposed model. It can be concluded that the present simple model leads to thermal efficiency values slightly higher than the values experimentally measured in other works, but present coherent values for the considered operational ranges. It may be explained due to some reasons, including generator coupling, considerations of auxiliaries pumps (cooling fluid and oil), losses, components performance, off-design conditions and different operational conditions (for example, presence of superheating).

The efficiency ranges obtained for each experimental set is evidence that the components should be adjusted to operational settings that meet the design point.

Then, adopting R123 as working fluid, making the above mentioned assumptions, cycles with and without regenerator were run and their responses were plotted as below in Fig. 5-12.

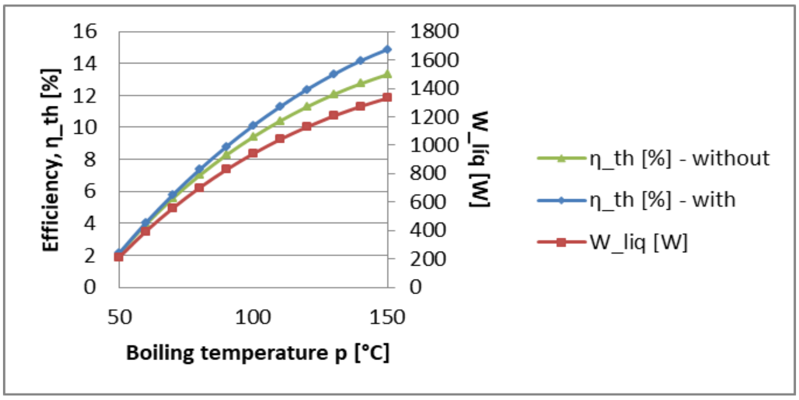

Figure 5. Thermal efficiency and net power output versus boiling temperature, for cycles with and without heat recuperator.

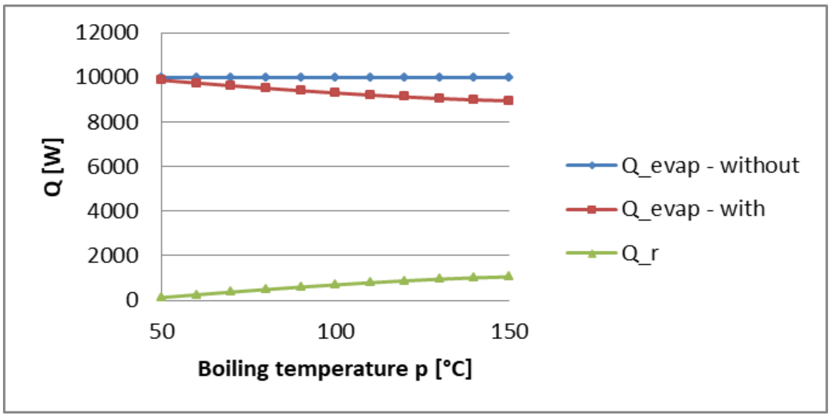

Figure 6. Heat exchange in evaporator and in recuperator versus boiling temperature, for cycles with and without heat recuperator.

As expected, a cycle with heat recuperator has higher efficiency, than the same cycle without this feature, keeping the same conditions, once heat input is lower and output work is the same. For the considered cycles, the presence of a heat recuperator saved from 1.3% to 10.5% the input heat for boiling temperatures of 50 °C to 150 °C, and increased thermal efficiency from 1.3% to 11.8%.

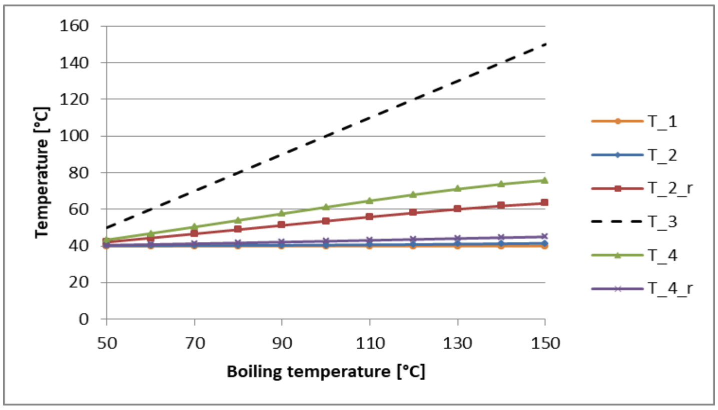

Figure 7. R123 temperatures versus boiling temperature, for cycles with recuperator.

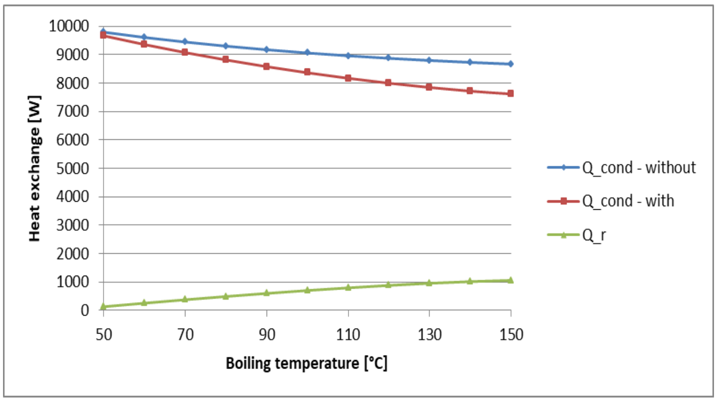

Figure 8. Heat exchanges on condenser and on recuperator versus boiling temperature, for cycles with and without recuperator.

Figure 7 shows that even the temperature of R123 on the recuperator outlet being close to the temperature on the expander outlet, this temperature is much smaller than the evaporation temperature, and the recovered heat is still much lower than the heat required to condense the refrigerant to restart the cycle, as illustrated by Fig. 8, that highlights that the presence of a heat recuperator alleviates heat exchanges on condenser and gets more significate on higher boiling temperatures. Moreover, as cycle efficiency increases with higher boiling temperatures, heat rejected by the condenser to the cooling fluid decreases, considering the same amount of input heat in the evaporator. This is important because heat exchangers sizes diminish, as well as their cost.

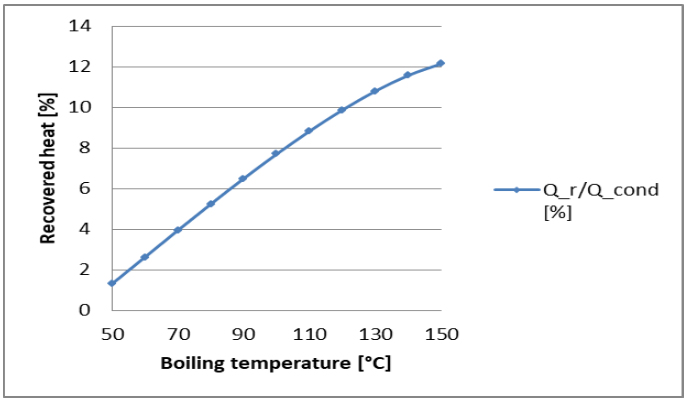

Figure 9. Fraction of rejected heat recovered versus boiling temperature, for cycles with recuperator.

Figure 9 shows that recovered heat gets more significant as boiling temperature increases: representing from 1.3% (for a boiling temperature of 50 °C) to 12.1% (for a boiling temperature of 150 °C) of the required heat for condensing the working fluid with no heat recovery at the same boiling point.

For R123 boiling at 50 °C, heat source temperature was let to increase. Exergectic and Carnot efficiencies were plotted below, as well as their modified forms:

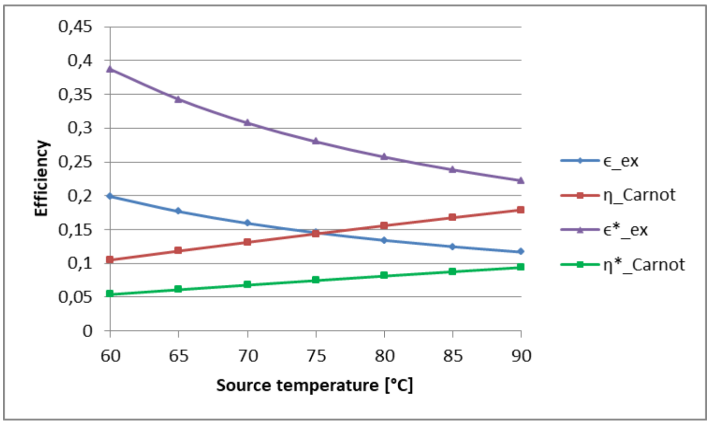

Figure 10. Exergectic and Carnot efficiencies for a cycle without recuperator with boiling temperature of 50 °C versus heat source temperature.

So, for an arbitrary boiling temperature, the exergectic efficiency will be grater the closer the boiling temperature is to source temperature.

Considering heat source temperature 10 °C higher than the boiling point, cycles were run and exergectic and Carnot efficiencies were plotted below:

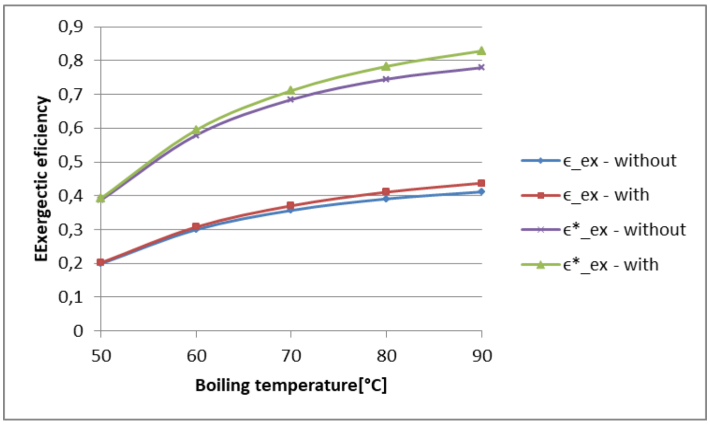

Figure 11. Isentropic, exergectic and Carnot efficiencies for cycles without recuperator versus boiling temperature.

Considering cycles with and without heat recuperator and heat source temperature 10 °C higher than the boiling point, exergectic efficiencies were plotted below:

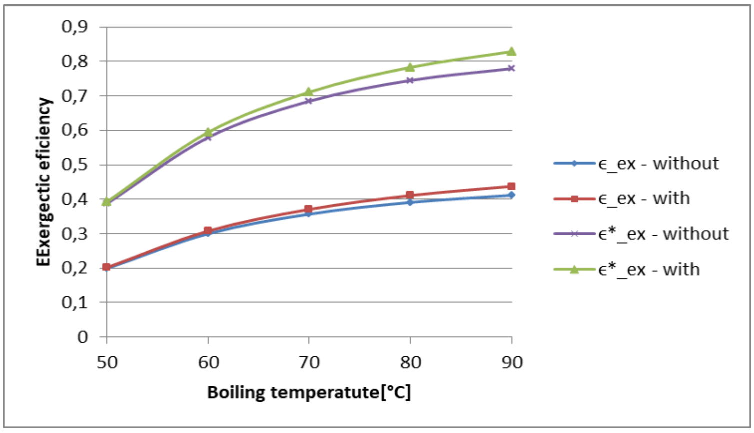

Figure 12. Exergectic efficiencies for cycles with and without recuperator versus boiling temperature.

Thermal exchanges within temperature ranges are limited by Carnot´s efficiencies; nevertheless, it is important to choose the proper indicators to evaluate whether a cycle is efficient or not. Considering modified Carnot´s efficiency, a system that would be considered as less efficient can be understood as more efficient, once the maximum theoretical power is adjusted. Despite the lower thermal efficiencies, considering low temperatures heat sources, exergectic efficiencies are within considerable ranges (around 80%), positioning ORC as a feasible technology to generate electric power from low temperature heat sources.

4. CONCLUSIONS AND RECOMMENDATIONS

It can be concluded that the present simple model leads to thermal efficiency values slightly higher than the values experimentally measured in other works, but present coherent values for the considered operational ranges.

Some fluids seem to be more suitable to ORC applications. Among hydrocarbons, isobutene and isopentane appear as most suitable because they have good thermal efficiencies, condensing pressures higher than atmospheric pressure and moderate boiling pressures; cyclohexane, n-hexane, n-heptane also present good cycle performance but they have sub atmospheric condensation pressures, which require a deaerator system. Among the main tested working fluid candidates, isobutene and isopentane (hydrocarbons), R123 (HCFC), R11 (CFC) and R245fa (HFC) provided the higher cycle efficiency.

Analyzing the cycles with R123 as working fluid, the presence of a heat recuperator increases the cycle efficiency (from 1.3% to 11.8%), but the greatest amount of heat is still lost on the condenser. For the same conditions, and same power output, heat input is reduced from 1.3% to 10.5%.

Providing a good match between working fluid boiling temperature and heat source temperature leads to higher exergectic efficiencies.

It is important to choose the proper indicators to evaluate cycles´ efficiencies. Despite the lower thermal efficiencies, considering low temperatures heat sources, exergectic efficiencies are within considerable ranges, positioning ORC as a feasible technology to generate electric power from low temperature heat sources.

5. SYMBOL LIST

The list of symbols below indicates the meaning of variables, symbols and subscripts.

c_p Constant pressure specific heat, J/kg.K
E Exergy, J
i Enthalpy, J/kg
$\dot{m}$ Mass flow, kg/s
Q Heat power, W
T Temperature, °C

Greek letters:
ε Effectiveness
ϵ Efficiency (2nd Law)
η Efficiency (1st Law)

Subscripts:
0 Ambient

1	Condenser outlet and pump inlet
2	Pump outlet and evaporator inlet
3	Evaporator outlet and expander inlet
4	Expander outlet and condenser inlet
atm	Atmospheric
c	Condenser
e	Evaporator
ex	Exergy
in	Inlet
out	Outlet
p	Pump
r	Recuperator
ref	Reference
s	Source
t	Turbine
th	Thermal
tot	Total
wf	Working fluid

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