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EVALUATION OF COMMERCIAL ORGANIC FLUIDS FOR WASTE HEAT RECOVERY

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Abstract. *Is not novel our strong fossil fuels dependence and the atmospheric pollution caused by burning them in the industrial systems. Therefore, the maximum utilization of energy obtained from this sources is very important. An option is the use of waste heat recovery systems. In this paper was presented the analysis of a simple organic Rankine Cycle (ORC) operating with five commercial refrigerants with low normal boiling point commonly used in refrigeration systems: R12, R22, R124, R134a and R152a with boiling points ranging from -40.81°C to -11.96°C, to convert waste energy to power from low-grade heat sources. These organic fluids were analyzed and compared, in order to determine with which of them the lowest components size and the best performance can be obtained. The evaluation was performed using a combined first and second law of thermodynamics by varying certain system operating parameters. The results showed that, the greater thermal efficiency by R124 was obtained. Additionally, this organic fluid has higher both boiling point and molecular weight. The devices sizes of the ORC are smaller for all the organic fluids in comparison with the water.*

Keywords: *Organic Rankine cycle, Refrigerants, Waste heat, Thermal efficiency*

1. INTRODUCTION

In the industry is very common find hot streams with temperatures lower that 370°C (Hung et al.,1997), for example, water steam and exhaust gases produced by burn of fossil fuel, which can be considered as a type of renewable and clean energy, since it is free energy and there is not direct carbon emission (Wang et al., 2011).

These hot streams, in the most cases are ejected to atmosphere no take advantage of its thermal energy. For this reason, new technologies that be environment friendly, capable of to use low-grade temperature energy and to reduce the dependence of fossil fuel need to be development.

The cycle with most use for power generation is the Rankine cycle, which use water as working fluid, but it needs high temperature for evaporate. That is why; the organic Rankine cycle (ORC) has attracted attention from scientific community (Marco, 2015).

The ORC use organic fluids instead of water as working fluid for the follow reasons (Marco, 2015):

- The organic fluids have a boiling point lower of water to elevated pressure allowing both high cycle efficiency and waste heat recovery.
- Possibility of selecting positive gauge condensing pressure, limiting the size of the low-pressure components (condenser, turbine discharge and low-pressure vapor piping) and avoiding air in-leakages.
- The use of organic fluids allows a configuration simple of cycle (no superheating and no reheating of steam, no multiple regenerative bleedings from the turbine).
- Reasonable volume flow rates and low enthalpy drops in the turbine. These points allow for a favorable turbine design, resulting in high isentropic efficiencies with a limited number of stages (even a single one), a reasonable size and hence competitive manufacturing costs.

A great number of scientific studies has been released on the adequate selection of the working fluid and on the optimization of the corresponding cycle parameters for a number of applications in waste heat recovery (Dai et al., 2009), biomass combustion (Drescher and Brüggemann, 2007), solar heat (Tchanche et al., 2009), geothermal sources (Invernizzi and Bombarda, 1997) and geothermal – solar hybrid concepts (Marco et al., 2011). The advanced cycle configurations such as supercritical cycles (Zhang and Jiang, 2012) and multi-level cycles (Walraven et al., 2013), the use of the mixtures (Chen et al., 2011) and predictive theoretical methods (Papadopoulos et al., 2010) to define the optimal working fluids are also being explored.

Since economic viewpoint, a simple organic Rankine cycle is the system more adequate for waste heat recovery, because of the few components used: evaporator, turbine, condenser and pump. Therefore, to found a working fluid that suits this cycle is very important for get transform the most amount of absorbed heat into power and thus increase the system thermal efficiency. However, the increase the restriction in the use of fluids with high GWP and ODP and safety issues (flammability and toxicity) make it important to investigate the performance of new environmental friendly and low risk fluids.

In this paper, a thermodynamic properties analysis of five working fluid with low normal boiling point, was carried out. The objective this paper is to find the best working fluid that used in a simple organic Rankine cycle under saturated operation conditions the higher system thermal efficiency, can be obtained.

2. METHOD AND MATERIALS

The organic working fluids have many different characteristics from water (Huijuan et al., 2010), as has been described above. The slope of the saturation curve of a working fluid in a T-s diagram can be vertical (e.g. R11, Fig. 1a), negative (e.g. R22, Fig. 1b) or positive (e.g. isopentane Fig. 1c), and the fluids are accordingly called "isentropic", "wet" and "dry" fluids. Wet fluids like water usually need to be superheated, while many organic fluids, which may be dry or isentropic, do not need superheating. Another advantage of organic working fluids is that the turbine build for ORCs typically requires only a single-stage expander, resulting in a simpler, more economical system in terms of capital costs and maintenance (Wendy and Thomas, 2005). The comparison of the temperature-entropy diagrams for dry, wet, and isentropic fluids are presented in Fig. 1, and the working fluids used in this study are present in the Table 2.

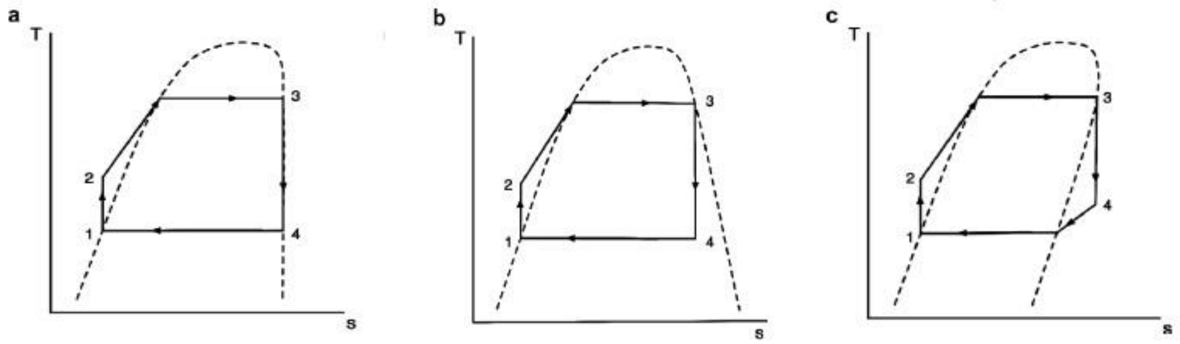


Figure 1. Comparison of the working fluids: a) isentropic, b) wet, c) dry (Pedro et al., 2008).

Table 1. Properties of the organic fluids used in this work

Organic fluid	R12 (Huijiuan et. al, 2010)	R22 (Huijiuan et. al, 2010)	R124 (Huijiuan et. al, 2010)	R134a (Alireza and Nenad, 2017)	R152a (Huijiuan et. al, 2010)
M (kg/kmol)	120.91	86.47	136.48	102.03	66.05
$T_{bp}(K)$ at 1 atm.	-29.75	-40.81	-11.96	-26.06	-24.02
P_{cr} (MPa)	4.14	4.99	3.62	4.06	4.52
T_{cr} (K)	385.15	369.35	395.45	374.25	386.45

2.1. Thermodynamic Analysis of the ORC

The equations used to determine the performance of a basic ORC configuration are presented in this section. Using the First and the Second Law of the Thermodynamics, the performance of an ORC can be evaluated under diverse working conditions for different organic working fluids. For the chosen configuration will be, assumes:

1. Steady state conditions,
2. No pressure drop in the evaporator, condenser, feed-water, and pipes,
3. Isentropic efficiencies for the turbine and pumps,

A schematic of basic ORC for converting waste heat into useful electrical power is shown in Fig. 2.

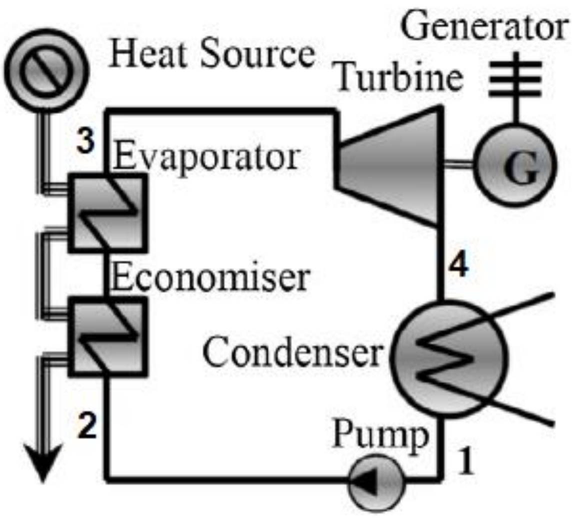


Figure 2. Flow diagram of a conventional Organic Rankine Cycle (Axel et al., 2017).

As observed in Fig. 2 there are four different process:

Process 1-2: the pump supplies the working fluid to the evaporator in compressed liquid state.

Process 2-3: the working fluid is heated and vaporized by the exhaust heat, so the compressed liquid pass to saturated vapor (high enthalpy)

Process 3-4: the generated high pressure vapor flows into the turbine and produced shaft power there.

Process 4-1: The low-pressure vapor is led to the condenser and condensed by cooling water. The condensed working fluid flows into the receiver and is pumped back to the evaporator, and a new cycle begins.

In the first step is chosen the thermodynamic state of each working fluid in saturated condition, i.e., $P_{sat} = P_3 = P_2$ and T_{sat}, for turbine inlet. The second step is choose condenser temperature in this case will be 10°C, the source cool temperature (T_L) is defined as the mean temperature between temperature inlet (T_{cw_in}) and outlet ($T_{cw_{out}}$) of cooling water, in this case assumed as 4°C and 8°C, respectively. It can be calculated by Equation (1).

$$T_L = \frac{T_{cw_{out}} - T_{cw\_in}}{\ln\left(\frac{T_{cw_{out}}}{T_{cw\ in}}\right)} \quad (1)$$

The third step is find the condenser pressure ($P_{cond} = P_1 = P_4$) assuming as saturated liquid condenser outlet ($x_1 = 0$) for the working fluid. Finally, we will find the enthalpies in each point of cycle: 1-2-3-4.

Choosing each component as a control volume, the first law of thermodynamics is applied to find the work out-put and the heat added or rejected. According to Yunus and Michael (2013), the energy balance equation can be expressed as:

$$\dot{Q} + \sum_i E_i = \dot{W} + \sum_o E_o \quad (2)$$

In addition, an isentropic efficiency of both the steam turbine and the pump can be expressed, respectability as:

$$\eta_t = \frac{\dot{W}_t/\dot{m}}{(\dot{W}_{t,ideal}/\dot{m})} \quad (3)$$

$$\eta_p = \frac{(\dot{W}_{p,ideal}/\dot{m})}{\dot{W}_p/\dot{m}} \quad (4)$$

2.2.1) Process 1-2 (pump):

Using Eq. (1) the pump power can be expressed as:

$$\dot{W}_p = \frac{\dot{W}_{P,ideal}}{\eta_p} = \frac{\dot{m}(h_1 - h_{2s})}{\eta_p} \quad (5)$$

where $\dot{W}_{p,ideal}$ is the ideal power of the pump, $\dot{m}$ is the working fluid mass flow rate, η_p is the isentropic efficiency of the pump, and h_1 and h_{2s} are the enthalpies of the working fluid at the inlet and outlet of the pump, respectively.

2.2.2) Process 2-3 (evaporator):

This is a constant-pressure transfer of heat. The evaporator heats the working fluid at the pump outlet to the turbine inlet condition. The heat transfer rate from the evaporator into the working fluid is given by:

$$\dot{Q}_e = \dot{m}(h_3 - h_{2s}) \tag{6}$$

where h_3 and h_{2s} are the enthalpies of the working fluid at the exit and inlet of the evaporator, respectively.

2.2.3) Process 3-4 (turbine):

The turbine power is given by:

$$\dot{W}_t = \dot{W}_{t,ideal}\eta_t = \dot{m}(h_3 - h_{4s})\eta_t \tag{7}$$

where $\dot{W}_{t,ideal}$ is the ideal power of the turbine, η_t is the turbine isentropic efficiency, and h_3 and h_{4s} are the enthalpies of the working fluid at the inlet and outlet of the turbine, respectively.

2.2.4) Process 4-1 (condenser):

The condenser heat rate can be expressed as:

$$\dot{Q}_c = \dot{m}(h_1 - h_{4s}) \tag{8}$$

where h_1 and h_{4s} are the enthalpies of the working fluid at the exit and inlet of the condenser, respectively.

2.2.5) System thermal efficiency

The system thermal efficiency is defined as the ratio between the net power of the cycle to the evaporator heat rate. It can be expressed as:

$$\eta_{th} = \frac{\dot{W}_t - \dot{W}_p}{\dot{Q}_e} \tag{9}$$

Substituting Equations (4), (5), and (6) into Equation (8) the system thermal efficiency for the organic Rankine cycle (ORC) can be written as:

$$\eta_{th} = \frac{(h_3 - h_{4s})\eta_t - (h_1 - h_{2s})\eta_p^{-1}}{(h_3 - h_{2s})} \tag{10}$$

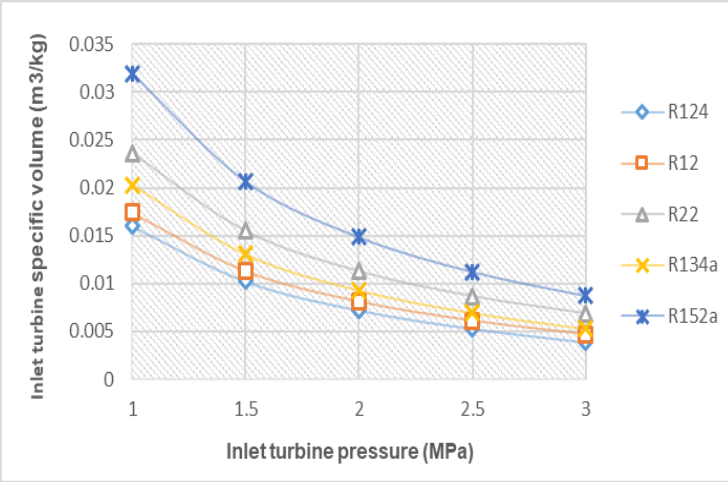
Observed that the system thermal efficiency does not depend on the mass flow of the working fluid.

3. RESULTS AND DISCUSSIONS

The performance of basic ORC system has been analyzed by using the appropriated thermodynamic properties for the various organic fluids. Energy losses due to irreversible process occurring in the cycle and heat transfer losses are ignored. For the purpose of this study five working fluids with boiling point ranging from -40,81°C to -11.96°C, were used: R12, R22, R124, R134a and R152a. In Fig. 3a present the specific volume, as a pressure function for the turbine inlet, of all the fluids organic in analysis for saturation conditions.

The working fluids with low specific volumes require smaller steam turbine due to volumetric flow as is the case of R124, R12, R134a, R22 and R152a, sequentially. Additionally, the specific volume decrease with the pressure increase.

a)



b)

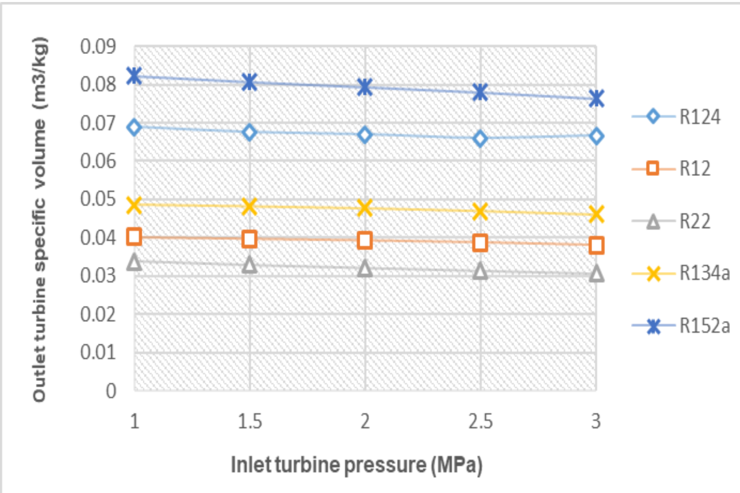


Figure 3. Variation of the specific volume with the inlet turbine pressure: $a)\ P_3\ vs\ v_3,\ b)\ P_3\ vs\ v_4$

In Fig. 3b present the specific volume as a pressure function for turbine outlet for saturation conditions. The working fluids with low specific volumes require smaller condensing equipment as is the case for the all organic fluids. However, the specific volume remains constant with the pressure increase. Generally, for the five organic fluids the size of condensation equipment will be smaller than one used for water.

In Fig. 4a is shown the variation of the temperature with the inlet turbine pressure. For all organic fluids, the saturation temperature increase with the inlet turbine pressure increase. The greater saturations temperatures are presented by R124, while that the smaller by R22 for the different pressures. The saturation temperature is a characteristic very important because can to limit the use of some refrigerants in specific applications. Therefore, the source temperature has that to be higher than saturation temperature of any organic fluid.

In Fig. 4b is shown the enthalpy drop with the inlet turbine pressure. For all organic fluids, the enthalpy drop increase with the inlet turbine pressure increase. The fluid that presents a greater enthalpy drop is R152a in comparison with the other organic fluids. However, it presents a less efficiency than R124 as shown in Figure 4. This is because, the pressure necessary to condensing R152a is greater than R124, hence, the power generated is smaller, consequently, its thermal efficiency also.

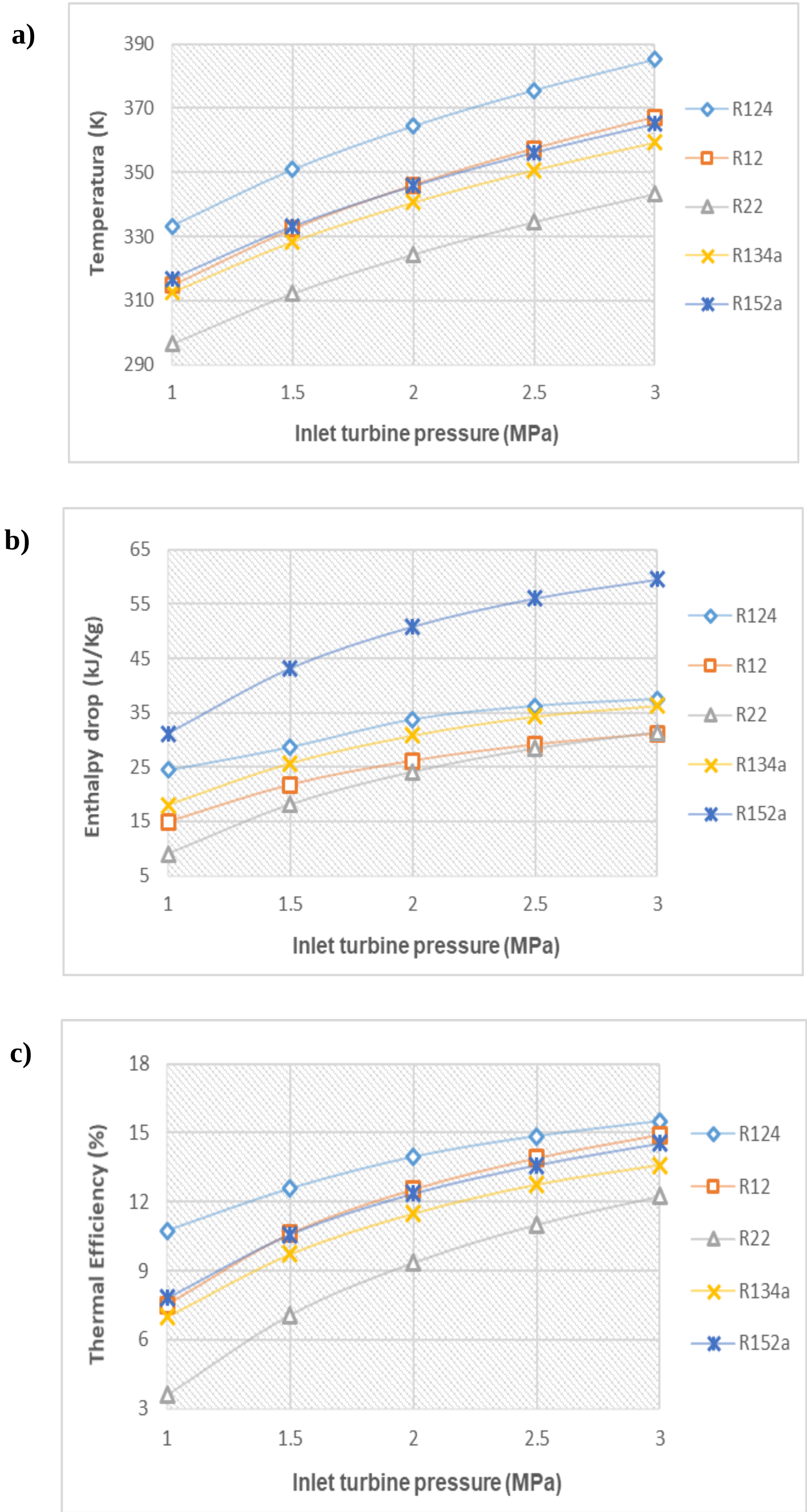


Figure 4. Evaluation of the major organic fluids parameters in an ORC: $a)\ P_3\ vs\ T_3,\ b)\ P_3\ vs\ (h_3 - h_4),$ $c)\ P_3\ vs\ \eta_{th}$

Finally, in Fig. 4c is shown that, the R124 has the greater thermal efficiency in the ranging from 10.74 to 15.5% with the pressure increase, while R22 present the worse thermal efficiency in ranging from 3.58 to 12.25% in the same conditions. Additionally, of the all organic fluids only R124 maintained its state of saturated vapor after the expansion process, different the other fluids that reached vapor qualities in the ranging from 0.9805 to 0.8809.

4. CONCLUSSIONS

A thermodynamics analysis of a simple organic Rankine cycle working with five different organic fluids: R124, R12, R22, R134a and R152a was presented. Due to behavior, shown in Figure 4c, of all organic fluids in saturated conditions, the R124 is considered dry fluid and the others wet fluids.

According to Figs. 3a and 3b, for all organic fluids the smaller devices both of the turbine and condenser, will be necessary, it in comparison with the water. Not always, a high enthalpy drop means a high thermal efficiency, as it was showed in Fig. 4b, it will depend on condensation temperature chosen.

Of the results showed in Fig. 4c, can be concluded that, with both higher molecular weight and boiling point temperature of the fluid, the higher system thermal efficiency will be obtained, as was the case of R124. Currently, the refrigerants R12 and R22 have been little considered because them detrimental to ozone layer, but in this work only they were used for make a comparison.

This research demonstrates that with organic Rankine cycle energy of way friendly with environment, can be obtained. However, is necessary a careful economic analysis for get a conclusion on viability of its application for power generation.

5. ACKNOWLEDGEMENTS

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