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Analysis of Vehicle Exhaust Waste Heat Recovery Potential Using a Rankine Cycle

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Abstract – With increasing oil prices and growing interest in cutting emissions of greenhouse gases, waste heat recovery techniques based in Rankine cycle systems, appear as a very promising path to enhance the thermal efficiency of internal combustion engines (ICE). This study evaluates the potential use of thermal energy contained in exhaust gases of vehicles equipped with ICEs. It was developed a numerical model for the thermodynamic analysis of a Rankine cycle that uses waste heat contained in the exhaust gases of an ICE. All characteristics related to a tubular heat exchanger have been incorporated in the thermodynamic model. For the simulations, it was used experimental results obtained in a vehicle tested on a chassis dynamometer. The thermodynamic analysis was performed for the following working fluids: water, R123 and R245fa. The results reveal the advantage of using water as the working fluid in applications of thermal recovery from exhaust gases of vehicles equipped with a spark-ignition engine. The simulations reveal increases in thermal efficiency and mechanical efficiency of around 3% and 16%, respectively, when using an ideal heat exchanger. Considering a tubular heat exchanger, the simulations show an increase of 1.2% in the thermal efficiency and an increase of 5.4% in the mechanical efficiency for an evaporating pressure of 2 MPa. The results confirm the advantages of the use of the thermal energy contained in the exhaust gases through a co-generation application of the Rankine cycle in vehicles.

Keywords – waste heat recovery; Rankine cycle; internal combustion engine; thermodynamic efficiency; heat exchanger.

1. Introduction

Internal combustion engines (ICE) are the major source of motive power in the world, a fact that is expected to continue well into this century. To increase the ICE thermal efficiency and reduce CO_2 emissions, recently waste heat recovery (WHR) based on thermodynamics systems have been explored widely and a number of new technologies have been developed. The Rankine cycles (RC) have a high potential to recover thermal energy from vehicles exhaust gases [1]. Since RC systems generate additional power without requiring extra fuel, both the specific fuel consumption and the pollutant emissions of the vehicle are reduced [2, 3]. The heat recovery RC system is an efficient method for recovering heat (in comparison with other technologies such as thermoelectricity and absorption cycle air-conditioning).

The theoretical Rankine thermodynamic cycle is used in numerous applications to generate electrical power. A heat engine with a vapor power cycle it is the practical engineering alternative to the idealized Carnot cycle. It consists of (1-2) an isentropic compression in a pump, followed by (2-3) an isobaric heat transfer (heating) in a boiler, followed by (3-4) an isentropic expansion through a turbine (or other work-extracting machine), followed by (4-1) an isobaric heat transfer (cooling) in a condenser. The cycle is depicted schematically in Figure 1. In practical Rankine cycles, process 1 to 2 and 3 to 4 are not isentropic.

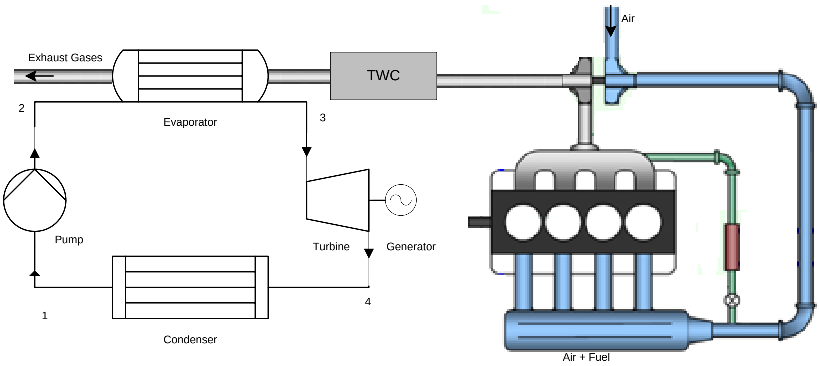

Figure 1 – Schematic of the Rankine cycle system.

A disadvantage of using water as the working fluid is the need to superheat the steam to prevent condensation during the expansion, a problem that results directly from the thermophysical properties of water. Such condensation is a problem because it can lead to erosion of the turbine blades. The choice of alternative working fluids can circumvent the superheating requirement, especially in applications that can be operated at lower temperatures, or that have a lower heat resource temperature. In these situations, an organic working fluid offers advantages in efficiency since in many cases superheating is unnecessary. These cycles are called organic Rankine cycles (ORC).

This paper is divided in two main parts. First, the article evaluates the potential of the thermal energy contained in exhaust gases of vehicles. Subsequently, emphasis is placed on a thermodynamic analysis that allows evaluating the RC efficiency, by developed a numerical model, for both organic (R123 and R245fa) and inorganic (water) fluids. The thermodynamic model uses experimental data as input. In the simulations it was used a tubular heat exchanger.

2. Thermodynamics analysis

The simulation model used in the present study consists of two main sub-models:

i) the simulation of the Rankine cycle thermodynamic processes;

ii) a simulation model used to perform the heat exchanger and dimensional calculations.

2.1. Potential of thermal energy contained in exhaust gases (Input data)

Chassis dynamometer measurements were carried out on a vehicle equipped with a spark ignition engine in order to measure the exhaust gas mass flow rate and temperature for several steady state operating conditions. The vehicle, equipped with a 2.8 litre VR6 spark ignition engine, was tested under steady state operating conditions (i.e., after engine warm-up). For each engine speed (2000, 3000 and 4000 rpm) tests were made for various loads (BMEP – break mean effective pressure). The operating points considered in this article are shown in Table 1. In the table, N is the imposed engine speed, F is the imposed load, BMEP is the break mean effective pressure, $V_{vehicle}$ is the vehicle speed, P_e is the effective power (or brake power) measured at chassis dynamometer, $\dot{m}_g$ is the vehicle exhaust mass flow rate, and T_g is the exhaust gas temperature measured downstream of the three way catalyst. The test conditions originated exhaust mass flow rates and temperatures ranging from 12.8 g/s to 59.7 g/s and 730.9 K to 1052.3 K, respectively.

Table 1 – Test conditions.

Operating point	N [rpm]	F [N]	$BMEP$ [bar]	$V_{vehicle}$ [km/h]	Pe [kW]	Be [N · m]	$\dot{m}_g$ [g/s]	Tg,in [K]
3	2000	1000	1.75	30.1	8.18	39.1	21.0	829.7
9	3000	1500	2.85	48.2	19.97	63.6	37.9	968.7
13	4000	2000	3.98	67.0	37.17	88.7	59.7	1052.3

In this work, the calculated mass composition of the exhaust gases (CO_2 = 20.4%, H_2O = 7.8%, N_2 = 71.8%) was used to evaluate the gas properties. Table 2 show the equations used for the evaluation of the gas properties. Neglecting minor components (CO, HC, NO_x emissions), stoichiometric gasoline engine exhaust gases is basically made up from CO_2, H_2O and N_2. The exhaust gas properties have been calculated using polynomial expressions (see table 2).

Table 2 – Exhaust gases properties.

	Exhaust gases properties
Specific heat capacity [J/kg K]	$c_{pg} = 956.0 + 0.3386 \cdot T_g - 2.476 \times 10^{-5} \cdot {T_g}^2$
Dynamic viscosity [N s/m^2]	$\mu_g = 10^{-6} \times \left(3.807 + 4.731 \times 10^{-2} \cdot T_g - 9.945 \times 10^{-6} \cdot {T_g}^2\right)$
Prandtl number	$Pr = 0.774 + 1.387 \times 10^{-4} \cdot T_g + 1.863 \times 10^{-7} \cdot {T_g}^2 + 7.695 \times 10^{-11} \cdot {T_g}^3$
Thermal conductivity [W/m K]	$k_g = 10^{-3} \times \left(4.643 + 6.493 \times 10^{-2} \cdot T_g\right)$
Density [kg/m^3]	$\rho_g = 1.665 + 2.404 \times 10^{-3} \cdot T_g - 1.121 \times 10^{-6} \cdot {T_g}^2$

Despite all the technological advancements in ICEs, this technology only transforms about 1/3 of the fuel energy into mechanical power. Figure 2 shows the exhaust thermal energy considering that the exhaust gases are cooling down to 200 ºC in the RC evaporator. It can be seen that the waste heat at the vehicle exhaust is equivalent to the vehicle effective power (see Table 1).

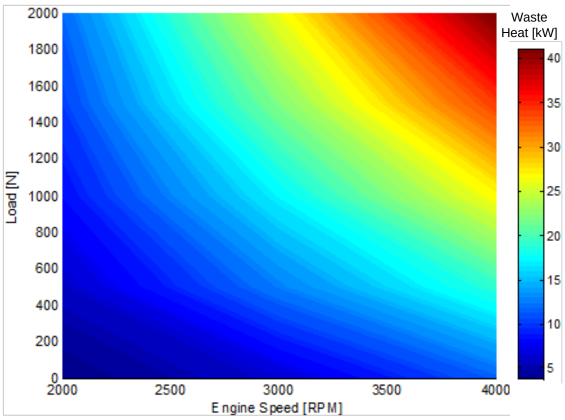

Figure 2 – Exhaust thermal energy.

2.2. Thermodynamic model

A detailed simulation model has been developed for the evaluation of the Rankine cycle system which is used to recover energy from the exhaust gas and other sources of waste heat available in the MCI. The required thermodynamic and transport properties for the water and organic working fluids have been calculated from REFPROP, a database developed by NIST (National Institute of Science and Technology) [4].

The availability of waste heat at different levels of temperature and with a wide range of mass flow rates is usually a problem for the application of RC onto exhaust systems of vehicles. The thermodynamics analysis reported below has been performed for the exhaust conditions corresponding to the operating point 9 (see Table 1). Below we examine the interactions between working fluid properties, pressure level in the evaporator and condenser, working fluid mass flow rate, and RC efficiency and power output. Several organic fluids for use in RC have been proposed in the literature. Amongst them, R123 and R245fa appear to be the most promising ones for the operating conditions used in this work, mainly due to its non-flammable behavior and thermodynamic performance. Therefore, R123 and R245fa have been selected for the present study analysis. Water was also used as working fluid in the present study for comparison purposes. Table 3 lists the main thermophysical properties of the working fluids considered here (R123, R245fa and water).

Table 3 – Summary of the thermophysical properties of the working fluids considered.

Working fluid	Category	p_{cr} [MPa]	T_{cr} [K]	$Normal\ Boiling$ [℃]	$p_{cond,\ T=323K}$ [bar]	Slope of the saturation vapor line
Water	-	22.06	746.95	100	0.123	Negative
R123	HCFC	3.66	456.68	27.8	2.125	Positive
R245fa	HFC	3.64	427.15	14.9	4.012	Positive

The simple RC consists of four main components: a pump, an evaporator, a turbine/generator and a condenser, as shown in Figure 1. There are other RC configurations that permit to increase the recovered thermal energy. However, RC with regeneration requires more piping and RC with reheat requires even more piping and more complex expansion devices.

The pump supplies the working fluid to the evaporator heat exchanger, where the working fluid is heated and vaporized by the exhaust heat. The generated high pressure vapor flows through the expander (e.g., a turbine) and produces power. After the expander, the cold source (water or air), cools and condenses the working fluid into the liquid state in the condenser. The RC system efficiency can be defined as the net power produced referred to the heat received at the evaporator:

$$\eta_c = \frac{W_{Turb} - W_{Pump}}{Q_{in}} \quad [1]$$

For the thermodynamics analysis of the RC the following assumptions were considered:

(i) evaporation pressure varying between condensation pressure, p_{cond}, and critical pressure, p_{crit};
(ii) dry expansion for all fluids (for preservation of the turbine expander);
(iii) isentropic turbine efficiency, $\eta_T = 0.7$;
(iv) isentropic pump efficiency, $\eta_p = 0.75$;
(v) negligible pressure losses in the heat exchangers and pipes.

For both R123 and R245fa the condensation temperature was $T_{cond} = 323\ \mathrm{K}$, which corresponds to the condensation pressures given in Table 2. For water the condensation temperature was $T_{cond} = 373\ \mathrm{K}$, which corresponds to a condensation pressure of 1 bar.

Figure 3 shows a typical T-s process diagram for: a) water; b) R123 and c) R245fa, respectively. The line referring to the engine exhaust gases temperature is imposed in the diagrams base on the experimental condition 9 (see table 1). The figure 3 depicts that the lower the critical temperature the greater the temperature difference between the exhaust gases and the working fluid in the evaporator.

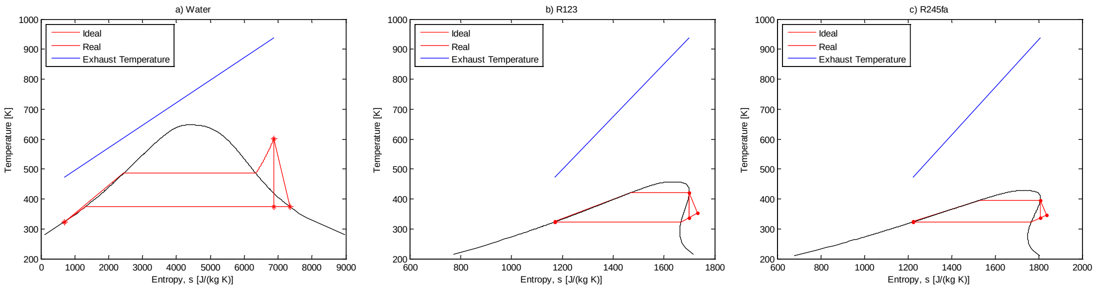

Figure 3 – A typical T-s process diagram for: a) water; b) R123 and c) R245fa.

2.3. Heat exchanger model

An objective function for such system-based optimization is influenced by the main features of heat exchanger operation.

The following characteristics are desirable in an evaporator heat exchanger (boiler) for vehicle exhaust applications:

i) maximize the heat exchanger efficiency;

ii) minimize the pressure drop trough the heat exchanger (this will minimize the negative effect of exhaust back pressure on the ICE);

iii) compactness (minimize the evaporator dimensions).

For a given set of input data (e.g., flow rates and inlet temperatures), heat exchanger geometry, the output data (e.g., the outlet temperatures) will depend on heat transfer and fluid flow phenomena that take place within the boundaries of the heat exchanger. So even though one seeks a system optimum, in the process of determining that optimum, one must fully understand the features of the exchanger as a component. The length and exhaust gas flow area $(A_0 = 2.561 \times 10^{-3}\, m^2$ correspond a diameter equal to $d_0 = 0.571\, m)$ of the duct representing the passage are known and fixed, as well as the fluid (vehicle exhaust gases) inlet temperature and mass flow rate.

The heat exchanger (evaporator) of the Rankine cycle has been simulated by developing appropriate sub-models based on the basic principles of thermodynamics and heat transfer. Heat exchanger is considered to belong to the shell-and-tube counter flow type. Table 4 summarizes the characteristics of the tubular heat exchanger considered in the RC model.

Table 4 – Summary of the characteristics of the tubular heat exchanger.

Number of tubes	$N_t = 43$
Diameter	$d_i = 0{,}01\ m$
Length	$L = 0{,}5\ m$
Distance between tubes[(a)]	$\delta = 0{,}004\ m$

[(a)]Tubes in a equidistant hexagonal arrangement

The heat exchanger was divided into three functional areas i.e. the preheater, the evaporator and the superheater (see figure 4). These are then considered as individual heat exchangers taking into account the necessary boundary conditions for temperatures, flow rate, etc. between them. The amounts of exchanged heat for superheating, evaporation and preheating of the Rankine cycle working fluid are estimated from the basic heat transfer relations [6, 7].

$T_{g,out}$ $T_{g,PP}$ $T_{g,PP2}$ $T_{g,in}$

(a) (b) (c)

T_2 T_{2x} T_{3x} T_3

Figure 4 – Heat exchanger functional area.

In the present simulation model the thermal rating of the heat exchangers is based on the *effectiveness – NTU* (*ε – NTU*) method. The overall heat transfer coefficient U is calculated according to the following relations [7]:

$$U = \frac{1}{\frac{d_e}{d_i h_g} + \frac{d_e R_{di}}{d_i} + \frac{d_e}{2k_m} ln\left({}^{d_e}/_{d_i}\right) + R_{de} + \frac{1}{h_f}} \qquad [2]$$

where k is the thermal conductivity of the tube material and R are the fouling factors. In the present study the tube is assumed to be made of Aluminum, with $k_m = 225\ W/(m \cdot K)$, $R_{de} = 0{,}00018\ m^2K/W$ and $R_{di} = 0{,}000088\ m^2K/W$.

The heat transfer coefficient h is calculated from the relation:

$$h = k_g \frac{Nu_d}{D_h} \qquad [3]$$

When Reynolds number Re_d is lower than 2100 then, the Nusselt number us provided from the Sieder and Tate correlation [6]:

$$Nu_d = 1{,}86 \left[Re_d \cdot Pr \cdot \left(\frac{D_h}{L}\right)\right]^{1/3} (\mu/\mu_w)^m \qquad [4]$$

where μ is the fluid viscosity at the bulk fluid temperature, μ_w is the fluid viscosity at the heat-transfer boundary surface temperature and m=0.14 when Reynolds number is lower than 8000 and for higher values of Reynolds number m=0.25.

For higher values of Reynolds number, the Nusselt number is provide from the Gnielinski correlation [6]:

$$Nu_d = \frac{(f/8)(Re_d - 1000)(Pr)}{1 + 12{,}7(f/8)^{1/2}(Pr^{2/3} - 1)} \left[1 + \left(\frac{D_h}{L}\right)^{2/3}\right] (\mu/\mu_w)^m \qquad [5]$$

where factor *f* is a logarithmic function of the Reynolds number:

$$f = (0{,}79 \cdot \ln(Re_D) - 1{,}64)^{-2} \qquad [6]$$

Figure 5 shows the algorithm developed: (a) the algorithm was implemented in Matlab subroutines to calculate the thermodynamic properties of the Rankine cycle; (b) the algorithm of the exchanger sub-model that was developed and implement in MS Excel.

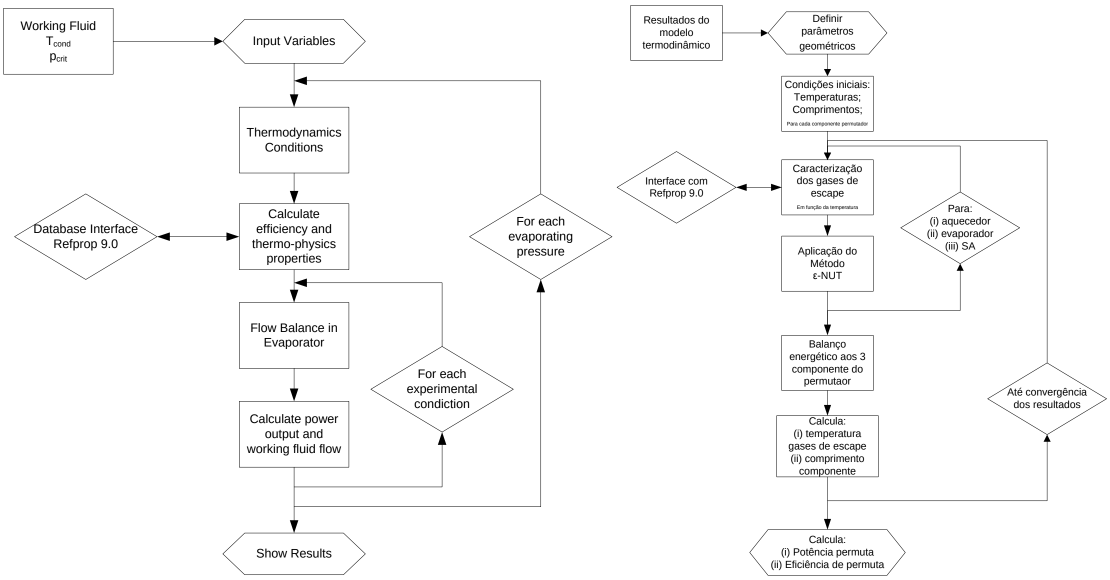

Figure 5 – Algorithm developed: (a) the algorithm was implemented in Matlab subroutines; (b) the algorithm of the exchanger sub-model that was developed and implement in MS Excel..

3. Results

3.1. Rankine cycle thermodynamic analysis

Optimal heat transfer conditions (evaporator heat exchanger with 100% efficiency) were considered in this section, this allows to study the maximum potential of the waste heat recovery. Figures 6a) and 6b) show the cycle efficiency and turbine outlet/inlet expansion ratio (v_4/v_3) as a function of the evaporating pressure for different fluids, respectively (operating point 9, see Table 1). The RC efficiency at the evaporating pressure of 2 MPa is 14.29% for water; 12.03% for R123 and 9.53% for R245fa. The higher temperature difference between the exhaust gases and the working fluid in the evaporator for R123 and R245fa (see Figure 3) induces irreversibility's that are the main cause for low thermodynamic efficiencies with R123 and R245fa as compared to water.

Figure 6b) shows that the R245fa presents the lower expansion ratio (v_4/v_3), regardless of the evaporating pressure. This is mainly due to the higher R245fa condenser pressure (4.012 bar) as compared to that of the water (1 bar). The expansion ratio (v_4/v_3) is particularly significant as it shows how much the fluid volume increases through the expansion process. It should be noted that the ratio (v_4/v_3) can change significantly depending on the characteristics of the working fluid. The expansion ratio is also very important for the expander selection. When the expansion ratio (v_4/v_3) is smaller than 50, expansion efficiencies higher than 0.8 can be achieved using a single stage axial turbine as expander [1].

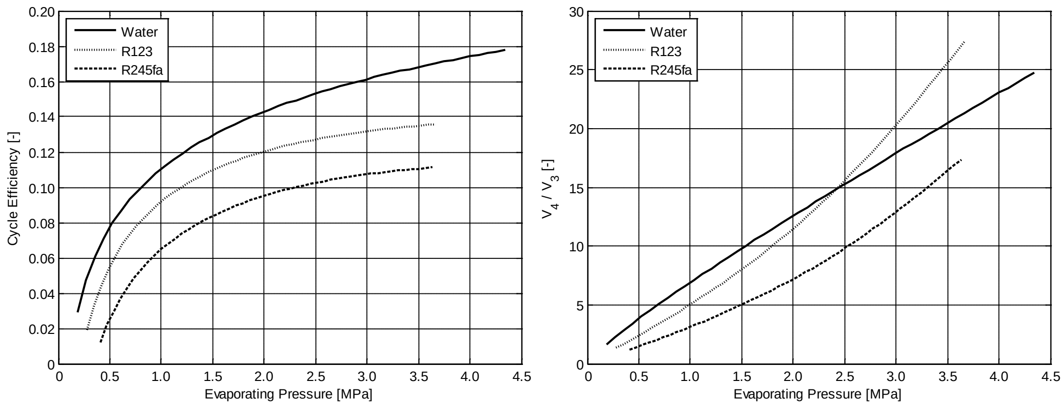

Figure 6a) and 6b) – Cycle efficiency and turbine outlet/inlet expansion ratio (v_4/v_3) as a function of the evaporating pressure for different fluids, respectively (operating point 9, see Table 1).

Figures 7a) and 7b) show the working fluid mass flow rate and net power output as a function of the evaporating pressure for different fluids, respectively (operating point 9, see Table 1). Figure 7a) depicts that the R123 requires the highest fluid mass flow rate as a consequence of the lowest enthalpy increase in the evaporation. The energy balance at the evaporator determines higher mass flow rates for the organic fluids (R123 and R245fa) in order to match the total energy of the vehicle exhaust gases. Figure 7b) shows that the power output of the expander is higher for water and lower for R245fa, regardless of the evaporating pressure. Considering an evaporating pressure of 2 MPa, the ratio of RC expander power output to the vehicle effective power corresponds to 15.95% for water, 13.43% for R123 and 10.64% for R245fa, which represents a considerable improvement of the vehicle efficiency.

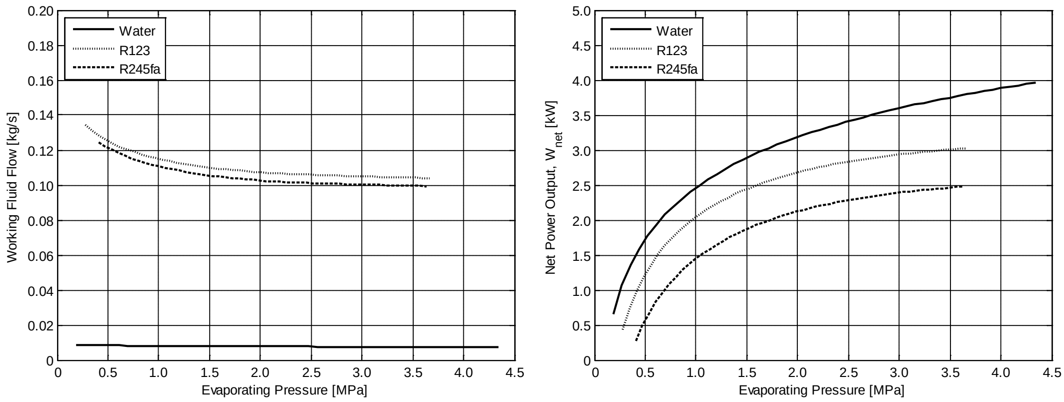

Figure 7a) and 7b) - Working fluid mass flow rate and net power output as a function of the evaporating pressure for different fluids, respectively (operating point 9, see Table 1).

3.2. Heat exchanger analysis

Figures 8 and 9 represent the results of all conducted calculations, introducing the tubular heat exchanger equations in a thermodynamic model. Figures 8a) and 8b) show the working fluid mass flow rate and heat exchanger efficiency as a function of the evaporating pressure for different fluids, respectively (operating point 9, see Table 1).

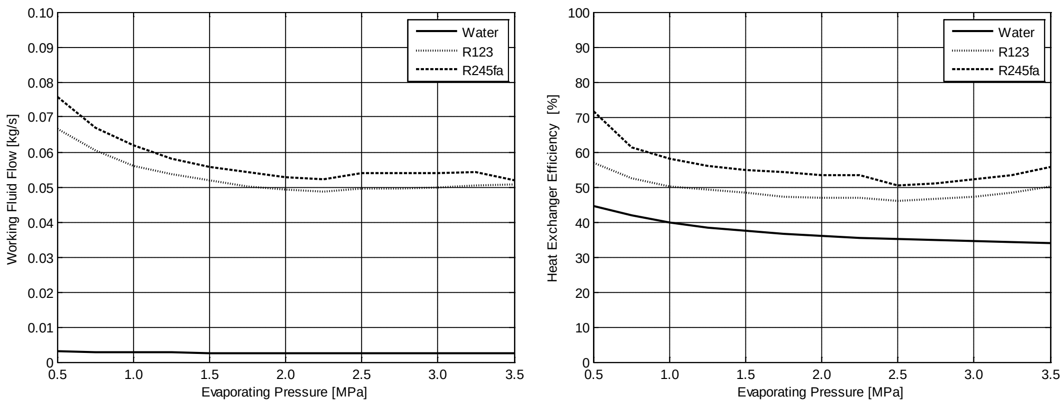

Figure 8a) and 8b) - Working fluid mass flow rate and heat exchanger efficiency as a function of the evaporating pressure for different fluids, respectively (operating point 9, see Table 1).

Figure 9 shows the net power output as a function of the evaporating pressure for different fluids, respectively (operating point 9, see Table 1).

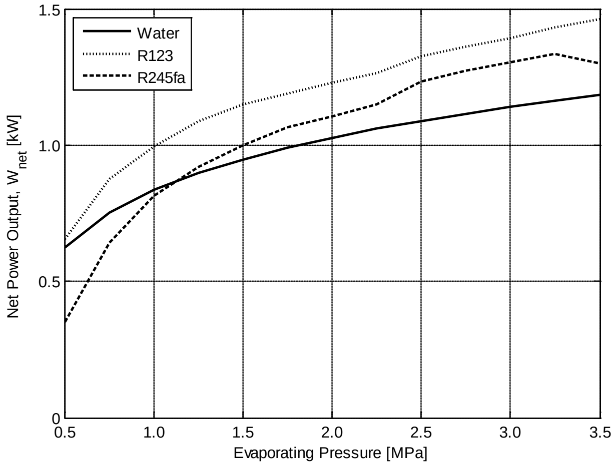

Figure 9 - Net power output (operating point 9, see Table 1).

Figure 9 depicts that the net power output is higher for R123 and lower for water, for evaporating pressures higher than 1.0 MPa in the operating point 9, consequence of the low heat exchanger efficiency for water.

3.3. Global results

Table 5 summarizes the efficiency increase estimated for the combined ICE–RC (ORC). These values have been calculated assuming: (i) 100% efficiency of the heat exchanger; (ii) heat exchanger study in section 2.3.

Table 5 – Global results.

	Working fluid	Thermal efficiency increases [%]			Mechanical efficiency increases [%]		
		(3)	(9)	(13)	(3)	(9)	(13)
Optimal heat transfer conditions (p_{evap}=2MPa)	Water	2.11	2.98	3.52	15.24	15.95	15.94
	R123	1.77	2.51	2.97	12.83	13.43	13.43
	R245fa	1.40	1.99	2.35	10.16	10.64	10.63
Tubular heat exchanger conditions (p_{evap}=2MPa)	Water	0.36	0.96	1.20	2.64	5.14	5.41
	R123	0.96	1.15	1.15	6.96	6.15	5.23
	R245fa	0.85	1.03	1.06	6.18	5.53	4.79

4. Conclusions

The present study demonstrated that the water is the working fluid with greater potential for use in an exhaust heat recovery system with a Rankine cycle for the following reasons: (i) higher thermodynamic efficiency, (ii) condenses easily at atmospheric pressure (low pressure line in Rankine cycle don’t need to be pressurized), (iii) lower quantity (mass) of the working fluid in the installation (less weight), (iv) low price and abundance, (v) no environmental risks.

However, the organic working fluid R123 can also be considered an appropriate use in an exhaust heat recovery system, adapted to ICE when: (i) the exhaust gas temperatures are relatively low, (ii) is necessary to minimize the impact on the efficiency of an ICE (minimized the contact area between the exchanger and the exhaust gas).

Using ideal (100% efficiency) heat exchanger the simulations reveal increases in thermal efficiency and mechanical efficiency of around 3% and 16%, respectively. Considering a tubular heat exchanger, the simulations show an increase of 1.2% in the thermal efficiency and an increase of 5.4% in the mechanical efficiency for an evaporating pressure of 2 MPa. However, it is important to note that the thermal efficiency and mechanical efficiency of a MCI can be improved with the increase in evaporation pressure of the working fluid.

5. References

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