ADVANCED EXERGY ANALYSIS FOR A BOTTOMING ORGANIC RANKINE CYCLE COUPLED TO AN INTERNAL COMBUSTION ENGINE

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Abstract

This paper deals with the evaluation and analysis of a bottoming ORC cycle coupled to an IC engine by means of conventional and advanced exergy analysis. Using experimental data of an ORC coupled to a 2 l turbocharged engine, both conventional and advanced exergy analysis are carried out. Splitting the exergy in the advanced exergy analysis into unavoidable and avoidable provides a measure of the potential of improving the efficiency of this component. On the other hand, splitting the exergy into endogenous and exogenous provides information between interactions among system components. The result of this study shows that there is a high potential of improvement in this type of cycles. Although, from the conventional analysis, the exergy destruction rate of boiler is greater than the one of the expander, condenser and pump, the advanced exergy analysis suggests that the first priority of improvement should be given to the expander, followed by the pump, the condenser and the boiler. A total amount of 3.75 kW (36.5%) of exergy destruction rate could be lowered, taking account that only the avoidable part of the exergy destruction rate can be reduced.

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Keywords

Organic Rankine Cycle, Gasoline engine, Waste Heat Recovery, Advanced Exergy analysis

NOMENCLATURE

Acronyms

ICE	Internal Combustion Engines
ORC	Organic Rankine Cycle
WHR	Waste Heat Recovery

Notation

Latin

$\dot{m}$	Mass flow	kg/s
h	Specific enthalpy	kJ/kg
T	Temperature	°C
$\dot{W}$	Mechanical power	kW
$\dot{Q}$	Thermal power	kW
$\dot{E}$	Exergy	kW
y	Exergy destruction ratio	
P	Pressure	bar
s	Specific entropy	kJ/kgK

e	Specific exergy	kJ/kg

Greek letters

ε	Exergetic efficiency	-
η	Isentropic efficiency	-
Δ	Increment	

Subscripts

F	Fuel
k	K^{th} component
P	Product
D	Refers to exergy destruction (internal exergy loss)
L	Refers to exergy loss (external exergy loss)
tot	Refers to the total system
pp	Pinch point
$1-8$	State points
et	Ethanol
exp	Expander

c	Condenser
b	Boiler
p	Pump
$cycle$	Cycle

Superscripts

UN	Unavoidable
AV	Avoidable
EN	Endogenous
EX	Exogenous
$\cdot$	Time rate

1. Introduction

Regulations for ICE-based transportation in the EU seek carbon dioxide emissions lower than 95 g CO_2/km by 2020 [1]. In order to fulfill these limits, improvements in vehicle fuel consumption have to be achieved [2]. One of the main losses of ICEs happens in the exhaust line. Internal combustion engines transform chemical energy into mechanical energy through combustion; however, only about 15-32% of this energy is effectively used to produce work [3], while most of the fuel energy is wasted through exhaust gases and coolant. Therefore, these sources can be exploited to improve the overall efficiency of the engine. Between these sources, exhaust gases show the largest potential of WHR due to its high level of exergy [4], [5]. Regarding WHR technologies, Rankine cycles are considered as the most promising candidates for improving diesel engines [6]. This technology has an impact on the engine performance:

- Increase of net engine power due to the WHRS
- Increase of cooling loads comparing to the original engine
- Increase of total engine weight
- Increase of pumping losses in the engine

According to Battista et al. [7], an overall mean value of 1% fuel consumption reduction can be achieved in light-duty vehicles taking into account the drawbacks of this system.

Exergy analysis can identify the sources, magnitude and location of thermodynamic inefficiencies of a system, which can give the appropriate information for improving the overall efficiency of the system focusing in the worse exergy balance elements. The conventional exergy analysis has traditionally been studied and applied to some applications in ICE [8], [9]. However, this analysis is used to evaluate the performance of an individual component, without taking into account interactions among components. Therefore, the advanced exergy analysis [10] was proposed to evaluate energy conversion systems by splitting the exergy in endogenous/exogenous and avoidable/unavoidable. Splitting the exergy into unavoidable and avoidable provides a measure of the potential of improving the efficiency of this component. On the other hand, splitting the exergy into endogenous and exogenous provides information between interactions among system components.

This type of analysis has been applied to different energy conversion systems: Kalina cycles in geothermal systems [11], gas turbine systems [12], [13], power plants [14], [15], refrigeration cycles [16], etc.. However, until now no bottoming ORC coupled to an ICE has been analyzed and evaluated using this method. Moreover, values of real experimental tests have been used to this model in order to reproduce actual conditions of these waste heat recovery systems. Hence, on this paper both conventional and advanced exergy analysis were performed to evaluate and analyze a bottoming ORC cycle coupled to an IC engine. The objective of this paper is to quantify, on the basis of reasonable assumptions, the impact of improvements in each component of the system to the global performance of the system using an advanced exergy analysis method [17].

The contents of this paper have been ordered in five different sections:

- Section 1. To introduce the paper.
- Section 2. To describe and present the experimental setup.
- Section 3. To explain the conventional and advanced exergy analysis applied to the case of study and the main assumptions.
- Section 4. To present and analyze the main results of the analysis.
- Section 5. To understand how the efficiency of the expander and the pinch point affects the cycle performance by means of sensitivity analysis.

2. Description of the ORC

Fig 1 and Fig 2 shows respectively the schematic diagram and the experimental installation of the ORC cycle.

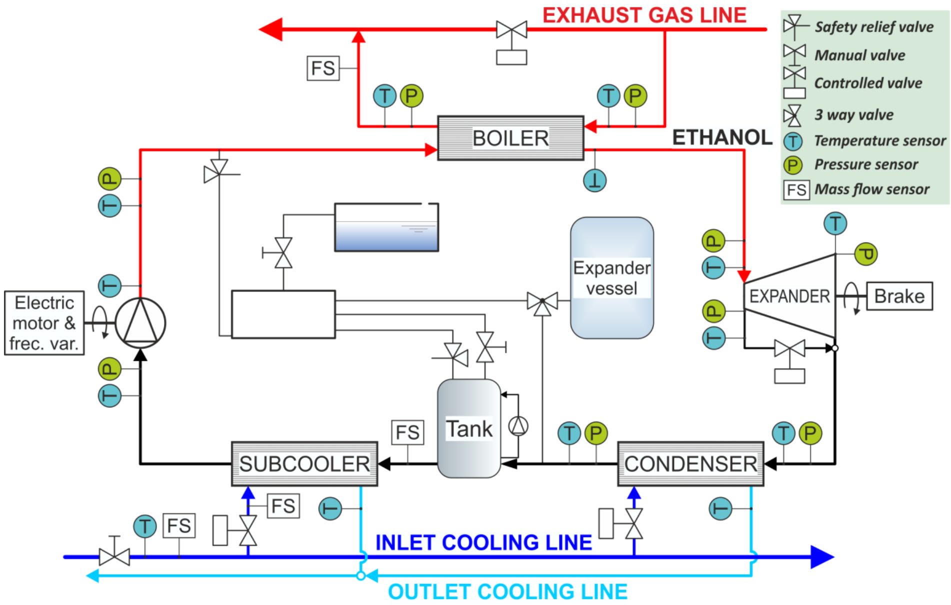

Fig 1. Schematic diagram of the installation

Fig 2. ORC Mock-up

The engine exhaust gases provide heat needed to vaporize the ethanol. Among organic fluids, several authors [18], [19] consider ethanol as a promising fluid due to its great features in energy recovery aspect in the temperature range of a vehicle application (450 °C - 100 °C). Ethanol has been taken into account for its environmental (low GWP and ODP), thermo-physical properties (high expansion ratios, condensation temperatures at atmospheric pressure and low freezing point) and cost features. The cycle efficiency is higher than other organic fluids such as the R245fa (with higher level of GWP) and new refrigerants such as R1234yf or R1233zd, so the power that can be delivered from the expander considering a machine with the same efficiency will be higher in the case of ethanol cycles. First, the working fluid is pumped from the tank at the condensing pressure to the boiler at the evaporating pressure. Then, the working fluid is pre-heated, vaporized and superheated in the heat exchanger. The ethanol vapor expands from the evaporating pressure to the condensing pressure in the expander machine. Finally, low pressure vapor is extracted from the expander and flows to the condenser, where it condenses using cooling water. The boiler ensures the heat transfer from exhaust gas to the working fluid. The condenser is followed by an expander vessel in order to impose the low pressure in the installation and a liquid reservoir. The expander prototype is a piston swash-plate.

3. Thermodynamic analysis

Fig 3 shows a simplified diagram of the ORC. References to this diagram will be made during the whole article. In this figure, the main elements of the cycle are presented, i.e. boiler, expander, condenser and a pump.

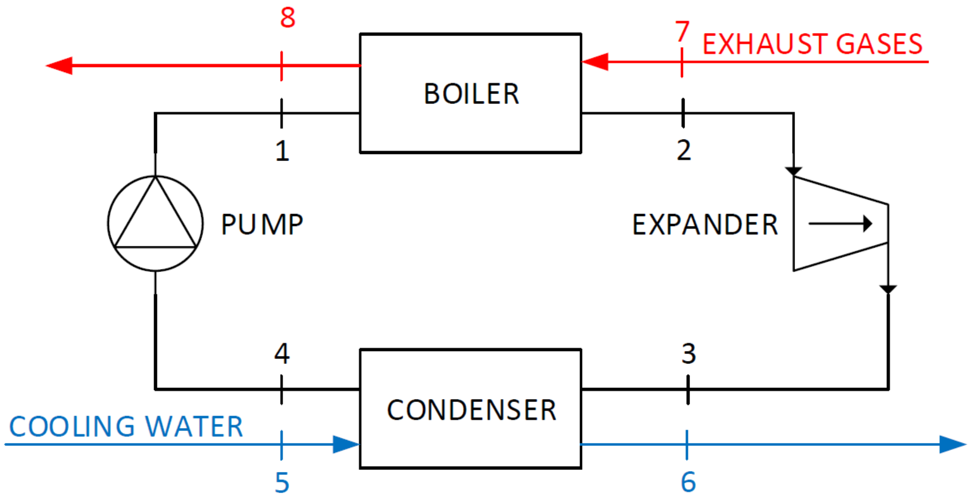

Fig 3. Cycle diagram for thermodynamic analysis

The main assumptions in analyzing the ORC are as follows:

- Thermodynamic cycle method [20] was chosen because it is the most convenient method and provides the best results for systems in which a thermodynamic cycle can be defined.
- The system works under steady state conditions.
- Energy losses and changes in kinetic and potential energies are neglected [21].

3.1. Conventional Exergy Analysis

The basic equations of the conventional exergy analysis for the Kth component of the system (boiler, expander, condenser or pump) are presented in Eq. 1 and 2. Using the thermodynamic terms of exergy of fuel and product, the exergy balances for the Kth component and for the overall system can be defined.

$$\dot{E}_{F,k} = \dot{E}_{P,k} + \dot{E}_{D,k} \quad (1)$$

$$\dot{E}_{F,tot} = \dot{E}_{P,tot} + \dot{E}_{L,tot} + \dot{E}_{D,tot} \quad (2)$$

$$\varepsilon_k = \frac{\dot{E}_{P,k}}{\dot{E}_{F,k}} = 1 - \frac{\dot{E}_{D,k}}{\dot{E}_{F,k}} \quad (3)$$

$$y_{D,k} = \frac{\dot{E}_{D,k}}{\dot{E}_{F,k}} \quad (4)$$

$$y_{D,k}{}^{*} = \frac{\dot{E}_{D,k}}{\dot{E}_{D,tot}} \quad (5)$$

Where $\dot{E}_{F,k}$, $\dot{E}_{P,k}$ and $\dot{E}_{D,k}$ are respectively the exergy rate of fuel, product and internal exergy loss in the Kth component. The subscript tot means the total amount of the overall system. $\dot{E}_{L,tot}$ corresponds to external exergy loss in the overall system. ε_k, $y_{D,k}$ and $y_{D,k}{}^{*}$ are the exergy efficiency, the exergy destruction ratio and the exergy rate of fuel with the total exergy destruction respectively.

The exergy destruction (or internal exergy loss) is the exergy destroyed due to irreversibility within a system or a Kth component. At the component level, exergy flows are associated with fuel or product in each component. Therefore, the exergy loss in the Kth component is related with the transfer of thermal energy to the ambient (heat loss). The exergy loss (or external exergy loss) is the exergy transfer from the system to the surroundings. Considering the boundaries of the component analysis fixed at ambient temperature, the exergy loss is 0 and the thermodynamic inefficiencies consist only of exergy destruction. Therefore, the exergy loss is related only with the overall system and not with the Kth component.

The energy and exergy balances for the system components as control volumes are presented in Table 1 and Table 2.

Table 1. Energy balance equations

Cycle component	Energy balance equations
Expander	$\eta_{exp} = \frac{\dot{W}_{exp}}{\dot{W}_{exp,iso}}, \dot{W}_{exp,iso} = \dot{m}_{et} * \left(h_2 - h_{3,iso}\right), \dot{W}_{exp} = \dot{m}_{et} * (h_2 - h_3)$
Pump	$\eta_p = \frac{\dot{W}_{p,iso}}{\dot{W}_p}, \dot{W}_{p,iso} = \dot{m}_{et} * \left(h_{1,iso} - h_4\right), \dot{W}_p = \dot{m}_{et} * (h_1 - h_4)$
Condenser	$\dot{Q}_c = \dot{m}_{et} * (h_3 - h_4)$
Boiler	$\dot{Q}_e = \dot{m}_{et} * (h_2 - h_1)$

Table 2. Exergy balance equations

Cycle component	Exergy balance equations
Expander	$\dot{E}_2 = \dot{E}_3 + \dot{W}_{exp} + \dot{E}_{D,exp}$
Pump	$\dot{E}_4 + \dot{W}_p = \dot{E}_1 + \dot{E}_{D,p}$
Condenser	$\dot{E}_3 + \dot{E}_5 = \dot{E}_4 + \dot{E}_6 + \dot{E}_{D,c}$
Boiler	$\dot{E}_7 + \dot{E}_1 = \dot{E}_2 + \dot{E}_8 + \dot{E}_{D,b}$

3.2. Advanced Exergy Analysis

In the advanced exergy analysis [17], the rate of exergy destruction in the Kth component of the system is split into endogenous / exogenous and avoidable / unavoidable.

Endogenous/Exogenous

Endogenous exergy destruction in Kth component is related to the irreversibility occurring inside this component, whereas the exogenous part is associated with the irreversibilites taking place in the rest of the components of the system [17]. Therefore, the endogenous exergy destruction in Kth component ($\dot{E}_{D,k}^{EN}$) is the part of the total exergy destruction in the Kth component ($\dot{E}_{D,k}$) obtained considering that all the components operate ideally and the component being examined operates with real efficiency (Hybrid Process). In a Hybrid Process (or Hybrid Cycle) only one component is real, i.e., operates with its real efficiency, while all other components operate in a theoretical way. In this case, the exergy destruction within the component being considered represents the endogenous exergy destruction. Thus, step-by-step introducing irreversibilities successively in each system component the endogenous exergy destruction within each component is calculated. Therefore, in order to compute the endogenous exergy destruction in the Kth component ($\dot{E}_{D,k}^{EN}$), a Hybrid Cycle for each component has to be simulated. Exogenous exergy destruction ($\dot{E}_{D,k}^{EX}$) is the difference between the exergy destruction value of the variable within the component in the real system ($\dot{E}_{D,k}$) and the endogenous part ($\dot{E}_{D,k}^{EN}$). Eq. 1 shows the splitting between both parts, where EN and EX indicate the endogenous and exogenous parts, respectively.

$$\dot{E}_{D,k} = \dot{E}_{D,k}^{EN} + \dot{E}_{D,k}^{EX} \quad (1)$$

Moreover, the exogenous exergy destruction can be further split (Eq. 2) as the effect of exergy destruction within the rth component caused by the exergy destruction of the Kth component and a term called mexogenous exergy destruction ($\dot{E}_{D,k}^{MX}$) [15], which considers the simultaneous interactions of all other n-1 elements.

$$\dot{E}_{D,k}^{EX} = \dot{E}_{D,k}^{MX} + \sum_{\substack{r=1 \\ r \neq k}}^{n} \dot{E}_{D,k}^{EX,r} \qquad (2)$$

Avoidable/Unavoidable

Unavoidable exergy destruction in Kth component cannot be reduced due to technological limitations (material characteristics, production costs and manufacturing methods), whereas the avoidable part, which is the remaining part, can be reduced improving the design of this component [17]. Therefore, the unavoidable exergy destruction ($\dot{E}_{D,k}^{UN}$) is the part of total exergy destruction within the Kth component ($\dot{E}_{D,k}$) considering that all the components operate in unavoidable conditions. Avoidable exergy destruction ($\dot{E}_{D,k}^{AV}$) in the Kth component is the difference between the exergy destruction value of the variable within the Kth component in the real system ($\dot{E}_{D,k}$) and the unavoidable part ($\dot{E}_{D,k}^{UN}$). Eq. 3 shows the splitting between both parts, where UN and AV indicate the unavoidable and avoidable parts, respectively.

$$\dot{E}_{D,k} = \dot{E}_{D,k}^{UN} + \dot{E}_{D,k}^{AV} \qquad (3)$$

In order to obtain the unavoidable exergy destruction Eq. 4 was proposed by Tsatsaronis et. al [22]. In order to obtain the ratio $\left(\frac{\dot{E}_D}{\dot{E}_P}\right)_k^{UN}$ for the Kth component, the system is solved considering that each component operates under the best possible conditions considering technological limitations. This ratio is the main parameter to calculate the unavoidable part of the exergy destruction rate of each individual component in a real process.

$$\dot{E}_{D,k}^{UN} = \dot{E}_{P,k} * \left(\frac{\dot{E}_D}{\dot{E}_P}\right)_k^{UN} \quad (4)$$

3.2.1 Combination of the splitting

By combining the two splitting approaches, the unavoidable-exogenous ($\dot{E}_{D,k}^{UN,EX}$), the unavoidable-endogenous ($\dot{E}_{D,k}^{UN,EN}$), the avoidable-exogenous ($\dot{E}_{D,k}^{AV,EX}$) and the avoidable-endogenous ($\dot{E}_{D,k}^{AV,EN}$) values can be obtained using Eq. 5, 6, 7 and 8.

$$\dot{E}_{D,k}^{UN,EN} = \dot{E}_{P,k}^{EN} * \left(\frac{\dot{E}_D}{\dot{E}_P}\right)_k^{UN} \quad (5)$$

$$\dot{E}_{D,k}^{UN,EX} = \dot{E}_{D,k}^{UN} - \dot{E}_{D,k}^{UN,EN} \quad (6)$$

$$\dot{E}_{D,k}^{AV,EN} = \dot{E}_{D,k}^{EN} - \dot{E}_{D,k}^{UN,EN} \quad (7)$$

$$\dot{E}_{D,k}^{AV,EX} = \dot{E}_{D,k}^{EX} - \dot{E}_{D,k}^{UN,EN} \quad (8)$$

3.3. Assumptions

In this analysis, three thermodynamic cycles were proposed, i.e. real, ideal and unavoidable. The following assumptions were adopted:

- In the real cycle, isentropic efficiencies of the expander machine and the pump, pinch point and pressure drops in the heat exchangers are obtained by experimental points presented in previous articles [23].

- In the ideal cycle, isentropic efficiencies of expander and pump and efficiencies of condenser and boiler are considered 100%. Pressure drops are assumed to be zero on condenser and boiler processes.
- In the unavoidable cycle, efficiencies of the expander and the pump are considered respectively 80% [24] and 95% [25], assuming this level the maximum that could be exceeded due to technological limitations. Improvements in valve timing and oil refrigeration loop should be made. Pinch points and pressure drops in this cycle are lower than the real cycle but also considering technological limitations [26].

Table 3 shows a summary of these variables used to define the different cycles.

Table 3. Assumptions for real, ideal and unavoidable

Component	Real		Ideal		Unavoidable	
Expander	η_{exp}	43%	η_{exp}	100%	η_{exp}	80%
Condenser	ΔT_{pp} (°C)	5.00	ΔT_{pp}(°C)	0	ΔT_{pp}(°C)	2
	ΔP	1.60%	ΔP	0%	ΔP	1%
	εc	83%	εc	100%	εc	90%
Pump	η_p	89%	η_p	100%	η_p	95%
Boiler	ΔT_{pp}(°C)	50.00	ΔT_{pp}(°C)	0	ΔT_{pp}(°C)	10
	ΔP	0.79%	ΔP	0%	ΔP	0.5%
	εb	98%	εb	100%	εb	99.0%

4. Simulation results and discussion

For the exergy analysis, steady-state simulations were performed by modeling the cycle described on Fig 3. As shown in Fig 4, the evaporation process corresponds to 1-2, the expansion process to 2-3, the condensation process to 3-4 and the pumping process to 4-1.The major difference between these cycles correspond to the expansion process (2-3). In order to optimize the cycle, get the maximum power in the expander and avoid entering the two-phase zone during expansion, the high pressure has been set to 0.65*$P_{critical}$ and the low pressure to 1 bar to avoid air intakes in the ducts. Depending on the cycle, different pressure drops have been taken into account.

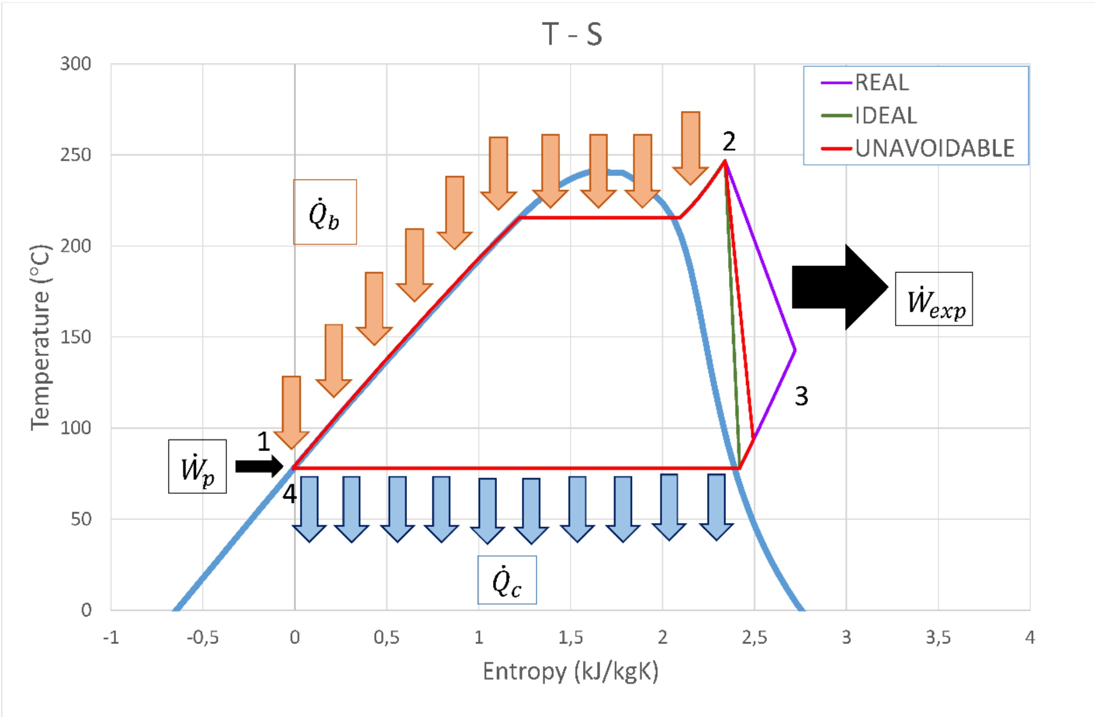

Fig 4. Ideal, real and unavoidable cycles in the T-S Diagram

Fig 5 shows the evaporation process in the three cycles: ideal, real and unavoidable. In all these cycles, the inlet temperature and the mass flow of exhaust gases have remained constant. Depending on the cycle, the pinch point changes from 50 ºC in the real case, 10 ºC in the unavoidable case and 0 ºC in the ideal one. A change of temperature at the outlet of exhaust gases implies a change in the power released by the boiler and thus, the ethanol mass flow flowing in the cycle. Therefore, in order to visualize the evaporation process in the three cycles, the power percentage in % have been plotted in the X axis. The exergy destruction in the boiler is proportional to the area between exhaust gases and ethanol. As it can be seen in this figure, the exergy destruction in the ideal case is lower than the real one.

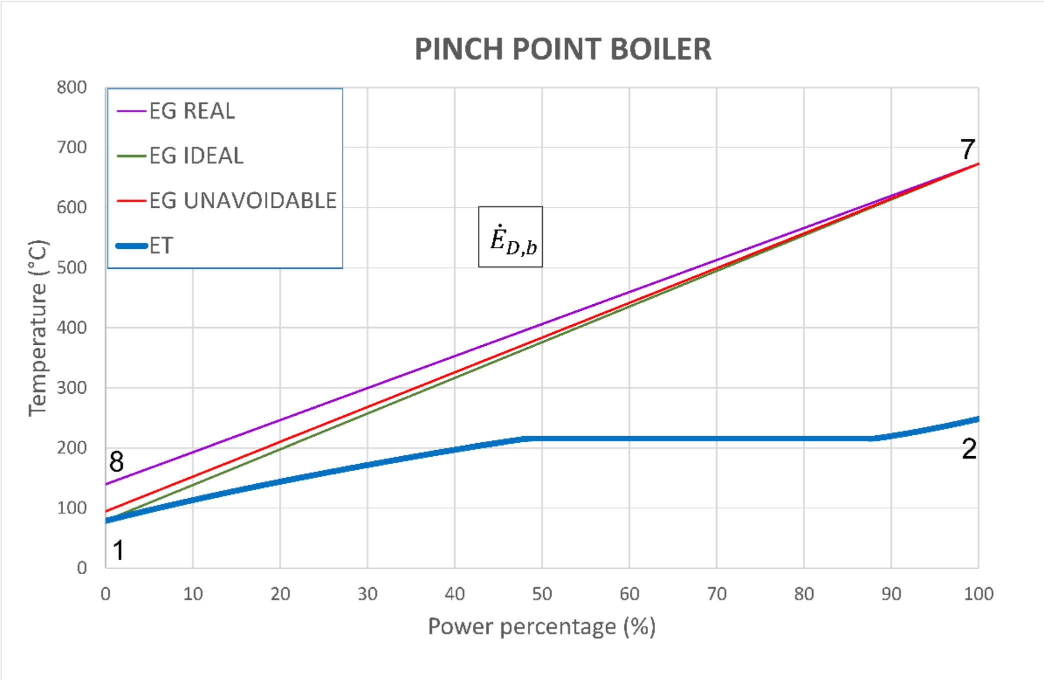

Fig 5. Ideal, real and unavoidable cycles in the evaporation process

Table 4, Table 5 and Table 6 indicate the thermodynamic properties and the mass flow rates at different state points of the ORC (Fig 3) under real, ideal and unavoidable conditions respectively. As previously mentioned, the ethanol mass flow in the cycle is higher in the ideal case than in the real one due to thermodynamic restrictions. Pressure drop in the boiler and the condenser have been considered both in the real and unavoidable case. The last two columns are related with the calculation of exergy in each state point.

Table 4. Thermodynamic properties and mass flow rates of the ORC under real conditions

Point	T (°C)	P (bar)	h (kJ/kg)	$\dot{m}$ (kg/s)	s (kJ/kgK)	e (kJ/kg)	$\dot{E}$(kW)
1	78.54	39.65	262.8	0.02623	1.028	17.49	0.459
2	246.6	39.65	1336	0.02623	3.39	386.8	10.146
3	142.7	0.984	1229	0.02623	3.768	166.7	4.373
4	77.53	0.984	256.9	0.02623	1.026	12.16	0.319
5	50	1.5	209.4	0.2083	0.7037	4.252	0.886
6	74.88	1.5	313.5	0.2083	1.014	15.87	3.306
7	672.8	1.071	985	0.04799	6.888	331	15.885
8	128.5	1.013	403	0.04799	5.996	14.86	0.713

Table 5. Thermodynamic properties and mass flow rates of the ORC under ideal conditions

Point	T (°C)	P (bar)	h (kJ/kg)	$\dot{m}$ (kg/s)	s (kJ/kgK)	e (kJ/kg)	$\dot{E}$(kW)
1	78.7400	39.96	263.5	0.02923	1.029	17.63	0.515
2	247	39.96	1336	0.02923	3.39	387.2	11.318
3	78.5	1	1087	0.02923	3.39	137.6	4.022
4	77.94	1	258.2	0.02923	1.029	12.35	0.361
5	50	1.5	209.4	0.2083	0.7037	4.252	0.886
6	77.81	1.5	325.8	0.2083	1.049	17.68	3.683
7	672.8	1.071	985	0.04799	6.888	331	15.885
8	78.74	1.013	352.7	0.04799	5.862	4.346	0.209

Table 6. Thermodynamic properties and mass flow rates of the ORC under unavoidable conditions

Point	T (°C)	P (bar)	h (kJ/kg)	$\dot{m}$ (kg/s)	s (kJ/kgK)	e (kJ/kg)	$\dot{E}$(kW)
1	78.58	39.76	262.9	0.02845	1.028	17.52	0.498
2	246.7	39.76	1336	0.02845	3.39	386.9	11.007
3	93.45	0.99	1136	0.02845	3.531	145	4.125
4	77.68	0.99	257.4	0.02845	1.027	12.23	0.348
5	50	1.5	209.4	0.2083	0.7037	4.252	0.886
6	75.84	1.5	317.6	0.2083	1.026	16.45	3.427
7	672.8	1.071	985	0.04799	6.888	331	15.885
8	88.58	1.013	362.6	0.04799	5.89	5.984	0.287

Table 7 shows the net power and the cycle energy efficiency and cycle exergy efficiency for ideal, unavoidable and real cases. As it can be seen, net power is reduced in the real cycle because two effects: lower ethanol mass flow due to higher pinch point in the boiler and lower isentropic efficiency in the expander. As a global consequence, both the cycle energy efficiency (defined as the net power divided by the power of the boiler) and cycle exergy efficiency (defined as the net power divided by the exergy rate of fuel in the boiler) are reduced comparing them to the ideal cycle. Therefore, cycle efficiency corresponds to 22.72% in the ideal case and technical limitations give a value of 18.13% in the case of unavoidable cycle. The real cycle gives a value of 9.42%. These values correspond to the ones found in literature [27].

Table 7. Power and efficiency of ideal, unavoidable and real cycles

	Ideal	**Unavoidable**	**Real**
$\dot{W}_{exp}$ (kW)	7.28	5.69	2.81
$\dot{W}_{p}$ (kW)	0.15	0.16	0.15
$\dot{W}_{net}$ (kW)	7.12	5.53	2.65
$\dot{Q}_{b}$ (kW)	31.35	30.53	28.15
$\dot{Q}_{c}$ (kW)	24.23	25.00	25.50
$\dot{E}_{b}$ (kW)	15.67	15.59	15.17
η_{cycle}	22.72%	18.13%	9.42%
ε_{cycle}	45.44%	35.48%	17.48%

Considering conventional exergy equations applied to this particular application and presented in Table 2, the following results are obtained. The total exergy fuel rate is the difference between exergy rates of exhaust gases entering and ethanol leaving the boiler. The expander power output is considered the rate of total products exergy. Table 8, Table 9 and Table 10 show the results of conventional exergy analysis for ORC under real, ideal and unavoidable conditions respectively.

Table 8. Results of conventional exergy analysis under real conditions

Component	E_F (kW)	E_P (kW)	E_D (kW)	ε	Y_k	Y_k^*
Expander	5.77	2.81	2.97	49%	51%	29%
Pump	0.33	0.14	0.19	42%	58%	2%
Condenser	4.05	2.42	1.63	60%	40%	16%
Boiler	15.17	9.69	5.48	64%	36%	53%
Overall System	15.17	2.81	10.84	18%	71%	100%

Table 9. Results of conventional exergy analysis under ideal conditions

Component	E_F (kW)	E_P (kW)	E_D (kW)	ε	Y_k	Y_k^*
Expander	7.30	7.28	0.02	99.8%	0.2%	0.3%
Pump	0.15	0.15	0.00	99.6%	0.4%	0.0%
Condenser	3.66	2.80	0.86	76.4%	23.6%	15.0%
Boiler	15.68	10.80	4.87	68.9%	31.1%	84.7%
Overall System	15.68	7.28	5.76	46.4%	36.7%	100.0%

Table 10. Results of conventional exergy analysis under unavoidable conditions

Component	E_F (kW)	E_P (kW)	E_D (kW)	ε	Y_k	Y_k^*
Expander	6.88	5.69	1.19	83%	17%	16%
Pump	0.22	0.15	0.07	67%	33%	1%
Condenser	3.78	2.54	1.24	67%	33%	16%
Boiler	15.60	10.51	5.09	67%	33%	67%
Overall System	15.60	5.69	7.59	36%	49%	100%

Based on the results obtained from conventional analysis (Table 8), the overall system should be improved following priorities for the components with higher exergy destruction: boiler (5.48 kW), expander (2.97 kW), condenser (1.63 kW) and pump (0.19 kW). In order to increase the ORC efficiency a reduction in exergy destruction rates are needed. From a conventional exergy analysis, it is not possible to distinguish between irreversibilities occurring in other components and the component itself. The advanced exergy analysis [28] evaluates the detailed interactions between components of the overall system, and the real potential of improvement a component within a system.

In the advanced exergy analysis the endogenous exergy destruction rate for the K^{th} component is obtained by calculating several cycles (the same as number of components) considering the component under study with real values of the parameter and the rest with ideal conditions. The exogenous exergy destruction rate will be calculated by difference to the total exergy destruction (Eq. 1). The unavoidable exergy destruction rate is obtained when all the components work under unavoidable conditions. Once the unavoidable conditions are calculated, the ratio of exergy destruction to the product exergy rate is computed (Eq. 4). The avoidable exergy destruction rate will be computed by difference (Eq. 3). To split the exergy between unavoidable-exogenous, unavoidable-endogenous, avoidable-exogenous and avoidable-endogenous Eq. 5 to 8 were applied.

Fig 6 shows a flow chart of exergy destruction rate in the K^{th} component.

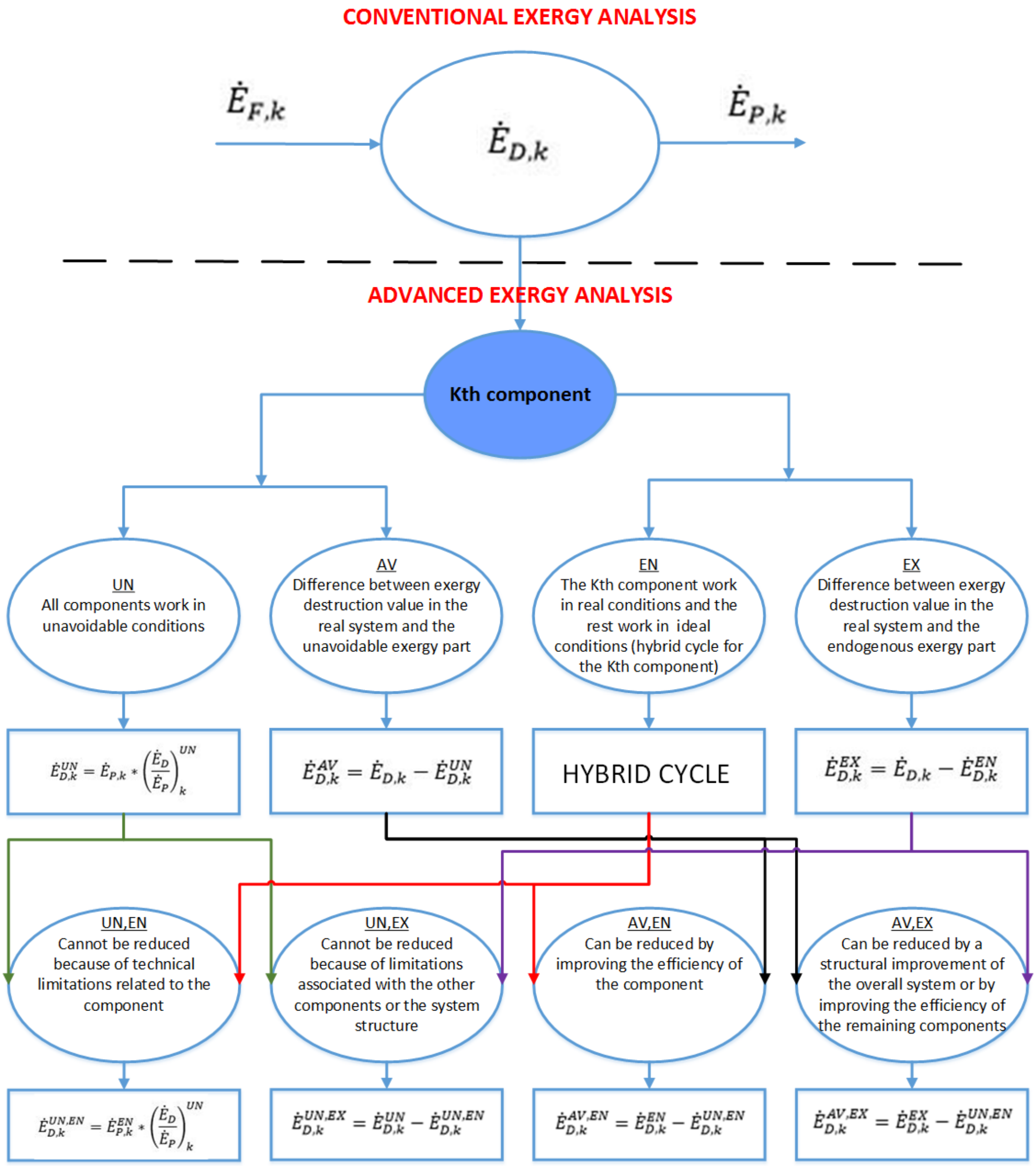

Fig 6. Flow chart of exergy destruction rate (Conventional and Advanced) in the K^{th} component

Results from the previous analysis are presented in Table 11.

Table 11. Results of advanced exergy analysis (kW)

Component	$\dot{E}_D$	$\dot{E}_D^{EN}$	$\dot{E}_D^{EX}$	$\dot{E}_D^{AV}$	$\dot{E}_D^{UN}$	$\dot{E}_D^{AV,EN}$	$\dot{E}_D^{UN,EN}$	$\dot{E}_D^{AV,EX}$	$\dot{E}_D^{UN,EX}$
Boiler	5.48	5.48	0.00	0.79	4.69	0.79	4.69	0.00	0.00
Expander	2.97	3.29	-0.32	2.38	0.59	2.63	0.66	-0.25	-0.07
Condenser	1.63	1.37	0.26	0.46	1.18	0.27	1.10	0.19	0.08
Pump	0.19	0.21	-0.02	0.12	0.07	0.13	0.08	-0.01	-0.01
Overall System	10.27	10.35	-0.07	3.75	6.52	3.82	6.53	-0.07	0.00

As shown in Table 11, the value of endogenous exergy is greater than the value of exogenous exergy in all the system components. Therefore, the greatest contribution to the exergy destruction rate in each of the components comes from the internal irreversibility of the component itself. Regarding to exogenous exergy, the condenser have the highest value (0.26 kW). Therefore, a modification in the other component efficiencies can lead to a reduction in the exergy destruction rate of this element and an improvement in overall cycle efficiency.

Interactions between different components can be positive or negative. These two impacts could be the result of mass flow changes or thermodynamic property variation of material flows through the K^{th} component due to the introduction of additional irreversibilities in the system. In this system, some components have values of endogenous exergy greater than the exergy destruction rate itself. This can be analyzed by the results of the specific advanced exergy analysis (Table 12). Table 12 shows that the system is more efficient (less exergy destroyed in kJ/kg for all the components) in the endogenous case than in the real case. However, changing the conditions from the ideal case to the endogenous case (the K^{th} component is real and the rest are ideal), the ethanol mass flow changes between both cases due to changes in the pinch point. Hence, the result is that the endogenous exergy destruction rate in kW is higher than the real exergy destruction rate in the expander and pump (Table 11).

Table 12. Results of specific advanced exergy analysis (kJ/kg)

Component	E_D	E_D^{EN}	E_D^{EX}	E_D^{AV}	E_D^{UN}	$E_D^{AV,EN}$	$E_D^{UN,EN}$	$E_D^{AV,EX}$	$E_D^{UN,EX}$
Expander	113.10	112.50	0.60	90.68	22.42	90.08	22.42	0.60	0.00
Pump	7.22	7.17	0.05	4.64	2.59	4.56	2.61	0.07	-0.02
Condenser	62.28	46.84	15.44	17.38	44.90	9.11	37.72	8.27	7.17
Evaporator	209.09	187.51	21.59	30.27	178.83	27.01	160.50	3.26	18.33
Overall System	391.70	354.02	37.68	142.97	248.73	130.77	223.25	12.20	25.48

Another important point observed from Table 11 is that a total amount of 3.75 kW could be lowered; taking into account that only the avoidable part of the exergy destruction rate can be reduced. This part of the exergy is higher than the unavoidable part in the expander (2.38 kW vs 0.59 kW) and the pump (0.12 vs 0.07 kW). These components will have the highest improvement potential by technical modifications of the components.

As the avoidable-endogenous part corresponds to the part of the exergy destruction rate, which can be reduced by increasing the efficient of the component, it will be the main focus. The avoidable-endogenous rate is higher than the unavoidable-endogenous rate in the expander and the pump. As stated before, technical modifications of these components will improve efficiency of the ORC system. Regarding the avoidable-exogenous rate, it is higher than the exogenous-unavoidable rate in the condenser. Therefore, an improvement in the efficiency of other components plays an important role in enhancing the efficiency of the condenser. The avoidable-endogenous part of the exergy destruction is higher than the avoidable-exogenous part in all the components. This difference is much higher in the expander, thus an optimization in this component will be essential to improve global ORC performance.

Results of splitting exergy destruction rate of the components are shown in Fig 7, Fig 8, Fig 9 and Fig 10. As a global consequence under the working conditions for the present work, there is a high potential of improvement in the ORC system, focusing in the expander. From the total exergy destruction rate in the expander (2.97 kW), the greater part of the exergy (88%) can be reduced by technological improvement of the component itself (avoidable-endogenous). In the pump, condenser and boiler this potential is reduced to 70%, 16% and 14% respectively.

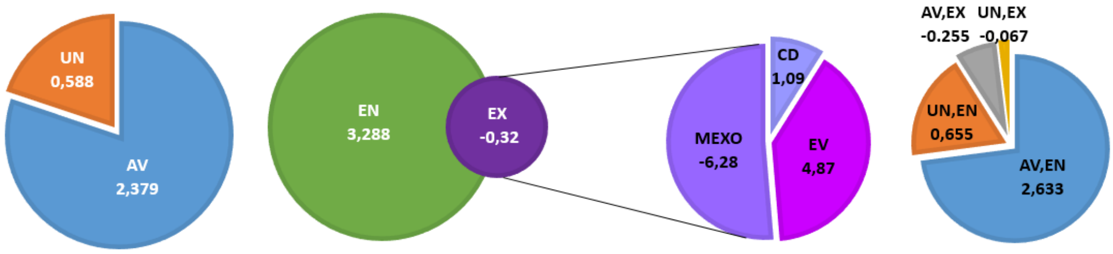

Fig 7. Results of splitting the exergy destruction rate for the expander

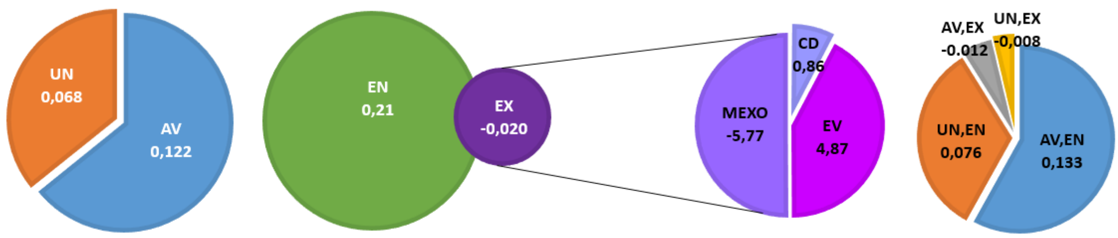

Fig 8. Results of splitting the exergy destruction rate for the pump

Fig 9. Results of splitting the exergy destruction rate for the condenser

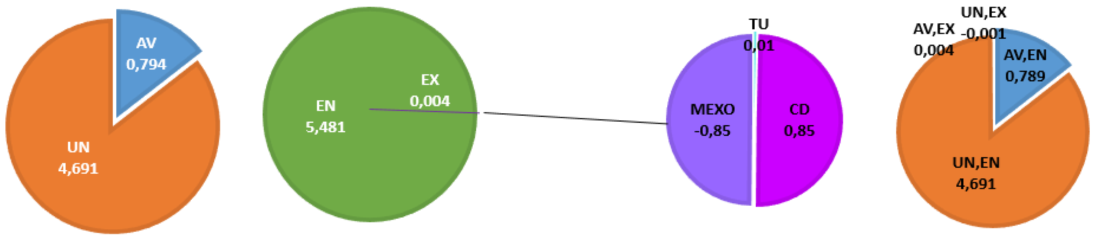

Fig 10. Results of splitting the exergy destruction rate for the boiler

5. Sensitivity analysis

The expander efficiency and the pinch point are the critical parameters of this cycle, therefore, a sensibility analysis varying both parameters is presented. Cycle efficiency as a function of both parameters is plotted in Fig 11. Moreover, blue lines correspond to boiler exchanged power and red ones to expander power in kW. The green point correspond to the unavoidable conditions.

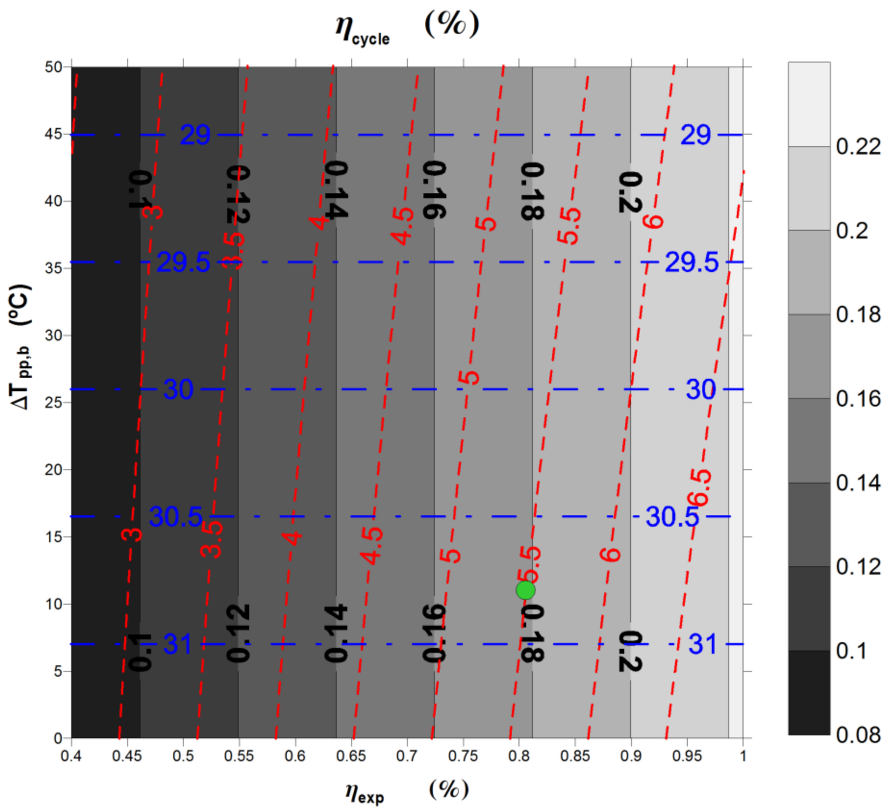

Fig 11. Cycle efficiency, boiler exchanged power (blue lines, kW) and expander power (red lines, kW) as a function of the pinch point and expander efficiency

As it is shown, the pinch point has a direct influence in the boiler power. The boiler power decrease with higher pinch points because the temperature difference in the boiler decreases. Regarding the expander power, it increases with higher expander efficiencies and lower pinch points. The former has an influence in the enthalpy drop in the expansion process and the latter on the ethanol mass flow in the cycle. The relation between both values give the cycle energy efficiency. A maximum value of 21% can be obtained in the best conditions of the cycle.

In order to discriminate the contributions of each component to the global exergy destruction rate, Fig 12 is presented. Blue lines correspond to the boiler, red ones to the expander, green ones to the condenser and pink ones to the exergy destruction rate of pump in kW. The sum of all components exergy destruction rate is plotted as a contour map behind them.

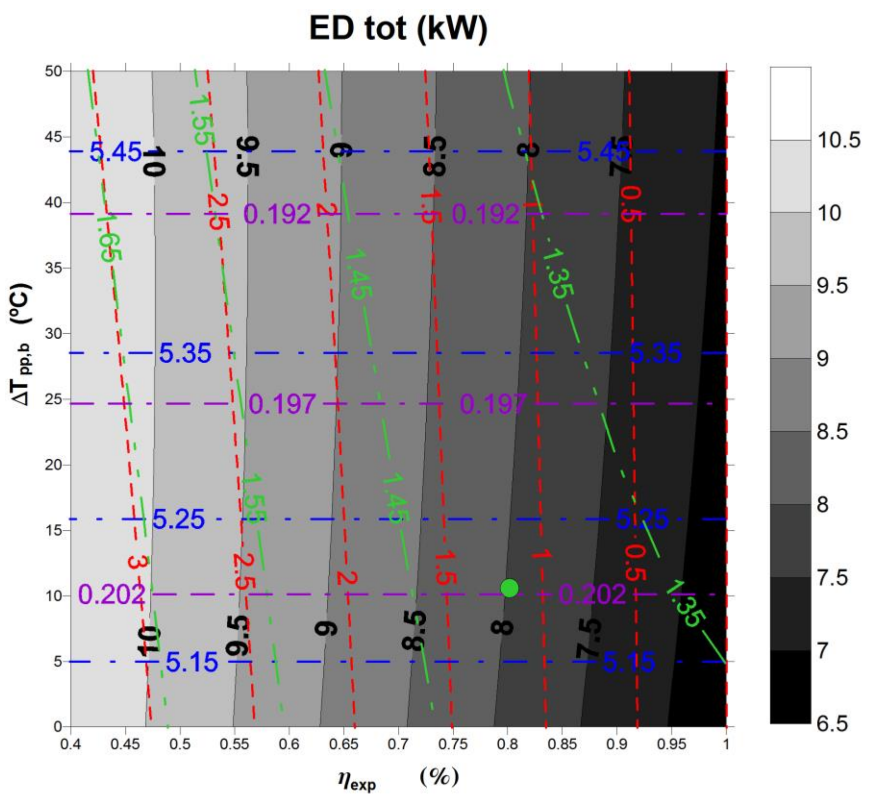

Fig 12. Contribution of boiler (blue lines, kW), expander (red lines, kW), condenser (green lines, kW) and pump (purple lines, kW) to the global exergy destruction rate (kW) as a function of the pinch point and expander efficiency.

As it can be shown in Fig 12, exergy destruction rate in the boiler and the pump depends on the pinch point. The former increases with pinch point and the latter decreases with it. Boiler exergy destruction rate increases because the area between ethanol evaporation process and the exhaust gases process increases too (Fig 5). As the temperature difference decrease with higher pinch points, the boiler power released to the cycle is lower and thus, the ethanol mass flow too. This is the reason why the exergy destruction in the pump decreases with the pinch point and remain approximately constant with expander efficiency. As stated before, increasing the expander efficiency will reduce the exergy destruction rate in the expander. The relation with the condenser depends on both parameters, the pinch point (and thus, the ethanol mass flow) and the expander efficiency. To sum up, reducing the pinch point in the boiler and increasing the expander efficiency will reduce the exergy destruction rate from 10.5 kW to 6.5 kW.

6. Conclusions

This paper evaluates and analyzes a bottoming ORC cycle coupled to an IC engine by means of conventional and advanced exergy analysis. The following results have been obtained:

1. Conventional analysis shows that the overall system should be improved following priorities for the components in this order: boiler, expander, condenser and pump. However, the advanced exergy analysis suggests that the first priority should be given to the expander, followed by the pump, condenser and boiler.

2. The value of endogenous exergy is greater than the value of exogenous exergy in all the system components. Therefore, the greatest contribution to the exergy destruction rate in each of the components comes from the internal irreversibility of the component itself and a minimum exergy destruction comes from other components as external irreversibility. Regarding to exogenous (external irreversibilities) exergy destruction, the condenser has the highest value (0.26 kW, 16% of the total exergy destruction rate). Therefore, a modification in the other component efficiencies can lead to a reduction in the exergy destruction rate of this component and an improvement in cycle efficiency.
3. A total amount of 3.75 kW, 36.5% of exergy destruction rate, could be lowered, taking account that only the avoidable part (considering an estimation of maximum efficiencies on the cycle components) of the exergy destruction rate can be reduced. This part of the exergy is higher than the unavoidable part in the expander (2.38 kW vs 0.59 kW) and the pump (0.12 vs 0.07 kW). These two components will have the highest improvement potential by technical modifications of the components.
4. Considering the sensibility analysis varying the pinch point from 0ºC to 50ºC in the boiler and the expander efficiency from 0.4 to 1 under the working conditions of the study, a maximum cycle efficiency of 21% can be obtained in comparison with 10% in the real conditions. Regarding the overall exergy destruction rate of the cycle, it could be lowered from 10.5 kW to 6.5 kW.

7. Acknowledgements

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