Thermodynamic Analisys and Simulation of an Organic Rankine Cycle

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Abstract. *An organic Rankine cycle is a power cycle that uses a working fluid with high molecular mass. These fluids have the characteristic to boil at low temperatures, in moderate pressures. A power cycle using an organic fluid can harvest energy from low grade heat sources, as exhaustion gases, geothermal sources or waste heat recovery. To operate properly and extract the maximum work output, some parameters are fundamentals to design correctly the cycle components. One of these parameters is the pinch point, the minimum temperature difference between two fluids in a heat exchanger. This paper studies the sensibility of the cycle efficiency and the heat exchange area to the evaporator pinch point. A simulation model is constructed using heat and mass balance, and validated against a reference work. The sensibility study is conduced varying the pinch point from 10 K to 1K. The results show that decrease of the pinch point leads to less efficient cycles and larger evaporator areas.*

Keywords: *Organic Rankine Cycle, Pinch Point Analysis, Waste Heat Recovery, Low Grade Heat*

1. INTRODUCTION

Nowadays, the energy generation has to be clean, efficient and environmentally friendly. The growing restrictions in energy production to achieve these goals has created the demand to explore alternative energy sources. Among the sources available to harvest energy, there are those from low grade heat. In these sources, the heat is available but at a low temperature, or low exergy. Examples are the geothermal, solar, biomass and waste heat recovery from industrial processes. These heat sources temperature range are from 60ºC to 200ºC, according to Yamamoto *et al*., (2000).

The organic Rankine cycle (ORC) is a power cycle that uses a high molecular mass working fluid. The boiling point of these fluids are lower than the water, this characteristic is necessary to generate power from low temperature heat sources. Vankeirsbilck *et al*., (2011) compared a steam Rankine cycle against an ORC up to 3MW. They concluded that for low temperature heat sources, the ORC has advantage over the steam Rankine cycle, since it achieves higher efficiency. This may be explained by the fact the organic cycles do not need to be reheated during the expansion process, so less superheating is necessary.

To explore properly the ORC, it is necessary to design and operate the cycle components as near as possible to its optimal point. According to Quoilin *et al*., (2011), there is an optimal evaporation temperature that maximizes the efficiency. The evaporator *pinch point* is the minimum temperature difference between the working fluid and the heat source. It is a fundamental parameter to find the optimal point, since it affects directly the cycle efficiency and the heat exchange area.

The present work analyses an ORC and the impact of the *pinch point* on the cycle overall efficiency and evaporator heat exchanger area, fixing an optimal evaporation temperature.

2. THE ORGANIC RANKINE CYCLE

In an ORC, the main components are an evaporator, a turbine, a condenser and a pump. Figure 1 shows the processes that forms the cycle in a T-s and P-h diagrams. Basically there are four processes, 1-4 is the isobaric heating with evaporation, 4-5 is the turbine expansion, 5-8 is the condensation process and 8-1 is the pump compression. Also, in the Fig. 1 is plotted the deviation from ideal to real cycle due to irreversibilities.

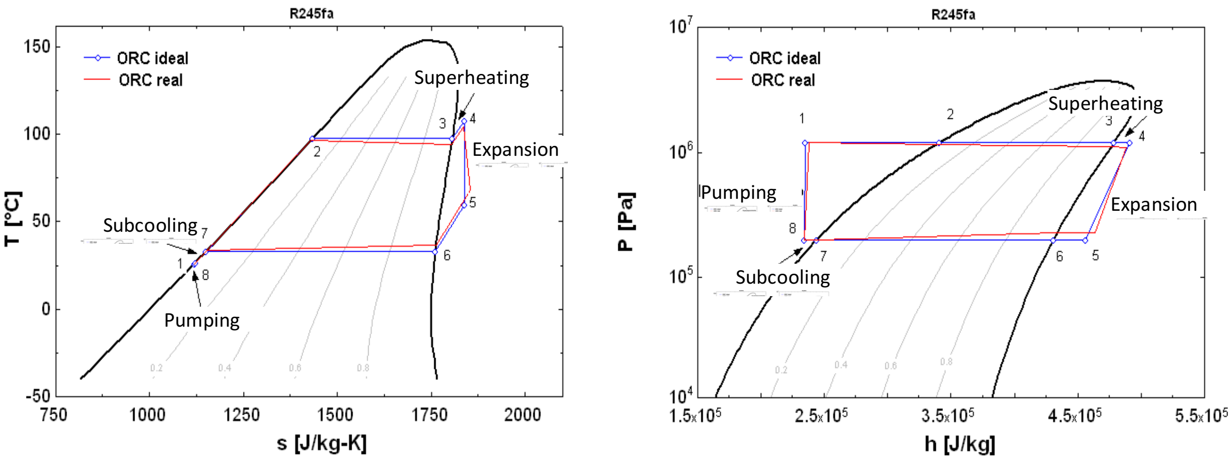

Figure 1. T-s and P-h diagrams for the real (red) and ideal (blue) ORC using R245fa (Quoilin *et al*., 2013)

Figure 2 shows a T-s diagram for water and organic fluids commonly used in ORC applications. The main characteristic of the organic fluids to be appreciated is the saturated vapor line, the half right line in the bell shape saturation region. For the water steam, as it expands in the turbine, the process goes directly inside the saturation region. Consequently, the expansion should be controlled (reheating) to avoid liquid in the last turbine stages. For the organic fluids, this does not happen due the inclination of the saturated vapor line. The expansion in the turbine ends in the superheated vapor zone. This characteristic avoids the reheating during the expansion.

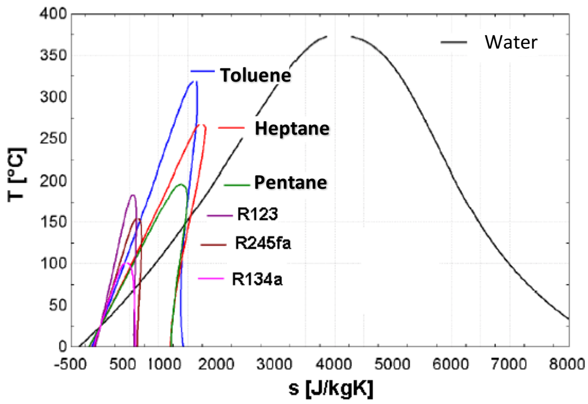

Figure 2. T-s diagram for water and organic fluids (Quoilin *et al*., 2011).

Figure 3 shows two ORC schemes with and without heat recovery, respectivelly. The superheat vapor in the end of expansion process can be used the pre-heat the condensate after the pump. This helps to reduce the heat rejected in the condenser and the heat for the vaporization in the evaporator. Mago et al., (2008), explain that this cycle configuration increases the efficiency. Quoilin *et al*., 2009 say that the use of the heat recovery in the cycle can increase the efficiency by 2% but reduces the net output power. Due this power reduction, it is not necessary for waste heat recovery applications.

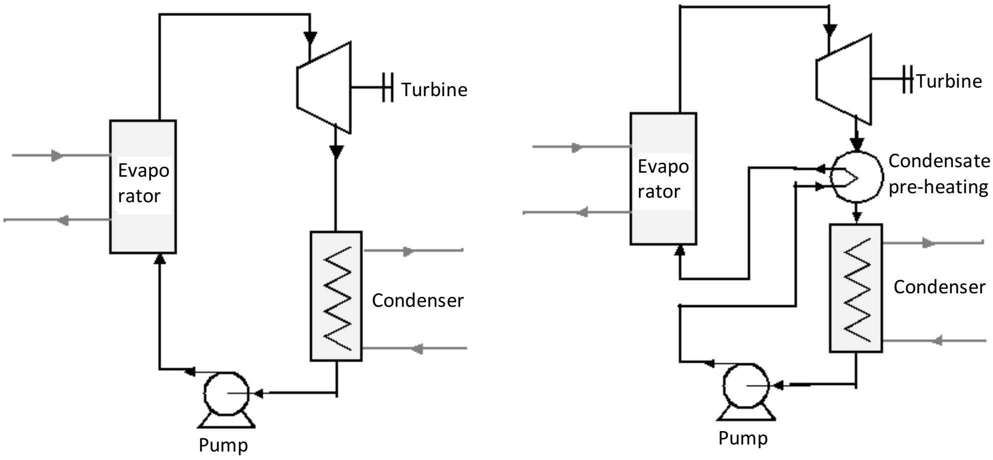

Figure 3. ORC scheme without heat recovery (left) and with heat recovey (right) (Quoilin *et al*., 2013).

2.1 Pinch Point

The work is proportional to the temperature difference between the evaporator and condenser. In an ORC (Quoilin, 2008) each heat exchanger can be divided in three zones according the fluid phase, liquid, biphase and vapor. The point where the temperature difference is minimum, is called *pinch point*. Figure 4 shows an example of the temperature profile in the heat exchangers, highlighting where the temperature difference is minimum between the vapor and heat source and liquid and cold source.

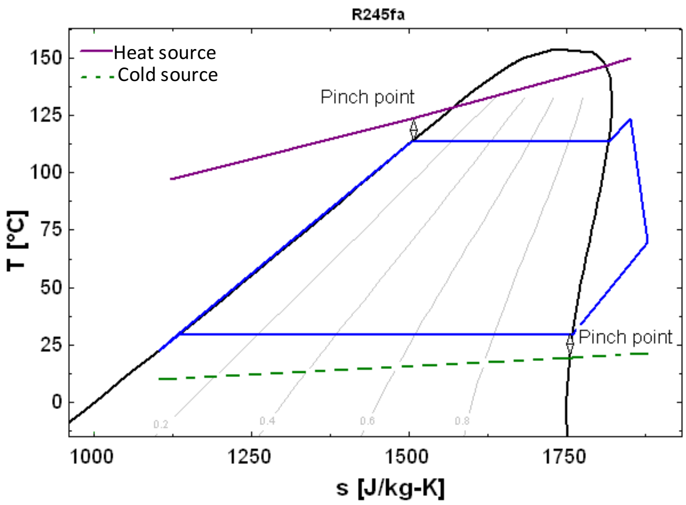

Figure 4. Pinch points of an ORC using R245fa

With a small *pinch point* is necessary a large heat exchanger to transfer the same amount of heat. This increases the pressure drop and consequently, decreases the cycle efficiency. A good practice is to use a pinch point in the range of 5 K to 10 K, according to the ORC application.

The cycle efficiency is expressed by the ratio between the heat exchanged in the evaporator $\dot{Q}_{ev}$ and the net output power $\dot{W}_{net}$,

$$\eta_{ORC} = \frac{\dot{Q}_{ev}}{\dot{W}_{net}} \tag{1}$$

To calculate the net output power all pumps should be considered, including the ones necessary to circulated the heat source fluid. A second way to express the cycle efficiency is multiplying the first law efficiency, Eq. (1), by the heat exchange performance of the evaporator,

$$\eta_{overall} = \varepsilon_{ev}\eta_{ORC} \tag{2}$$

The evaporator performance is the ratio between heat exchanged and the heat exchanged if the heat source fluid is cooled to the ambient temperature,

$$\varepsilon_{ev} = \frac{T_{in} - T_{out}}{T_{in} - T_{amb}} \tag{3}$$

3. MODELING

To model the ORC, heat and mass balances are done for each component, using ideal processes in the cycle. Figure 5 presents a block diagram of the system, the numbers corresponds to the points in Figure 1, the indexes *hf* and *cf* correspond to the heat and cold fluids, respectively. The heat exchangers are modeled according the mean logarithmic temperature difference methodology, when there is not phase change, and according to Stoecker, (2011), when there is phase change. The equations system is solved using the software Engineering Equation Solver - EES. The model is validated using the study of Quoilin *et al*., (2011), for the optimal evaporator temperature, as stated by the authors for the chosen fluid. To expand the analysis around this optimal point, the cycle efficiency and the heat exchanger area are investigated using the *pinch point* as a model parameter.

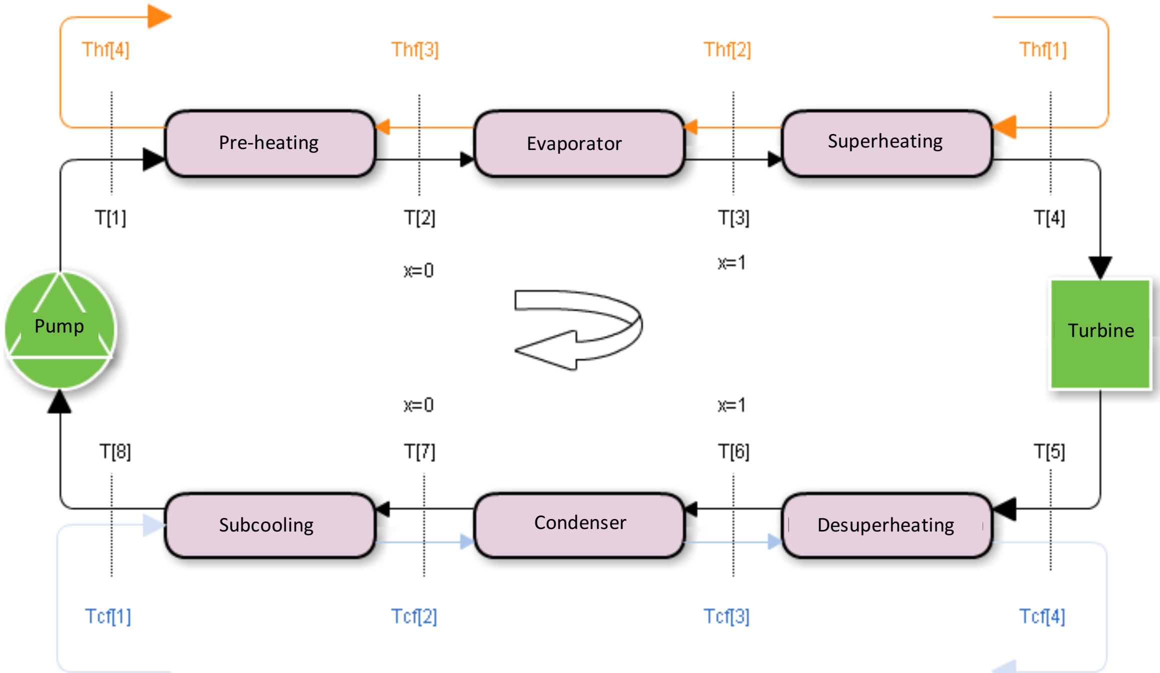

Figure 5. Block diagram for the cycle components as implemented in the software EES. T_{hf} is the heat source temperature, T_{cf} is the cold source temperature and T is the working fluid temperature, x is the vapor quality. The numbers are the reference state points.

Table 1 presents the parameters set in the simulation, they are the same of the of Quoilin *et al*., (2011). The heat source *pinch point* is the temperature difference between $T_{hf}[3]$ and T[2], and the cold source *pinch point* is the difference between T[6] and $T_{cf}[3]$. Both pinch points correspond to the beginning of the evaporation and condensing process, respectively.

Some considerations about the system modeled.

1. The pressure drop in the heat exchangers is proportional to the enthalpy variation.
2. The pressure drop in the liquid and vapor lines is not considered
3. The heat exchange from the heat source is done in three parts: pre-heating, evaporation and superheating. Each one with the proper global heat transfer coefficient and area, *UA*. The same for the condenser, with desuperheating, condensing and subcooling.

Table 1 Simulations parameters

Heat source	Fluid	Therminol VP-1
	Temperature	180ºC
	Mass flow	0,3 kg/s
	Pinch point	10 K → 1 K
	Inlet pressure	100 kPa
Cold source	Fluid	Water
	Mass flow	0,3 kg/s
	Temperature	15ºC
	Pinch point	10 K
	Inlet pressure	100 kPa
System	Fluid	R425fa
	Superheating temperature	5 K
	Subcooling temperature	5 K
	Evaporator pressure drop	10 kPa
	Condenser pressure drop	20 kPa
	Pump isentropic efficiency	60%
	Turbine isentropic efficiency	70%
	Evaporator temperature	113,5 ºC

4. RESULTS

4.1. Validation

The validation is done against the work of Quoilin *et al.*, (2011), for the same conditions. Table 2 presents the results obtained by the reference work and the results of the present work. The great difference is in the overall efficiency, 5,07%, what is sufficient to validate the proposed model.

Table 2. Comparison between the reference work (Quoilin et al. 2011) and actual work for the same simulation parameters.

	Reference work	**Present work**	**Variation [%]**
$\eta_{OVERALL}$ **[%]**	5,13	4,87	-5,07
η_{ORC} **[%]**	7,78	7,64	-1.80
Output power [W]	4764	4652	-2.35

4.2 Pinch Point Analysis

The pinch point between the heat source and working fluid is tested in the range from 10 K to 1 K, and all the others parameters are kept constant. The consequence of the *pinch point* variation is the change in heat exchanger area and cycle efficiency. As the global heat transfer coefficient U is kept constant, the relation UA expresses the area variation. Table 3 summarizes the results.

The results show that the UA increases for the pre-heating and for evaporation zones, while the UA of the superheating zone is not affect by the pinch point variation. For the pre-heating process, the variation rate is not linear, being 3,4%/K for a pinch point of 10K and 16,5%/K for a pinch point of 1K. For the phase change process, from 10 K to 5 K the variation rate is linear and around 10%/K, from 6 K to 1 K, the variation rate increases more dramatically until 24,2%/K. As the evaporation temperature and superheating temperature are set in the modeling, they are insensible to the *pinch point*.

The overall cycle efficiency decreases almost linear 3%/K with respect to the *pinch point*.

Table 3 Analysis result of the *pinch point* over the evaporator UA and overall cycle efficiency.

Pinch point [K]	$\eta_{OVERALL}$ [%]			$UA_{pre-heating}$			$UA_{evaporation}$			$UA_{super-heating}$		
	[%]	Relative Variation	Absolute Variation	W/K	Relative Variation	Absolute Variation	W/K	Relative Variation	Absolute Variation	W/K	Relative Variation	Absolute Variation
10	4,87	-	-	3,40	-	-	26,28	-	-	0,23	-	-
9	4,73	-2,9%	-2,9%	3,52	3,4%	3,4%	28,92	10,0%	10,0%	0,23	0,0%	0,0%
8	4,59	-3,0%	-5,7%	3,64	3,5%	7,1%	31,42	8,6%	19,6%	0,23	0,0%	0,0%
7	4,45	-3,1%	-8,6%	3,79	4,1%	11,5%	34,27	9,1%	30,4%	0,23	0,0%	0,0%
6	4,33	-2,7%	-11,1%	3,97	4,7%	16,8%	37,55	9,6%	42,9%	0,23	0,0%	0,0%
5	4,21	-2,8%	-13,6%	4,18	5,3%	22,9%	41,41	10,3%	57,6%	0,23	0,0%	0,0%
4	4,10	-2,6%	-15,8%	4,44	6,2%	30,6%	46,17	11,5%	75,7%	0,23	0,0%	0,0%
3	4,00	-2,4%	-17,9%	4,78	7,7%	40,6%	52,29	13,3%	99,0%	0,23	0,0%	0,0%
2	3,90	-2,5%	-19,9%	5,27	10,3%	55,0%	60,92	16,5%	131,8%	0,23	0,0%	0,0%
1	3,80	-2,6%	-22,0%	6,14	16,5%	80,6%	75,67	24,2%	187,9%	0,23	0,0%	0,0%

5 CONCLUSION

An organic Rankine cycle was modeled. The model was solved in the software Engineering Equation Solver – EES – and the results were validated against the work of Quoilin *et al*. (2011). The validation study showed an agreement of 5% for the overall cycle efficiency between the present work and the reference work.

It was studied the model sensibility to the *pinch point* parameter. From this analysis was concluded that:

1) The efficiency has a linear decrease with the decrease of the pinch point from 10 K to 1 K. The relative variation rate is 3%/K.
2) The heat exchanger area for the evaporator section increased, firstly linear for the pinch point range from 10 K to 5 K, with a variation rate of 10%/K, and secondly for the range form 5 K to 1K, going up to 24,2%/K.
3) The heat exchanger area for the pre-heating section increased in a non-linear way for the whole analyzed *pinch point* range.

As a general conclusion can be stated that the reduction of the *pinch point* makes the cycle linearly less efficient and increases the heat exchanger area.

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