THERMODYNAMIC ANALYSIS OF AN ORGANIC RANKINE CYCLE DRIVEN BY WASTE HEAT RECOVERY

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Abstract. *The organic Rankine cycle (ORC) is a suitable form of generating electricity, mainly when recovering low grade waste heat. For most industrial applications, it is important to know how much power can be gained by recovering the rejected waste heat. The purpose of this study is to determine the power output of an ORC while varying the flow and temperature of the waste heat. A simulation model of the cycle is presented and the system performance was optimized using* R245fa, R1234yf *and* R1234ze *as working fluids. The results show that, by means of higher mass flow rates and inlet temperatures of the waste heat, the system net power output and thermal efficiency are improved.*

Keywords*: organic Rankine cycle (ORC), low grade waste heat, simulation*

1. INTRODUCTION

In many industries waste heat, on gaseous or liquid form, is usually rejected during a process due to its low temperature. The use of conventional CHP (combined heat and power) methods to recover energy from these heat sources is not economically feasible. This wasted heat, however, can be used to generate power on a Rankine cycle working with an organic fluid instead of H_2O, since the evaporating temperature of such organic fluids is much lower than that of water.

The most important concern in the study of an organic Rankine cycle (ORC) is the improvement of the overall cycle efficiency, since the heat input temperature is generally low. The efficient operation of the cycle depends mostly on two factors: the working conditions (namely, the pressure levels) and the thermodynamic properties of the working fluids.

In the literature, experimental research on organic Rankine cycles has been widely explored, and authors have proposed different arrangements for the system depending on design, heat sources and applications.

The vast majority of prototypes were designed to generate electric power; with the exception of those used for reverse osmosis desalination (Monolakos *et al.* 2009) and to power vapor compression refrigeration cycles (Wang *et al.* 2011).

In terms of the heat source that was used in the experimental evaluation, some authors used hot air (Lemort *et al.* 2009 ; Quoilin *et al.* 2010) to simulate waste heat gases, while others used hot water (Nguyen *et al.* 2001; Yamada *et al.* 2011) and hot oil (Wang *et al.* 2011). A few researchers started using a prototype in laboratory conditions, and then, tested it with real industrial waste heat, such as exhaust and combustion gases (Larjola, 1995). Solar panels have also been used to provide the heat source to the system (Monolakos *et al.* 2009; Zhang *et al*. 2007).

Many studies have been conducted for the proper selection of a working fluid to recover heat from a particular source. Analytical studies that seek a better performance of the cycle were carried out comparing different working fluids.

Some authors examined the thermodynamic and physical properties of up to twenty (Maizza and Maizza 2001) and thirty working fluids (Saleh *et al.* 2007; Chen *et al*. 2010). The parametric optimization of the cycle has also been carried out in order to compare different working fluids (Roy *et al.* 2010; Dai *et al*. 2009; Wei *et al.* 2007). Several studies focused on the optimization of the cycle and fluid selection from specific heat sources, such as: solid biomass power plants (Drescher and Brüggemann 2007), combined cycle power plants (Chacartegui *et al.* 2009), micro-gas turbines (Invernizzi *et al*. 2007), internal combustion engines (Bombarda *et al.* 2010; Vaja and Gambarotta 2010; Srinivasan *et al.* 2010), and solar panels (Delgado-Torres and Garcia-Rodriguez 2010; Rayegan and Tao 2011).

Other investigations developed simulation models of the cycle to be used for: design control and diagnostic systems (Wei *et al.* 2008), optimization of the cycle (Hettiarachchi *et al*. 2007; Bruno *et al.* 2008; Nguyen *et al*. 2010; Jing *et al.* 2010), fluid selection (Tchanche *et al*. 2009; Papadopoulos *et al.* 2010; Heberle and Brüggemann, 2010), and viability of a system design (Schuster *et al.* 2009; Gang *et al*. 2010).

Based on the saturation curve of the temperature-entropy (*T-s*) diagram, organic fluids can be classified as dry, wet, and isentropic. As it is presented on Fig. 1, a dry fluid has a positive slope (*dT/ds*), a wet fluid has a negative slope, and an isentropic fluid has an infinitely large slope.

In an ORC, the working fluid expands isentropically through the turbine from state (3) to state (4s). The negative slope of the saturation vapor curve for wet fluids indicates the presence of condensate while passing through the turbine. Since the impact of liquid droplets in the turbine blades is undesirable, only dry and isentropic fluids are recommended to be used in ORCs.

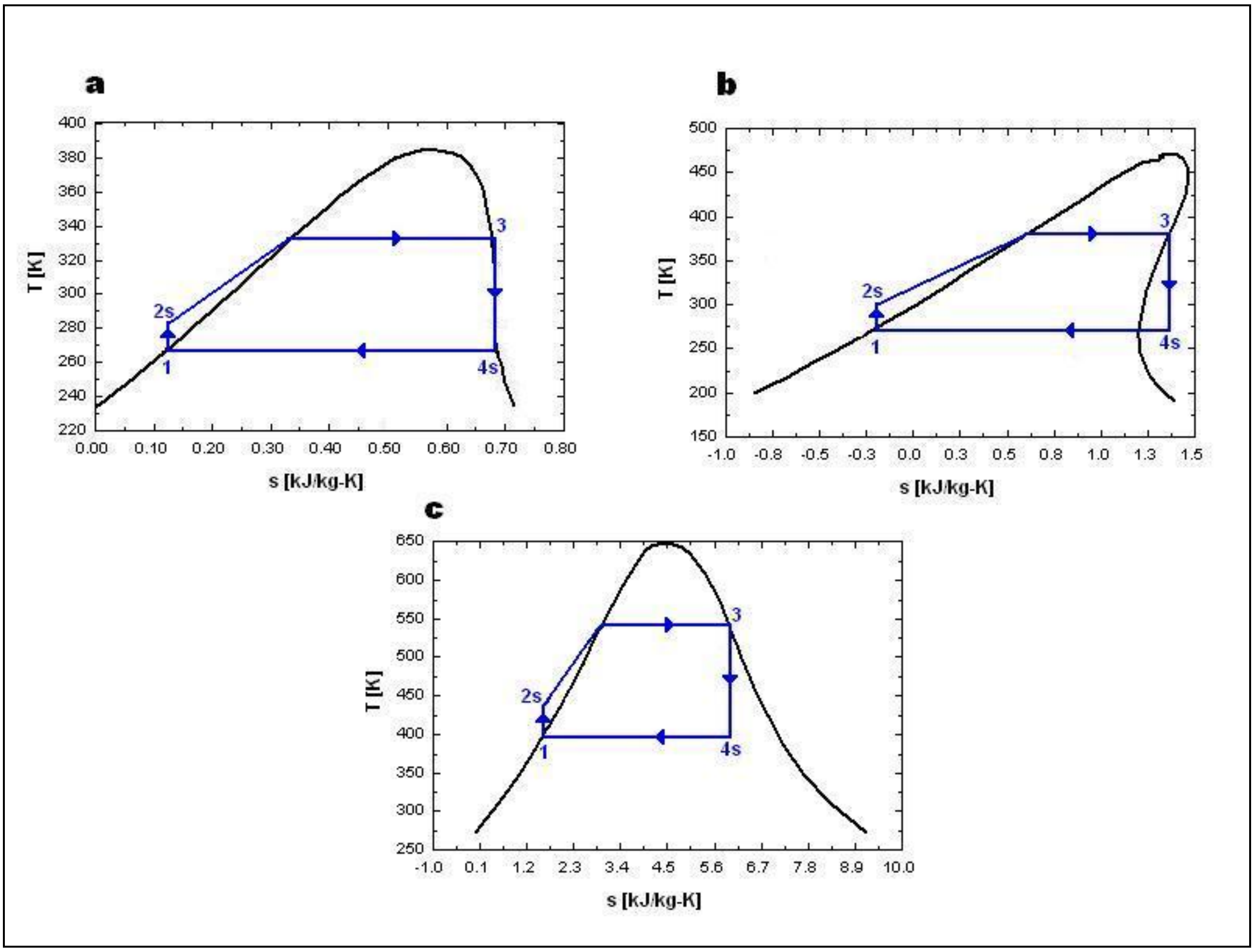

Figure 1. *T-s* process diagram comparison of working fluids: (a) isentropic, (b) dry, and (c) wet.

Comparisons between dry, wet and isentropic fluids have been carried out in order to identify which type yields higher system efficiency (Hung *et al.* 2010; Liu *et al.* 2004; Desai and Bandyopadhyay, 2009), all concluding that dry fluids are the most favored for an ORC.

Other studies have focused just on dry fluids, either to select the fluid with better performance (Hung, 2001) or to evaluate the efficiency between the basic and the regenerative cycle (Mago *et al.* 2008).

As it has been previously established, many of the researchers have conducted studies on the selection of the proper working fluid, that is, the one that showed best performance under the same operating parameters. Other authors have concentrated their investigations on optimizing the performance of the cycle with different working fluids. However, even when the cycle's performance was evaluated, the operating parameters, such as the heat source flow and temperature, were maintained constant during these simulations.

To the authors' knowledge, there has been little or no analysis on the cycle's performance under a broad range of operating parameters.

It is important to evaluate the performance of the cycle while varying the operating parameters in order to obtain the optimum power output under different working conditions. The objective of this study is to determine the power output and thermal efficiency of an organic Rankine cycle for various flows and properties of the waste heat source using three different dry working fluids.

2. THERMODYNAMIC MODEL OF AN ORGANIC RANKINE CYCLE

The traditional ORC system consists of a pump, boiler, turbine and condenser. However, in order to best describe the changes of thermodynamic states in the working fluid, the boiler has been replaced by three heat exchangers, namely: economizer, evaporator and superheater.

Figure 2 shows a schematic diagram of waste heat recovery for power production through an organic Rankine cycle. Liquid from the condenser is pumped from 1 to 2 and enters the economizer, at high pressure, followed by the evaporator and the superheater. High pressure superheated vapor leaves the superheater to enter the turbine or expander, where it expands to the lower pressure of the cycle, to enter the condenser. The condensate then flows to the pump, thus closing the cycle.

Heat is transferred from the waste heat fluid, in this case assumed consisting of exhaust gases, to the working fluid by means of the three exchangers.

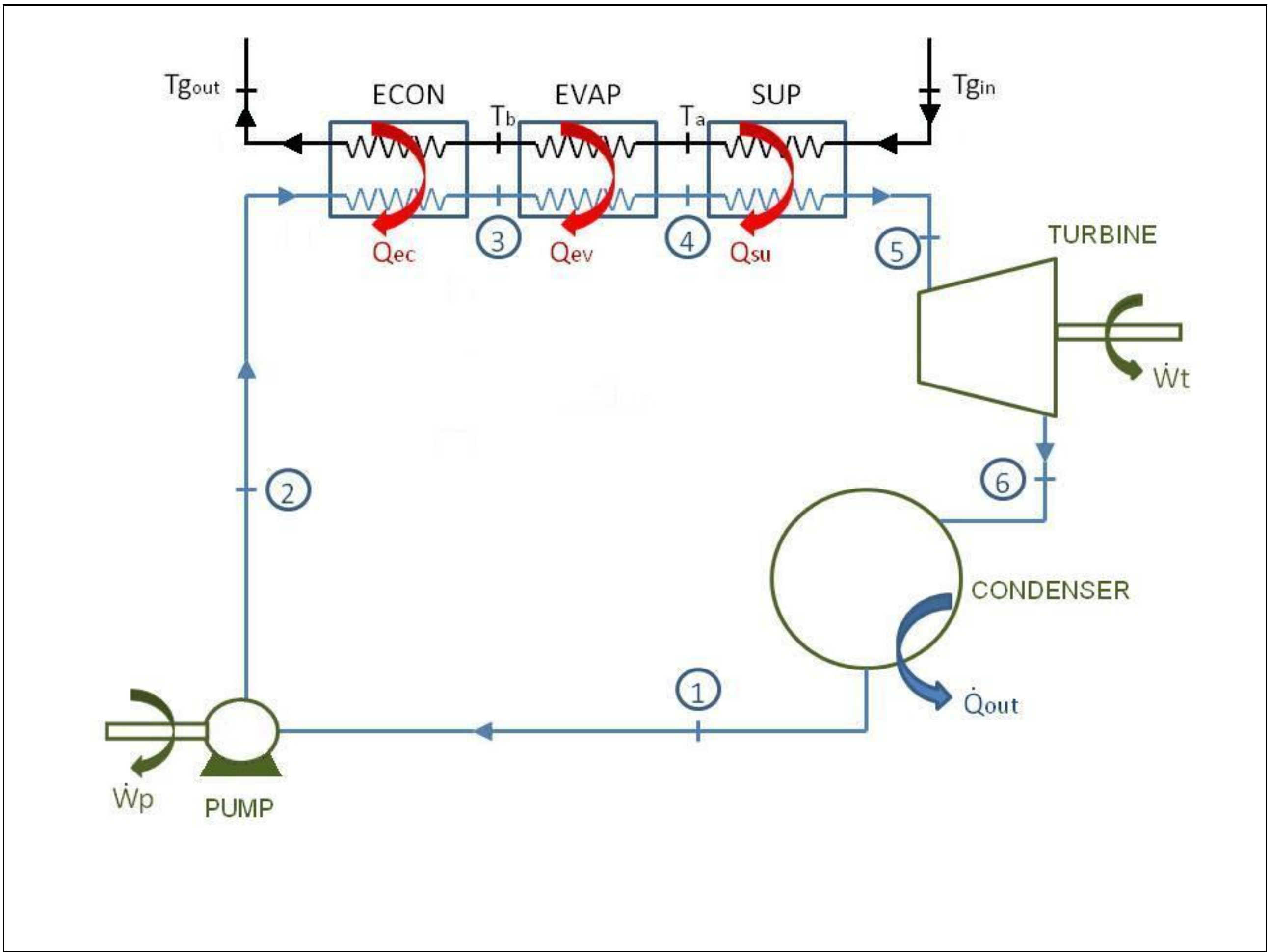

Figure 2. Diagram of waste heat recovery for power production through an organic Rankine cycle.

It is considered that only certain data from the cycle are known in advance, and that the rest of the waste heat fluid and working fluid thermodynamic states will depend on the operating conditions of the cycle.

The ambient temperature determines the cycle lower pressure, since the condenser will work at a pressure close to the corresponding saturation pressure relative to ambient temperature. At the turbine inlet, superheated steam conditions are assumed and, therefore, a value of the required degree of superheat is provided as an input.

As it was stated before, the boiler has been substituted by three heat exchangers, specifically the economizer, the evaporator and the superheater. From Fig. 2 we can see that there are three different heat input rates ($\dot{Q}_{ec}, \dot{Q}_{ev}, \dot{Q}_{sh}$) and only one heat output ($\dot{Q}_{out}$) at the condenser. The total heat addition rate to the cycle ($\dot{Q}_{in}$) is the sum of the heat input rate at each of the heat exchangers ($\dot{Q}_{in} = \dot{Q}_{ec} + \dot{Q}_{ev} + \dot{Q}_{sh}$).

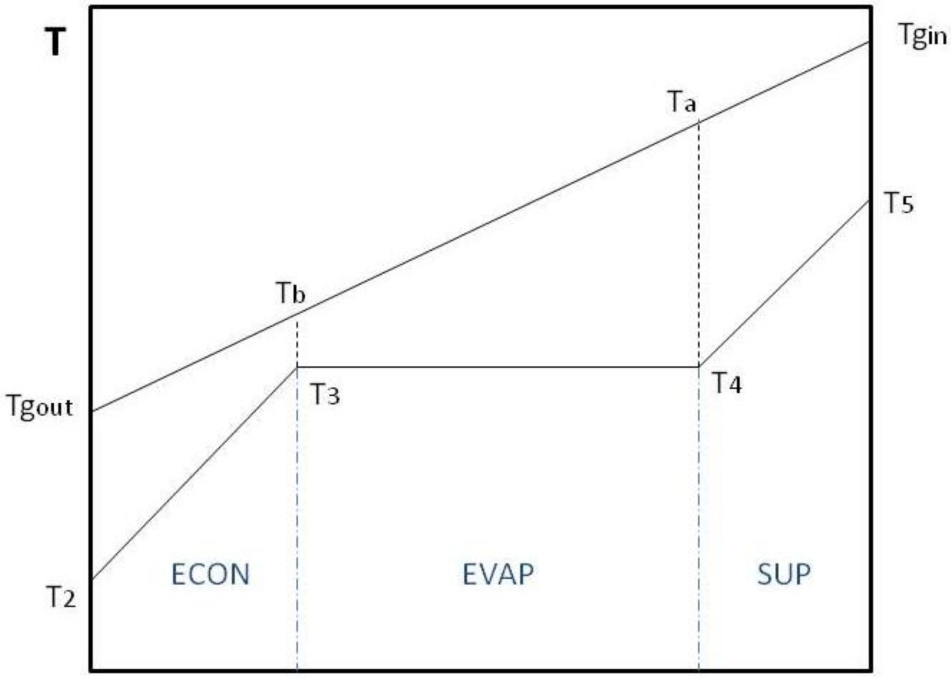

Figure 3. Temperature Diagram at the heat exchangers

The temperature variation in the cycle for the waste heat fluid as well as for the working fluid is shown in Fig.3.

The difference between the saturation temperature of the working fluid and the temperature of the waste heat fluid in the evaporator is called pinch point temperature difference.

The pinch point temperature difference can be obtained by subtracting T_3 (working fluid temperature after the economizer) to T_b (intermediate temperature of the heating fluid).

2.1. Mathematical Model

2.1.1 Process 1-2 (Pump)

The compression of the working fluid in the pump is assumed to be isentropic. The work done by the pump is:

$$W_p = \frac{\dot{m}_{wf}(h_2 - h_1)}{\eta_p} \quad (1)$$

where $\dot{m}_{wf}$ (kg/s) is the mass flow rate of the working fluid, h_1 and h_2 (kJ/kg) are the specific enthalpies of states 1 and 2, and η_p is the efficiency of the pump.

2.1.2 Process 2-3 (Economizer)

The heat transfer process in the economizer can be better understood as a control volume, represented in Fig.4. The working fluid enters the economizer at a temperature T_2 and leaves it at T_3, and the waste heat fluid enters at T_b and leaves it at Tg_{out}.

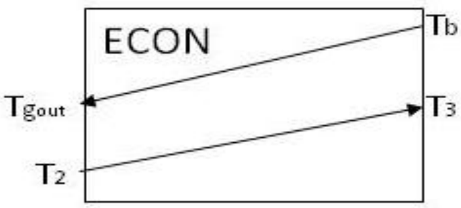

Figure 4. Control volume for the economizer.

The single-phase heat exchange rate in the economizer is defined by the following equations:

$$\dot{Q}_{ec} = \dot{m}_g c_{pg}(T_b - T_{gout}) \quad (2)$$

$$\dot{Q}_{ec} = \dot{m}_{wf}(h_3 - h_2) \quad (3)$$

$$\dot{Q}_{ec} = C_{\min,ec}\varepsilon_{ec}(T_b - T_2) \quad (4)$$
$$C_{min,ec} = \min\left(\dot{m}_g c_{p,g}; \dot{m}_{wf} c_{p,wf}\right) \quad (5)$$

where $\dot{m}_g$ (kg/s) is the mass flow rate of the waste heat fluid, c_p (kJ/kg·K) is the fluid specific heat, and C (kW/K) the thermal capacity rate. The effectiveness ε for this zone (assuming counter-flow single-phase heat transfer) is:

$$\varepsilon_{ec} = \frac{1-\exp[-NTU_{ec}(1-C^*)]}{1-C^*[-NTU_{ec}(1-C^*)]} \quad (6)$$

or

$$\varepsilon = \frac{NTU_{ec}}{NTU_{ec}+1}\text{, if } C^* = 1 \quad (7)$$

where the non-dimensional number transfer of units, NTU, is given in terms of the overall heat transfer coefficient, U (kW/m^2·K), the heat transfer area, A (m^2), and the minimum to maximum capacity rates ratio, C^*.

$$C^* = \frac{C_{min,ec}}{C_{max,ec}} \quad (8)$$

$$NTU_{ec} = \frac{U_{ec}A_{ec}}{C_{min,ec}} \quad (9)$$

2.1.3 Process 3-4 (Evaporator)

The process in the evaporator is characterized by the phase change of the working fluid. The control volume is presented in Fig.5.

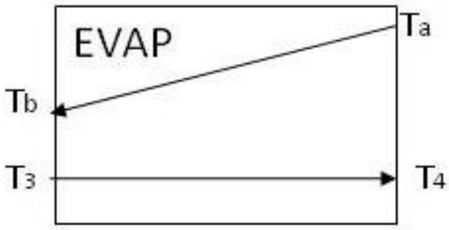

Figure 5. Control volume for the evaporator

The heat transfer rate and effectiveness equations for the evaporating zone are as follows:

$$\dot{Q}_{ev} = \dot{m}_g c_{p,g}(T_a - T_b) \quad (10)$$
$$\dot{Q}_{ev} = \dot{m}_{wf}(h_4 - h_3) \quad (11)$$
$$\dot{Q}_{ev} = \dot{m}_g c_{p,g}\varepsilon_{ev}(T_a - T_3) \quad (12)$$
$$\varepsilon_{ev} = 1 - \exp(-NTU_{ev}) \quad (13)$$

$$NTU_{ev} = \frac{U_{ev}A_{ev}}{\dot{m}_g c_{p,g}(T_a - T_b)} \quad (14)$$

2.1.4 Process 4-5 (Superheater)

The process in the superheater is a single-phase heat exchange between the entering waste heat gas and the working fluid, already in vapor phase. The control volume of the superheater is presented in Fig.6.

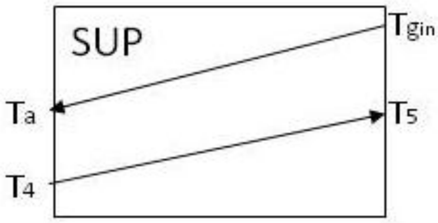

Figure 6. Control volume for the superheater

The equations for the heat transfer and effectiveness in the superheater are as follows:

$$\dot{Q}_{sh} = \dot{m}_g c_{p,g}\left(T_{g,in} - T_a\right) \quad (15)$$

$$\dot{Q}_{sh} = \dot{m}_{wf}(h_5 - h_4) \quad (16)$$
$$\dot{Q}_{sh} = C_{min,sh}\,\varepsilon_{sh}(T_{g,in} - T_4) \quad (17)$$
$$C_{min,sh} = \min(\dot{m}_g c_{p,g}; \dot{m}_{wf} c_{p,wf}) \quad (18)$$

$$\varepsilon_{sh} = \frac{1-\exp[-NTU_{sh}(1-C^*)]}{1-C^*[-NTU_{sh}(1-C^*)]} \quad (19)$$
or
$$\varepsilon_{sh} = \frac{NTU_{sh}}{NTU_{sh}+1}\text{, if } C^* = 1 \quad (20)$$

$$C^* = \frac{C_{min,sh}}{C_{max,sh}} \quad (21)$$

$$NTU_{sh} = \frac{U_{sh}A_{sh}}{C_{min,sh}} \quad (22)$$

2.1.5 Process 5-6 (Turbine)

The expansion of the superheated working fluid vapor in the turbine generates power. The expansion isentropic process is never 100% efficient in converting all of the energy available, it is necessary to include the turbine efficiency, η_t . The power generated by the turbine is:
$$\dot{W}_t = \eta_t(h_5 - h_6) \quad (23)$$
The overall thermal efficiency of the cycle is:
$$\eta_{th} = \frac{W_t - W_p}{Q_{in}} \quad (24)$$
where $\dot{Q}_{in}$ is the sum of the heat inputs of the economizer, evaporator and superheater, $\dot{Q}_{in} = \dot{Q}_{ec} + \dot{Q}_{ev} + \dot{Q}_{su}$, as was established before.

3. ANALYSIS AND RESULTS OF THE ORC

As was mentioned before, only certain data inputs were considered, these are: type of working fluid, ambient temperature, efficiencies of the pump and turbine, specific heat of waste heat gases, mass flow rate of waste heat gases and inlet and outlet temperatures of the waste heat gases.

The thermodynamic properties of the three working fluids: R245fa, R1234yf and R1234ze were obtained by REFPROP 8.0 developed by the National Institute of Standards and Technology (NIST). The critical properties of these dry fluids are listed on Tab.1. The simulation of the organic Rankine cycle was developed using a Fortran 90 platform.

Table 1. Properties of the different working fluids

		R245fa	R1234yf	R1234ze
Molar mass (kg/kmol)		134.05	114.04	114.04
Triple point temperature (K)		0	220	168.62
Critical Point	Temperature (K)	427.2	367.85	382.52
	Pressure (MPa)	3.64	3.38	3.63
	Density(kg/m³)	517	475.55	489.24
Normal boiling point (K)		288.05	243.7	254.2

In order to simulate the performance of the cycle, the following assumptions were made:

1. The ambient temperature is 15°C (288 K).
2. The inlet temperature of the waste heat fluid varied from 250°C (523 K) to 350°C (623 K).
3. The mass flow rate of the waste heat fluid varied from 10 to 20 kg/s.
4. The waste heat fluid has a specific heat of 1,05 kJ/kg·K.
5. The outlet temperature of the waste heat fluid was maintained at 120°C (393 K) to avoid dew point corrosion.
6. The efficiencies of the pump and turbine are 70% and 90%, respectively.

In order to determine the greatest work output of the cycle without knowing the cycle high pressure, a parametric analysis was considered, starting from the cycle low pressure and ending at a reduced pressure of 0.9 (P/P_c), which was considered the operational limit for the cycle not to be transcritical.

From Eqs. (2) to (22) the heat inputs in all of the heat exchangers were obtained, as well as the overall heat transfer coefficient and the necessary mass flow rate of the working fluid.

The power consumption of the pump was calculated with Eq.(1), the power generated by the turbine with Eq.(23) and the overall thermal efficiency of the cycle with Eq.(24).

Figures 7 thru 9 show the net power output of the system with varying temperatures and mass flow rates of the waste heat gases for the three different working fluids. The net power output increases for all working fluids with the increase in temperature and mass flow rate of the waste heat gases. However, for R245fa, the increase in net power output is greater than the other two working fluids with the same increase in temperature and mass flow rate, and thus resulting in a greater maximum output. The second best working fluid is R1234yf in increase of power output for varying temperatures and mass flow rates.

It should also be noted that since the different working fluids have different critical temperatures, close to which their properties start to "degrade", they will be more suitable at different temperature ranges. Again, R245fa has the largest zone of applicability of all the three working fluids.

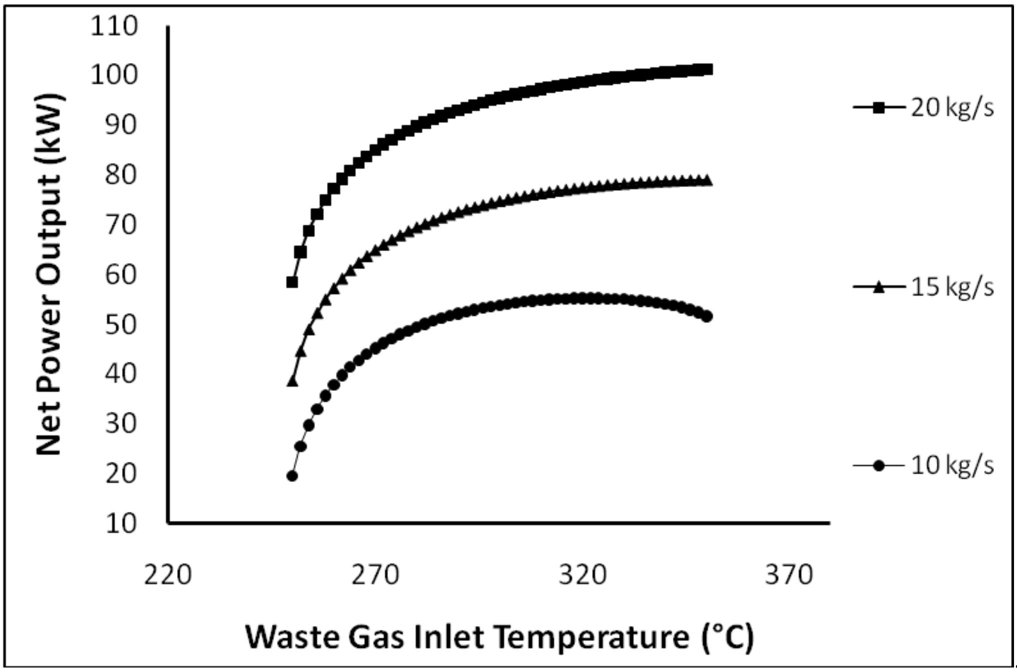

Figure 7. Net power output versus waste gases inlet temperature for various waste gas mass flow rates using R245fa.

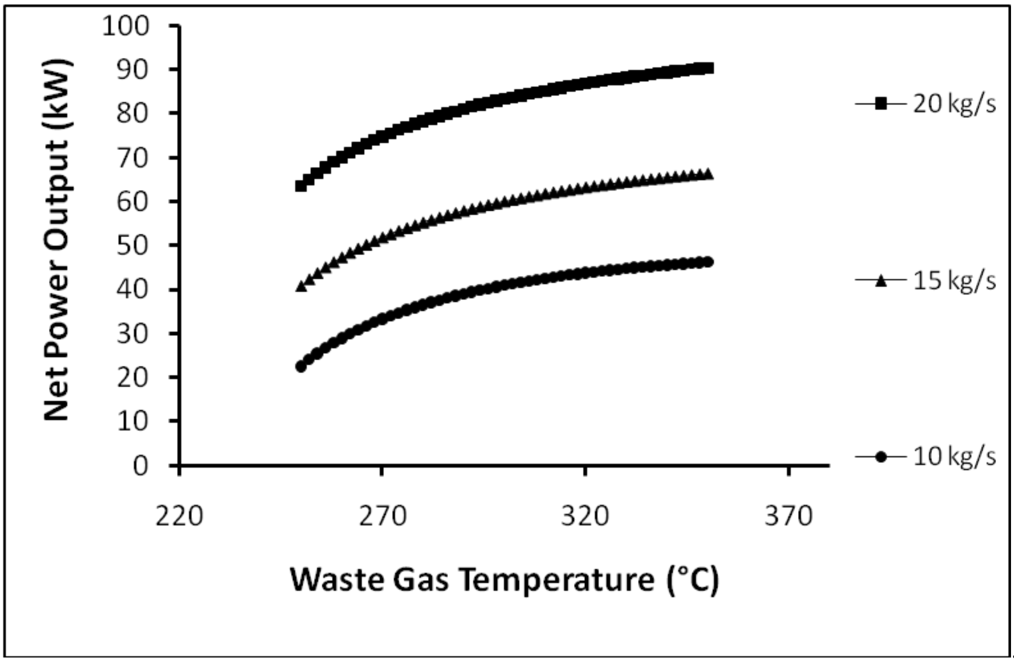

Figure 8. Net power output versus waste gases inlet temperature for various waste gas mass flow rates using R1234yf.

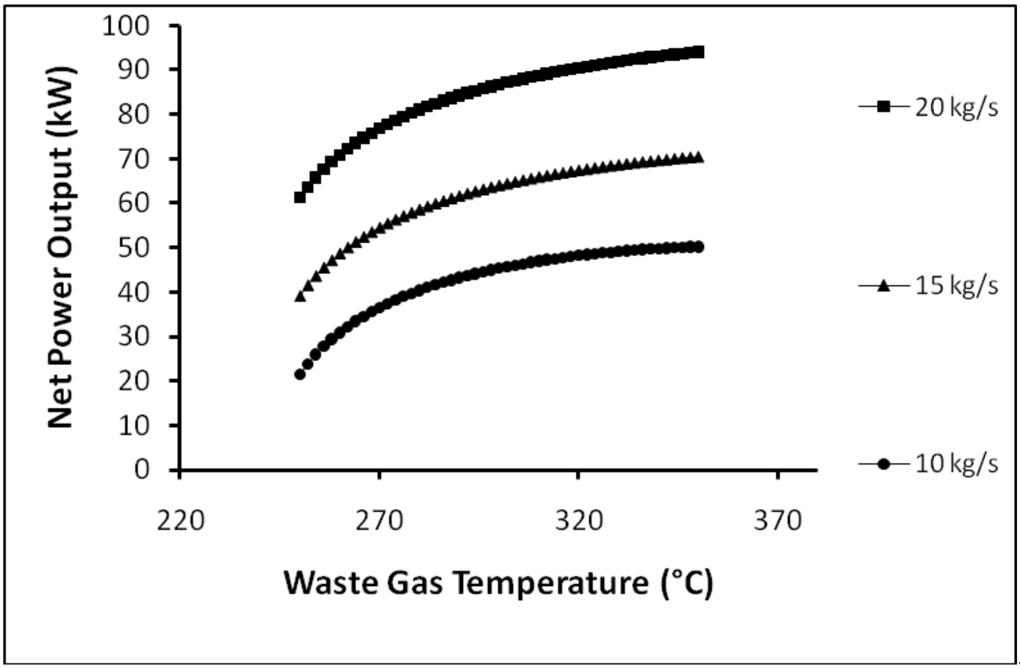

Figure 9. Net power output versus waste gases inlet temperature for various waste gas mass flow rates using R1234ze.

As it is shown in Fig.10, the thermal efficiency of the cycle increases with increasing cycle high pressure, that is, the pressure of the working fluid in the heat exchangers.

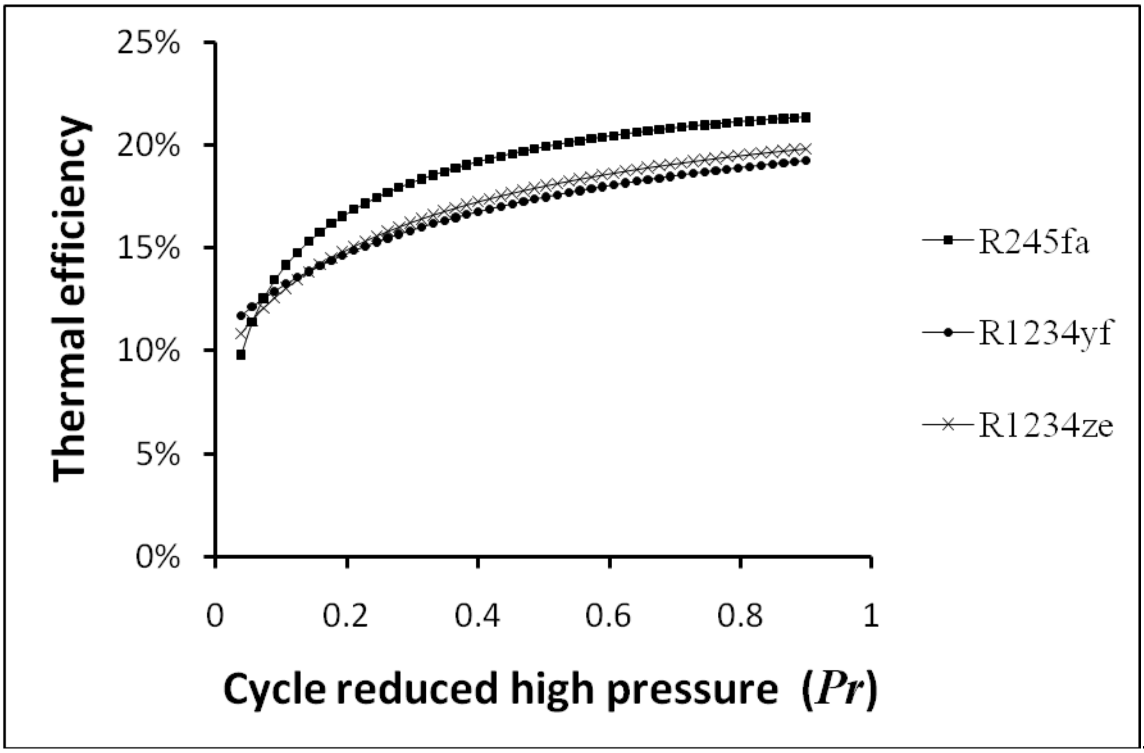

Figure 10. Efficiency versus cycle reduced high pressure for the three working fluids: R245fa, R1234yf, and R1234ze

Since the pressure in the heat exchangers is determined, to a certain extent, by the input of waste gases, this also confirms that the cycle will work best at higher temperatures and mass flow rates of the waste heat. For all three working fluids the efficiency tends to reach an asymptote close to the 20% of efficiency, limited by the properties of the waste heat.

Because all of the working fluids studied are "very dry", there was no significant loss of vapor quality even at the highest cycle pressures and thus the use of a reheater resulted unnecessary.

4. CONCLUDING REMARKS

(1) Using greater flow rates and higher temperatures of the waste heat source, a higher net power output was obtained for all working fluids.
(2) The working fluid, R245fa demonstrated a greater ability to convert energy and is more appropriate to work over a wider range of temperatures.
(3) The thermal efficiency increased as the cycle worked with higher pressures, indicating, again, the need for the cycle to work with the highest temperatures of the waste heat.

5. ACKNOWLEDGEMENTS

Thanks are due to CAPES, FAPERJ and CNPq for the financial support.

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