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INFLUENCE OF DRY FLUIDS ON PERFORMANCE OF ORGANIC RANKINE CYCLE FOR WASTE HEAT RECOVERY

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Abstract: *This paper presents an analysis of basic organic Rankine cycle (ORC) using five type dry fluids: R123, Iso-butane (R600a), R114, Rc318, Toluene with boiling points ranging from -11.7°C to 110.6°C, to convert waste energy to power from low-grade heat sources. These organic fluids have been analyzed and compared, in order to determine with which of them the best performance can be obtained. The evaluation was performed using a combined first and second law analysis by varying certain system operating parameters at saturation conditions. Results from these analysis shown that the higher system thermal efficiency was obtained with the working fluids toluene and R123 in comparison with the others working fluids. In addition, the fluid that shown the best thermal efficiency is the one that has higher boiling point and lowest molecular weight.*

Keywords: *Organic Rankine cycle, Dry fluids, Thermal efficiency, Waste heat*

1. INTRODUCTION

The invention of the electric power is the core impetus of the second industrial revolution, and the steam Rankine cycle driven by fossil fuels is still the dominant power supply method on large-scale. As is known to all, the accelerate consumption of fossil fuels has caused many serious environmental problems such as air pollution, global warming, ozone layer depletion and acid rain (Junjiang and Li, 2013).

Main components of a steam power cycle are condenser, pump, evaporator, turbine and working fluid. In this cycle, water used as working fluid has following thermodynamics characteristics (Bertrand et al., 2011):

- molecular weight: 18 kg/kmol
- boiling point: 373.15 K – 101.325 kPa
- freezing point: 273.15 K – 101.325 kPa
- triple point: 273.16 K – 0.611 kPa
- critical point: 647 K – 22.06 MPa
- latent heat: 2256.6 kJ/kg – 101.325 kPa
- specific heat: 4.18 kJ/kg K

Advantages presented by water as working fluid are (Wali, 1980):

- Very good thermal/stability (no risk of decomposition)
- very low viscosity (less pump work required)
- good energy carrier (high latent and specific heat)
- non-toxic, non-flammable, no threat to the environmental (zero ODP, zero GWP)
- cheap and abundant (presents almost everywhere on earth)

However, many problems are encountered when the water is used as working fluid (Wali, 1980):

- need of superheating to prevent condensation during expansion process
- risk of erosion of turbine blades
- excess pressure in the evaporator
- complex and expensive turbines

Because of the aforementioned reasons, water is more suitable for high temperature applications and large centralized systems (Bertrand et al., 2011).

The interest of the industrial sector in the energy conservation has encouraged the development of small power system that can use heat normally exhausted from various industrial process. Gaseous streams with temperatures in the range of 260 - 593°C and condensing streams at about 149°C are quite common in a number of industries. These heat streams are not only generated by industrial plants but also by small utilities that use gas turbines or diesel engines as their primary source of power. The greater part of energy, however, is lost at temperatures less than 370℃, the temperature at which Rankine cycle power systems using water as working fluid become significantly less efficient and capital cost increase (Marciniak et al., 1897).

These waste heat is generally classified according to the temperature, such as, low-grade waste heat for stream temperature below 230℃; medium-grade waste heat between 230 and 650°C; high-grade waste heat for 650℃ or higher. A large portion of the heat created expensively is finally released to the environment due to lack of effective methods for recovery. It is reported that industrial low-grade waste heat accounts for nearly 50% or more of total heat input (Roy et al., 2011)

Heat exhaustion causes considerable hardware costs and operating losses, such as fuel, electricity, water, etc., and also reinforces emissions of greenhouse gases (GHGs). Despite various efforts to maximize the efficiency of heat utilization in industrial process, an effective method to recover the waste heat from low to medium temperature sources and transform it into useful mechanical work or electricity is an attractive alternative to reduce the total fuel consumptions and also the thermal pollution (Cheng-Liang et al., 2016). In this case, the organic compounds characterized by higher molecular mass and lower ebullition/critical temperature than water have been proposed in so called “Organic Rankine Cycles” (Bertrand et al., 2011). Therefore, the Organic Rankine Cycle (ORC) is one of the feasible technologies to recover waste heat from low to medium temperature heat sources and to generate electricity (Cheng-Liang. et al., 2016). The ORC has the characteristics of simple structure, high reliability; easy maintenance and the same system configurations as steam Rankine cycle (Junjiang and Li, 2013).

The working fluid of an ORC determines thermal efficiency, safety, stability, environmental impact, and economic profitability of the system. In recent years, working fluids selection for ORC has drawn significant attention. Different performance evaluation criteria lead to different optimum working fluids. Therefore, a reasonable evaluation criterion is the key issue for working fluid selection (Haoshui et al., 2016).

Bertrand et al. (2011), pointed some differences between the water and organic fluids, as for example that condenser pressure acceptable in organic fluids while for water is low, it which would allow infiltrations. For other hand, the water is free available for use while for the organic fluid exist supply problem. Finally, the installation cost for vapor cycle is lowest than for organic Rankine cycle, but its use is justified by the heat source type.

Due to big benefit that represent the organic Rankine cycle in the heat recovery of low and medium temperature sources, many studies was dedicated for select the best working fluid with the objective of enhanced the performance of the power cycle. Maizza and Maizza (1996), Sanjai and Yogi (2005), are some of the researchers who have reported performances and characteristics of different working fluids used for waste heat recovery systems.

Lee et al. (1993), pointed out that the system efficiency of an ORC correlates with the fluid´s normal boiling point, critical pressure and molecular weight. Hung et al. (1997), Hung (2001), Liu et al. (2004) and Santiago et al. (2017), studied some dry fluids for organic Rankine cycle in waste heat recovery and the efficiencies of ORC using cryogens such as benzene, ammonia, R11, R12, R134a and R113, as working fluids. Chandramohan et al. (2006), presented an analysis of the performance of ORC using R113 an R134a in which it was shown that organic fluids can be used to generate power using low-temperature waste heat.

Negoc et al. (2011), conducted the working fluid selection research for high temperature ORC based on thermal efficiency. Haoshui et al. (2015), proposed a new method for simultaneous selection of working fluid and operating conditions for an ORC recovering waste heat with maximum power output as the working fluid selection criterion. Bertrand et al. (2011) and Fredy et al. (2012) carried out a review of various application of organic Rankine cycle having as heat sources: solar energy geothermal energy, biomass products, surface seawater and waste heat from various thermal process.

In this work will be analyzed and will explained the influence of working fluids in the thermal efficiency of the Organic Rankine cycle (ORC) using dry fluids in saturation condition for recovery low-grade waste heat. Superheat conditions were not included in this study inasmuch, dry fluids then of pass through the turbine (expansion process) its thermodynamic state falls in the superheat zone, which avoid risks erosion in the turbine blade, guarantying thus, the safe operation. Different procedures for the calculated can be found in Michael and Howard (2003), Yunus and Michael (2013) among others.

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.

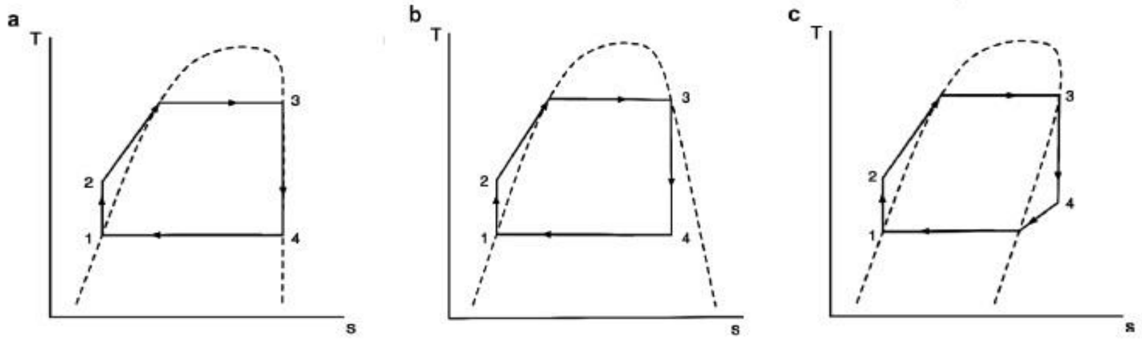

Figure 1. Comparison of the working fluids: a) isentropic, b) wet, c) dry (Pedro et al., 2008).

Table 1. Properties of the dry fluids used in this investigation

Organic fluid	R123 (Huijuan et. al, 2010)	R600a (Huijuan et. al, 2010)	R114 (Alireza and Nenad, 2017)	Rc318 (Huijuan et. al, 2010)	$Toluene$ (Huijuan et. al, 2010)	Water (Hung et. al, 1997)
M (kg/kmol)	152.93	58.12	170.92	200.03	92.14	18
T_{bp} (K) at 1 atm.	300.98	261.45	276.85	267.35	383.75	373
P_{cr} (MPa)	3.66	3.63	3.25	2.78	4.13	22.06
T_{cr} (K)	456.83	407.81	418.83	388.38	591.75	647

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.

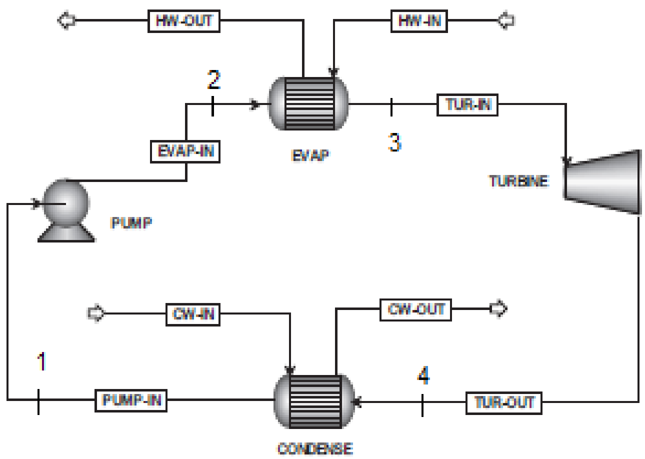

Figure 2. Flow diagram of a conventional Organic Rankine Cycle (K. Satanphol et al., 2017).

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

Process 1-2: the pump supplies the working fluid to the evaporator.

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 power there.

Process 4-1: The low-pressure vapor is led to the condenser and condensed by cool 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, which is defined as the mean temperature between temperature inlet and outlet of cooling water, in this case assumed as 15°C and 25°C, respectively. It can be calculated as $T_L = (T_{cw\_in} + T_{cw\_out})/2$ as done by Donghong et al. (2007). 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:

$$\dot{Q} + \sum_i E_i = \dot{W} + \sum_o E_o \tag{1}$$

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

$$\eta_t = \frac{W_t/\dot{m}}{(W_{t,ideal}/\dot{m})} \tag{2}$$

$$\eta_p = \frac{(W_{p,ideal}/\dot{m})}{W_p/\dot{m}} \tag{3}$$

2.2.1) Process 1-2 (pump):

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

$$\dot{W}_p = \frac{\dot{W}_{P,ideal}}{\eta_p} = \frac{\dot{m}(h_1 - h_{2s})}{\eta_p} \tag{4}$$

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 (isentropic expansion) of the pump.

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:

$$\dot{Q}_e = \dot{m}(h_3 - h_{2s}) \tag{5}$$

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:

$$\dot{W}_t = \dot{W}_{t,ideal}\eta_t = \dot{m}(h_3 - h_{4s})\eta_t \tag{6}$$

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 (isentropic expansion) of the turbine.

2.2.4) Process 4-1 (condenser):

The condenser heat rate can be expressed as:

$$\dot{Q}_c = \dot{m}(h_1 - h_{4s}) \tag{7}$$

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:

$$\eta_{th} = \frac{W_t - W_p}{Q_e} \quad (8)$$

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

$$\eta_{th} = \frac{(h_3 - h_{4s})\eta_t - (h_1 - h_{2s})\eta_p^{-1}}{(h_3 - h_{2s})} \quad (9)$$

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 -11,7°C to 110,6°C, were used: R123, R600a (Iso-butane), R114, Rc318 and toluene, the water was used for comparison in the calculate of thermal efficiency.

Figure 3 present the specific volume, for the turbine inlet, as a function of pressure, of all the fluids organic in analysis for saturation conditions. The working fluids with low specific volumes require smaller condensing equipment as is the case of R123, R114, Rc318, R600a and toluene in comparison with water. Additionally, the specific volume decrease with the pressure increase.

Figure 4 shows that the Toluene has the higher enthalpy drop, after the water, and increase with increase of inlet turbine pressure. Additionally, Figure 4 shown that the others working fluids increase the enthalpy drop with the increase of inlet turbine pressure, too. However, the Rc318 presented the lowest enthalpy drop of all working fluids.

Figure 5 shown that the saturation temperature of all working fluids increase with pressure increase. The toluene presents the higher saturation temperatures and both organic fluids Rc318 and R600a present the lowest saturation temperatures. Its indicate that for temperatures of waste heat in the ranging from 489.93 K to 566.265 K is more convenient to use toluene and for temperature in the ranging from 313.47 K to 382.95 K is more convenient to use Rc318 as working fluid. All fluids used in this work are dry fluids except the water considerate as wet fluid, so its use would be limited due that after expansion process are reached qualities in the ranging from 0.7037 to 0.7605 in saturated conditions which would be detrimental for the turbine operation.

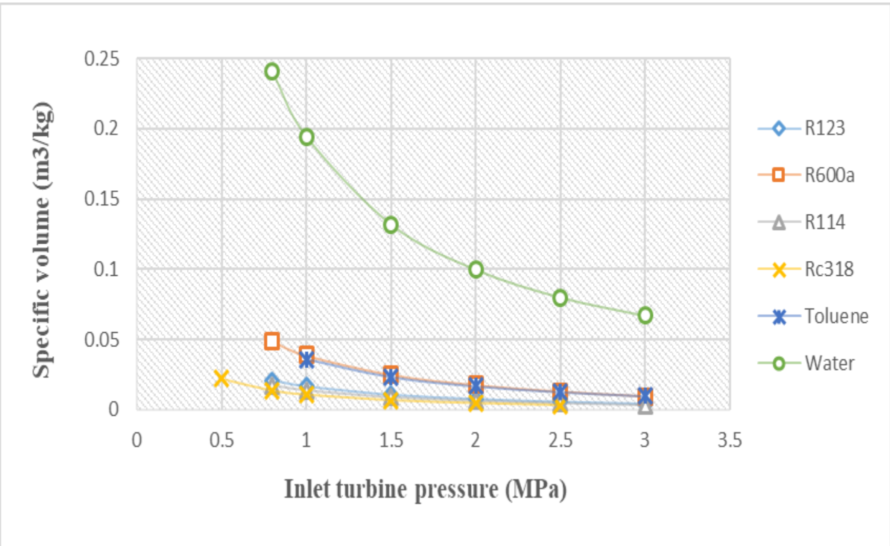

Figure 3. Variation of the specific volume with the inlet turbine pressure.

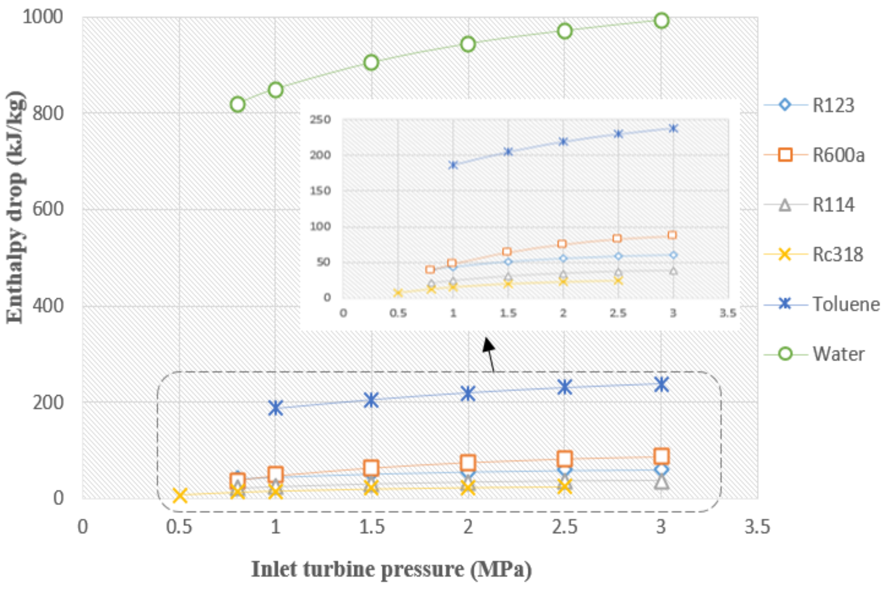

Figure 4. Variation of the enthalpy drop with the inlet turbine pressure for all working fluids

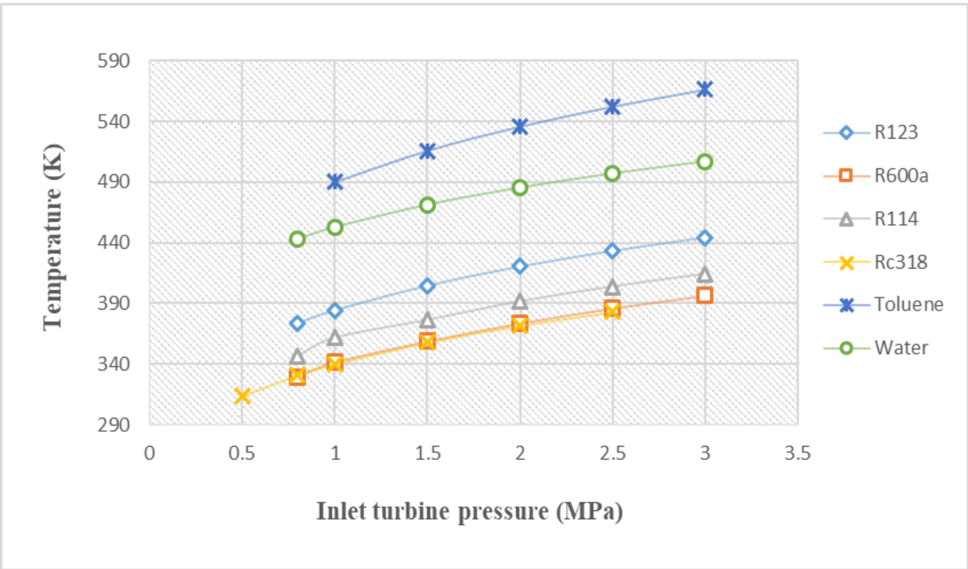

Figure 5. Variation of the temperature with the inlet turbine pressure in saturated conditions.

Finally, Figure 6 shown the variation of the system thermal efficiency with the turbine inlet pressure and its temperature corresponding at saturated conditions for the basic organic Rankine cycle. For this configuration, the condenser temperature was kept constant at 293,15K (20°C). The isentropic efficiencies of the turbine and the pump were considered 80% and 85%, respectively. Additionally, Figure 6 demonstrates that the system thermal efficiency increase with the increment of the turbine inlet pressure. This figure illustrates that toluene and R123 have the best performance among the working fluids for saturation temperature range from 489.93 K (216.78°C) to 566.265 K (293.115°C) and 373.95 K (100.8°C) to 444.45 K (171.3°C), respectively. On the other hand, the fluid with the worst thermal efficiency is Rc318, with saturation temperature in the range from 313.47 K to 382.95 K. Naturally, the temperature and pressure are dependents in saturation conditions.

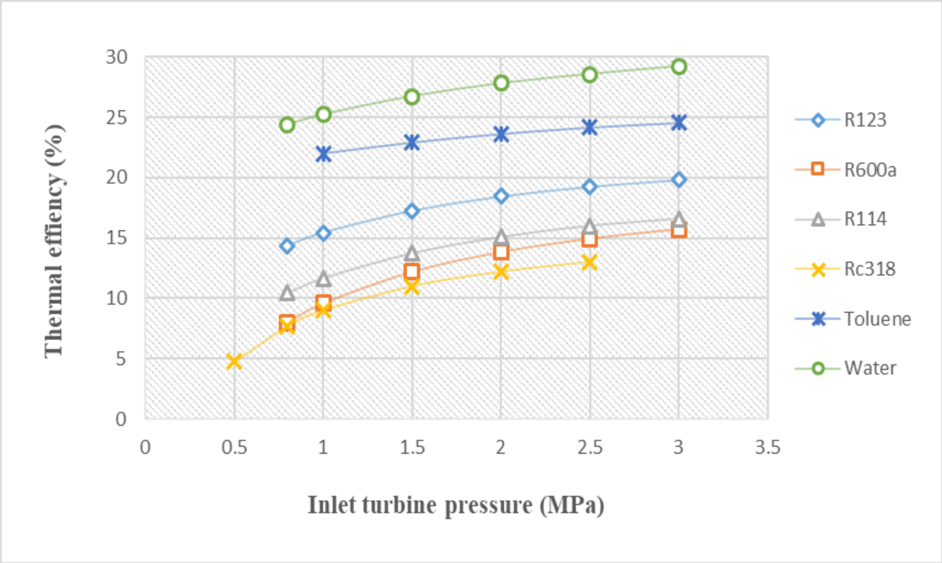

Figure 6. Variation of the thermal efficiency with the inlet turbine pressure in saturated conditions for all working fluids.

Figure 6 shows the best cases are representing by toluene in which the system thermal efficiency increase of 21.98 – 24.55% for the lowest and highest pressure, respectively, and using R123 the system thermal efficiency increase of 14.35 – 19.82% for 0.8 MPa and 3 MPa, respectively. On the other hand, the worst case is representing by Rc318, which shown an increase of 4.75 – 13% for 0.5 MPa and 3 MPa, respectively. These results clearly demonstrate that using toluene or R123 is obtained the better system thermal efficiency. The use of either will depend of source temperature.

Therefore, it can be concluded that, with the lowest molecular weight and higher boiling point temperature of the fluid, the higher system thermal efficiency will be obtained, for all the organic fluids chosen. One exception is toluene that compared with water has higher boiling point but produces a smaller system thermal efficiency, possibly due to its greater molecular weight. However, in the practice is not recommendable to use the water in saturated conditions due its smaller vapor quality.

4. CONCLUSIONS

The analysis of an Organic Rankine Cycle (ORC) operating with five dry organic fluids, was realized. This study shown that, the high molecular weight and the lower normal boiling point contribute with the worsening of the cycle efficiency.

The specific volume higher (or lowest density) correspond to the organic fluid with lower molecular weight as the case R600a (Iso-butane). In contrast, the specific volume lowest (or higher density) correspond to the organic fluid with higher molecular weight as the case Rc318. Additionally, the specific volume of all working fluids decrease with pressure increase.

For the working fluids with higher molecular weight (R123<R114<Rc318) the drop enthalpy value is constant for pressure above of 2 MPa, and for the other working fluids (R600a and toluene) the drop enthalpy increase with pressure increase.

The toluene considered an alkane has the higher enthalpy drop and the higher thermal efficiency. In contrast, the refrigerant Rc318 has the lowest enthalpy drop and the lowest thermal efficiency.

The dry fluids presented relatively elevated temperature after of expansion process and its thermodynamics state is localized in the superheated zone. It means that, this energy can be usable. Also means that, the dry fluids not representing detrimental operation for the turbine.

5. ACKNOWLEDGMENT

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7. RESPONSABILIDADE AUTORAL

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