PERFORMANCE EVALUATION OF A COMBINED ORGANIC RANKINE CYCLE AND AN ABSORPTION REFRIGERATION SYSTEM

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Abstract. In this paper, exergy analysis is used to assess the exergetic performance of a solar powered Organic Rankine Cycle (ORC) and a LiBr Absorption Refrigeration System. The exergy efficiency and the exergy destruction rates are calculated for the whole combined system. The goal of this research is to highlight that this kind of systems is able to satisfy air-cooling and electrical power need for an existing building. Thus, cogeneration system using solar energy for combined cooling and electrical power in buildings is studied. The simulations for the ORC are performed for different working fluids and show that using R245fa as working fluid in the low-temperature solar Rankine cycle is suitable. For the LiBr absorption refrigeration system the coefficient of performance is calculated, as well as the influence of the inlet temperature in the Generator.
Keywords: Organic Rankine Cycle, Absoption Refrigeration System, solar system, combined cycles, exergy analysis.

1. INTRODUCTION

In the current economical and energetic context, implementing technologies using renewable energy as heating source is offering a double advantage: the reduction of pollution and of the fuel cost. One of the main concerns of the modern human is to provide comfort in buildings. The main utilities that make a building „alive" are: electricity, domestic hot water, heating/cooling according to external ambiance. In this study the attention is focused on providing electricity and cooling during summer for a public establishment.

The usual way of providing this two utilities to a building are to connect to local electrical network and to install an electric driven chiller. In this paper the performance of a combined cycles (production of electricity using an ORC and cooling load assured by a single-effect lithium bromide/water absorption system) is studied. The two systems work together and the energy that drives the combined system is represented by the solar energy transferred to steam of 140ºC. Using heat as an energy source offers the possibility to consider the sun as the fuel of the combined power - cooling system [1].

The whole system and each sub-system is analyzed according to first and second law of the thermodynamics. A computational model is developed in Thermoptim software in order to study the performance of the systems with different operating temperatures.

The ORC system is very similar to a steam Rankine cycle which uses an organic fluid instead of water in order to for generate electricity [2]. Organic Rankine Cycle is a promising process for conversion of low and medium temperature heat to electricity, which makes it suitable for solar applications. Organic fluids can operate at a much lower evaporation temperature and pressure than a conventional turbine using water as working fluid and still perform at a high efficiency [3-6]. The usual working fluids for an ORC are: HFC's Freon, ammonia, butane, iso-pentane, toluene and they have in general a high molecular mass. Several fluids were compared in previous work and it was found that R245fa presents interesting performances for an ORC solar system, although for a high pressure [4].

The solar fluid at the exit of the heat exchanger of the ORC is conducted into a second heat exchanger which is the generator of the absorption cooling system (ACS). In this way potential of the lower temperature heat is used in the ACS in order to produce cooling effect in the evaporator (Figure 1).

The reason for choosing $LiBr/H_2O$ as working fluid instead of ammonia/water pair is:

– it does not requires a rectification column and dephlegmator;

– it has a large latent heat of vaporization;

– the agent is non-toxic, non-flammable and non-explosive, so the installation does not require special supervision;

– system pressure is reduced, leading to a lower metal consumption, so the unit is compact

This reasons presented above makes $LiBr/H_2O$ solution more suitable for domestic cooling.

In this paper, the effect of varying the temperature of the generator over the system and over the components is studied. Also the computer simulation performed in EES showed in which components occur the highest exergy losses, with important influence over the performance of the whole system.

The simulation was run for different values of the generator temperatures, with a variation in the interval 70 – 100°C, considering an increase step of 5°C.

2. DESCRIPTION OF THE SYSTEM

The system is composed by three sub-systems: solar collector, Organic Rankine Cycle system and Absorption Cooling System.

The working fluid from ORC is pumped to a high pressure by the feed pump through the heat exchanger. The superheated vapour obtained at the outlet of the heat exchanger sets in motion the ORC turbine then the mechanical work is converted on electricity. The exhausted vapour from the turbine is directed to the condenser where is cooled by cooling water. A part of the heat received on the solar collector is used on the generator of the absorption cooling system and heats the solution $LiBr/H_2O$.

3. MATHEMATICAL METHOD

3.1. Organic Rankine Cycle

3.1.1. Energy analysis of the Organic Rankine Cycle

The energy and exergy analysis were performed taking into account the parameters from the table 1. Three principal assumptions are considered: the system reaches a steady state, and pipe pressure drop and heat losses to the environment in components of the ORC are neglected.

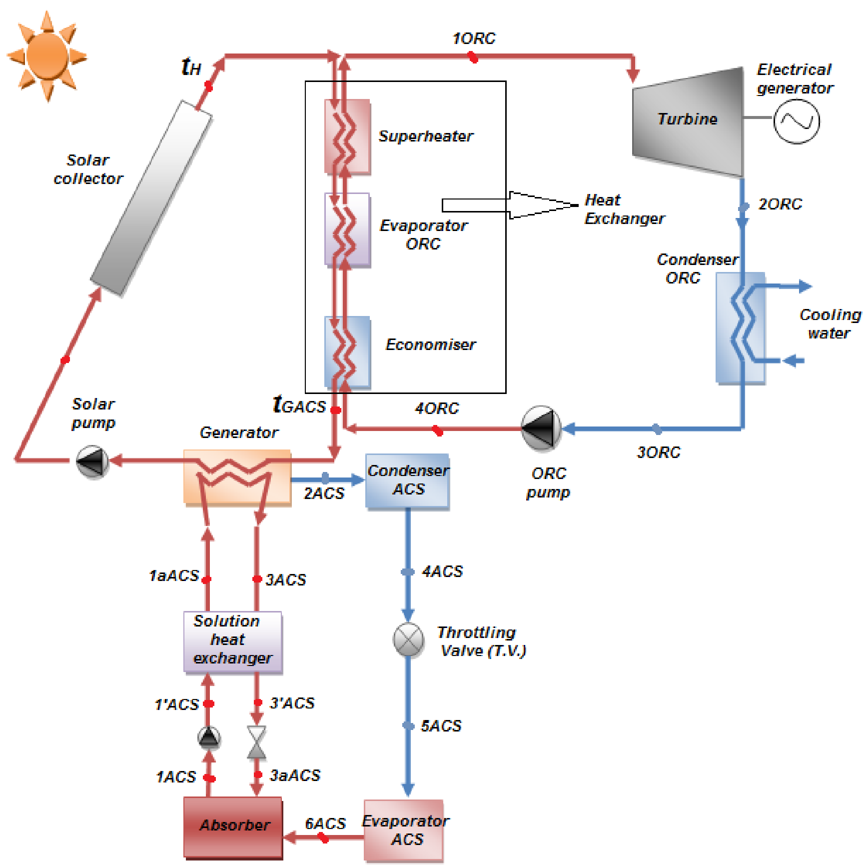

Fig. 1. Simplified scheme of the combined ORC and ACS system.

Table 1

Input parameters for the ORC

Parameters	Value	Unit
t_H	140	°C
$\Delta t_H = t_H - t_{1ORC}$	20	°C
$\Delta t_{solHE} = t_H - t_{GACS}$	40	°C
$\dot{m}_{sol}$	0.14	kg/s
Δp_{sol}	0.5	bar
η_T	85	%
T_{mw}	303.12	K

Equations used to perform the energy analysis are:

Heat exchanger:

$$\dot{Q}_{HE\_ORC} = \dot{m}_{ORC}(h_{1ORC} - h_{4ORC}) \quad (>0) \tag{1}$$

Turbine:

$$\dot{W}_{T\_ORC} = \dot{m}_{ORC}(h_{2ORC} - h_{1ORC}) \quad (<0) \tag{2}$$

Condenser:

$$\dot{Q}_{Cd\_ORC} = \dot{m}_{ORC}(h_{3ORC} - h_{2ORC}) \quad (<0) \tag{3}$$

Pump:

$$\dot{W}_{P\_ORC} = \dot{m}_{ORC}(h_{4ORC} - h_{3ORC}) \quad (>0) \tag{4}$$

Thermal efficiency is:

$$\eta_{ORC} = \frac{\left|\dot{W}_{T\_ORC}\right| - \dot{W}_{P\_ORC}}{\dot{Q}_{HE\_ORC}} \tag{5}$$

Some results are presented on Table 2, corresponding to parameters from Table 1.

Table 2

Installation performances for t_H = 140 °C

t_H	p_{1ORC}	$\dot{m}_{ORC}$	$\dot{W}_{ORC}$	$\dot{Q}_{HE\_ORC}$	η_{ORC}
[°C]	[bar]	[kg/s]	[kW]	[kW]	[%]
140	19.2015	0.103	2.78	23.779	11.69

3.1.2. Exergy analysis of the Organic Rankine Cycle

Method using exergy analysis is employed to evaluate the performance of the system as irreversibility is occuring in every component of the ORC system. Exergy is the maximum amount of work that can be done by a subsystem as it approaches thermodynamic equilibrium with its surrounding by a sequence of revesible processes. The exergy of a subsystem is a measure of its „distance" from equilibrium. Thus it can signify the quality of the energy of the subsystem. Exergy destruction can be obtained from the exergy balance. The exergy destruction of the pump process I_{P_ORC}, the evaporation process I_{HE_ORC}, the expansion process I_{T_ORC} and the condensation process I_{Cd_ORC}. Specific exergies for each point of the cycle were obtained by using Thermoptim Software [18].

Heat exchanger:

Fuel :

$$Cb_{HE\_ORC} = \dot{E}x_{Q_{HE}}^{T_{cold}} = \dot{Q}_{HE\_ORC}\left(1 - \frac{T_0}{T_{mH}}\right) \tag{6}$$

Product:

$$P_{HE\_ORC} = \dot{m}x_{ORC}(ex_{1ORC} - ex_{4ORC}) \tag{7}$$

Irreversibility:

$$I_{HE\_ORC} = Cb_{HE\_ORC} - P_{HE\_ORC} \tag{8}$$

with: $$T_{mH} = \frac{t_H - t_{GACS}}{\ln\left(\frac{T_H}{T_{GACS}}\right)} \tag{9}$$

$$\eta_{exHE\_ORC} = \frac{P_{HE\_ORC}}{Cb_{HE\_ORC}} \tag{10}$$

$$Ir_{HE\_ORC} = \frac{I_{HE\_ORC}}{\dot{E}x_{Q_{HE\_ORC}}^{T_{mH}}}100 \tag{11}$$

Turbine:

Fuel : $$Cb_{T\_ORC} = \dot{m}_{ORC}(ex_{1ORC} - ex_{2ORC}) \tag{12}$$

Product: $$P_{T\_ORC} = \dot{W}_{T\_ORC} \tag{13}$$

Irreversibility:

$$I_{T\_ORC} = Cb_{T\_ORC} - P_{T\_ORC} \tag{14}$$

$$\eta_{exT\_ORC} = \frac{P_{T\_ORC}}{Cb_{T\_ORC}} \tag{15}$$

$$Ir_{T\_ORC} = \frac{I_{T\_ORC}}{\dot{E}x_{Q_{HE\_ORC}}^{T_{mH}}}100 \tag{16}$$

Pump:

Fuel : $$Cb_{P\_ORC} = \dot{W}_{P\_ORC} \tag{17}$$

Product: $$P_{T\_ORC} = \dot{m}_{ORC}(ex_{4ORC} - ex_{3ORC}) \tag{18}$$

Irreversibility:

$$I_{P\_ORC} = Cb_{P\_ORC} - P_{P\_ORC} \tag{19}$$

$$\eta_{exP\_ORC} = \frac{P_{P\_ORC}}{Cb_{P\_ORC}} \quad (20)$$

$$Ir_{P\_ORC} = \frac{I_{P\_ORC}}{\dot{E}x_{Q_{HE\_ORC}}^{T_{mH}}} 100 \quad (21)$$

Condenser:

Fuel :

$$Cb_{Cd\_ORC} = \dot{m}_{ORC}(ex_{2ORC} - ex_{3ORC}) \quad (22)$$

Product: $P_{Cd\_ORC} = \dot{Q}_{Cd\_ORC}\left(1 - \frac{T_0}{T_{mw}}\right)$ (23)

Irreversibility:

$$I_{Cd\_ORC} = Cb_{Cd\_ORC} - P_{Cd\_ORC} \quad (24)$$

$$\eta_{exCd\_ORC} = \frac{P_{Cd\_ORC}}{Cb_{Cd\_ORC}} \quad (25)$$

$$Ir_{Cd\_ORC} = \frac{I_{Cd\_ORC}}{\dot{E}x_{Q_{HE\_ORC}}^{T_{mH}}} 100 \quad (26)$$

with $\dot{E}x_{Q_{HE\_ORC}}^{T_{mH}} = \dot{Q}_{HE\_ORC}(1 - \frac{T_0}{T_{mH}})$ (27)

The exergy efficency of ORC system is calculated using the equation here below:

$$\eta_{exORC} = \frac{\left|\dot{W}_{T\_ORC}\right| - \dot{W}_{P\_ORC}}{\dot{E}x_{Q_{HE\_ORC}}^{T_{mH}}} \quad (28)$$

3.2. Absorption Cooling System

3.2.1. Energy analysis of the Absorption Cooling System

In the past recent years, several experimental studies and simulations have been done in order to establish the performance of solar absorption cooling systems. Different scientist people studied the absorption system and components performance with the variation of different parameters: Kilic and Kaynakli, Kaushik and Arora and González-Gil et al. [7, 8, 9].

The simulation in EES (Engineering Equation Solver) starts from the parameters presented in Table 2.

Considering the input parameters, a first set of data is necessary to develop the simulation of the absorption chiller, and to calculate.

The mass flow rate of the chilled water in the evaporator is:

$$\dot{m}_{wACS} = \frac{\dot{Q}_{Ev\_ACS}}{c_p(t_{EviACS} - t_{EveACS})} = 2.175\,\text{kg/s} \quad (29)$$

Table 3

Input parameters for ACS

$\dot{Q}_{Ev\_ACS}$ **[kW]**	**45.6**
t_{EviACS} [°C]	12
t_{EveACS} [°C]	7
$t_{AbiACS}=t_{CdiACS}$[°C]	25
$t_{AbeACS}=t_{CdeACS}$[°C]	29

The temperatures of the working fluid in the key heat exchangers of the absorption refrigeration system are calculated such that:

$$t_{EvACS} = t_{EviACS} - \Delta t_{EvACS} = 4^\circ\text{C} \quad (30)$$

$$t_{CdACS} = t_{CdeACS} + \Delta t_{CdACS} = 33^\circ\text{C} \quad (31)$$

$$t_{AbACS} = t_{CdACS} = 33^\circ\text{C} \quad (32)$$

$$t_{GACS} = t_{GiACS} - \Delta t_{GACS} = t_{GiACS} - 8^\circ\text{C} \quad (33)$$

The parameter t_{Gi} was varying during the simulation in the range 70-100°C.

The saturation pressures corresponding to the temperatures T_{Ev} and T_{Cd}, are $p_{EvACS} = 0.00813\,\text{bar}$ and $p_{CdACS} = 0.05033\,\text{bar}$.

In order to simulate the absorption chiller, the mathematical model of the machine was built of a set of energy and mass balance equations with EES (Engineering Equation Solver). The equations that describe the heat capacities, the mass and energy balance for the various components of the absorption system are presented below [19]:

Evaporator:

$$q_{Ev\_ACS} = h_{6ACS} - h_{5ACS} \quad (34)$$

$$\dot{m}_{ef\_ACS} = \frac{\dot{Q}_{Ev\_ACS}}{\dot{q}_{Ev\_ACS}} \quad (35)$$

Condenser:

$$q_{Cd\_ACS} = h_{2ACS} - h_{4ACS} \quad (36)$$

$$\dot{Q}_{Cd\_ACS} = \dot{m}_{ef\_ACS} q_{Cd\_ACS} \quad (37)$$

Generator:

$$\dot{Q}_{G\_ACS} = \dot{m}_{ef\_ACS}(h_{2ACS} - h_{3ACS}) + \dot{m}_{p\_ACS}(h_{3ACS} - h_{1aACS}) \quad (38)$$

$$q_{G\_ACS} = \frac{\dot{Q}_{G\_ACS}}{\dot{m}_{r\_ACS}} \quad (39)$$

$$\dot{m}_{r\_ACS} = \dot{m}_{p\_ACS} + \dot{m}_{ef\_ACS} \quad (40)$$

Absorber:

$$q_{Ab\_ACS} = (h_{6ACS} - h_{3aACS}) + \\ + f(h_{3aACS} - h_{1ACS}) \quad (41)$$

$$\dot{Q}_{Ab\_ACS} = \dot{m}_{p\_ACS} q_{Ab\_ACS} \quad (42)$$

The solution flow ratio, (f) can be defined as the ratio of the mass flow rate of the solution through the pump to the mass flow rate of the working fluid. The flow ratio should be noted as it represents the required pumping energy. [10]

Coefficient of performance (*COP*) represents the measure of performance of a cooling machine and it is defined as:

$$COP_{ACS} = \frac{\dot{Q}_{Ev\_ACS}}{\dot{Q}_{G\_ACS} + \dot{W}_{P\_ACS}} \quad (43)$$

3.2.2. Exergy analysis of the Absorption Cooling System

The exergy analysis was performed over the absorption cooling system to point out the exergy destructions and the occurring malfunctions. Also each component of the system was individually studied, considering the exergy analysis tools. Similar studies were published by Sencan et al., Kaushik and Arora, and Zadeh and Bozgoran,[8][11][12]. Every component was studied from an exergetic point of view, and so, for every component there was associated: a Fuel (the exergetic resources supplied or the exergy potential at the beginning of the process), a Product (what offers the component exergetically) and the Irreversibilities, meaning the exergy consumed. The corresponding expressions for the Fuel, Product and Irreversibility of the system components are presented below for the generator. The relations for the other components of the system are similar to those for the generator, [13-16], [20].

The reference state to which we relate is the exergy of the water at $t_0 = 25$ºC and $p_0 = 101,325$ kPa.

$$P_{G\_ACS} = \dot{E}x_{2ACS} + \dot{E}x_{3ACS} = \\ = \dot{m}_{ef\_ACS} ex_{2ACS} + \dot{m}_{pACS} ex_{3ACS} \quad (44)$$

with $ex_{3ACS} = h_{3ACS} - T_0 s_{3ACS} - A_{ACS}$ (45)

and $A_{ACS} = h_{CdiACS} - T_0 s_{CdiACS}$ (46)

$$Cb_{G\_ACS} = \dot{E}x_{1aACS} + \dot{E}x_{Q_{G\_ACS}} = \\ = \dot{m}_{p\_ACS} ex_{1aACS} + \dot{E}x_{Q_{G\_ACS}} \quad (47)$$

with $ex_{1aACS} = h_{1aACS} - T_0 s_{1aACS} - A_{ACS}$ (48)

$$\dot{E}x_{Q_{G\_ACS}} = \dot{Q}_{G\_ACS}\left(1 - \frac{T_0}{T_{mG}}\right) \quad (49)$$

with $T_{mG} = \dfrac{t_{GiACS} - t_{GeACS}}{\ln\left(\dfrac{T_{GiACS}}{T_{GeACS}}\right)}$ (50)

$$\eta_{exG\_ACS} = \frac{P_{G\_ACS}}{Cb_{G\_ACS}} \quad (51)$$

$$I_{G\_ACS} = Cb_{G\_ACS} - P_{G\_ACS} \quad (52)$$

$$Ir_{G\_ACS} = \frac{I_{G\_ACS}}{\dot{E}x_{Q_{G\_ACS}}} 100 \quad (53)$$

We introduced the I_r parameter to indicate the irreversibility of each component, relative to the starting potential, meaning the exergy of the heat provided by solar panels to the generator.

$$\eta_{exACS} = \frac{\dot{E}x^{T_{ev}}_{Q_{EvACS}}}{\dot{E}x^{T_{mG}}_{Q_{G\_ACS}} + \dot{W}_{P\_ACS}} \quad (54)$$

Energy and exergy analysis of the global installlation allows to calculate the thermal efficiency:

$$\eta_{Global} = \frac{\dot{Q}_{Ev\_ACS} + \left|\dot{W}_{T\_ORC}\right| - \dot{W}_{P\_ORC}}{\dot{Q}_{G\_ACS} + \dot{Q}_{HE\_ORC}} \quad (55)$$

and the exergy efficiency:

$$\eta_{exGlobal} = \frac{\left|\dot{W}_{T\_ORC}\right| - \dot{W}_{P\_ORC} + \dot{E}x^{T_{ev}}_{Q_{Ev\_ACS}}}{\dot{E}x^{T_{mG}}_{Q_{G\_ACS}} + \dot{E}x^{T_{mH}}_{Q_{HE\_ORC}}} \quad (56)$$

4. RESULTS

The effects of turbine inlet temperature on the cycle performances are examined for the following assumptions:

- the temperature from the solar collector (t_H) is varing from 110 to 140°C;
- the ambient temperature is considered to be 25°C;
- the isentropic efficiencies for the turbine is assumed to be 85%.
- the high pressures inlet turbine are calculate function of the t_H [17]
- as a result, the average of ORC turbine power output is found to be 2.404 kW;
- the average Organic Rankine Cycle of η_{ORC} is 10.15 %.
- the COP_{ACS} reaches a maximum value at t_{GACS} =88 °C.

Table 4

The calculation of the exergy destructions for each component

tH	t_{GACS}	I_{AbACS}	I_{CdACS}	I_{GACS}	I_{ScACS}	I_{EvACS}	I_{Cd_ORC}	I_{T_ORC}	I_{HE_ORC}
[°C]	[°C]	[kW]	[kW]	[kW]	[kW]	[kW]	[kW]	[kW]	[kW]
110	70	2.9	1.34	1.54	1.08	0.95	1.076	0.316	0.46
115	75	2.63	1.35	1.55	0.59	0.95	1.079	0.373	0.5418
120	80	2.83	1.36	1.8	0.45	0.95	1.077	0.353	0.629
125	85	3.1	1.38	2.11	0.38	0.95	1.082	0.412	0.725
130	90	3.4	1.39	2.45	0.35	0.95	1.079	0.396	0.829
135	95	3.71	1.4	2.78	0.33	0.95	1.087	0.456	0.932
140	100	4.04	1.42	3.12	0.31	0.95	1.09	0.462	1.032

Table 5

Irreversibility relative versus the variation of the high temperature for every component

tH	t_{GACS}	Ir_{AbACS}	Ir_{CdACS}	Ir_{GACS}	Ir_{ScACS}	Ir_{EvACS}	Ir_{Cd_ORC}	Ir_{T_ORC}	Ir_{HE_ORC}
[°C]	[°C]	[-]	[-]	[-]	[-]	[-]	[-]	[-]	[-]
110	70	28.60	13.23	15.18	10.70	9.40	25.64	7.53	10.96
115	75	27.35	14.09	16.12	6.11	9.92	24.16	8.35	12.13
120	80	28.38	13.67	18.07	4.50	9.54	22.77	7.46	13.30
125	85	29.44	13.06	20.06	3.65	9.04	21.70	8.26	14.54
130	90	30.43	12.45	21.91	3.12	8.53	20.59	7.56	15.83
135	95	31.42	11.89	23.59	2.76	8.07	19.81	8.31	16.99
140	100	32.53	11.4	25.09	2.53	7.66	19.02	8.06	18

Table 6

Exergy efficiency for each component

t_H	P_{1ORC}	$\eta_{exAbACS}$	$\eta_{exCdACS}$	η_{exGACS}	$\eta_{exScACS}$	$\eta_{exEvACS}$	$\eta_{exCdORC}$	η_{exTORC}	$\eta_{exHEORC}$
[°C]	[bar]	[-]	[-]	[-]	[-]	[-]	[-]	[-]	[-]
110	10.044	0.830	0.255	0.940	0.573	0.722	0.247	0.866	0.89
115	11.274	0.714	0.254	0.909	0.589	0.722	0.246	0.854	0.878
120	12.614	0.609	0.252	0.880	0.600	0.722	0.245	0.871	0.866
125	14.069	0.523	0.250	0.855	0.607	0.722	0.244	0.858	0.854
130	15.648	0.452	0.248	0.831	0.611	0.722	0.243	0.871	0.842
135	17.357	0.394	0.246	0.811	0.613	0.722	0.241	0.858	0.83
140	19.2015	0.346	0.244	0.793	0.613	0.722	0.24	0.863	0.819

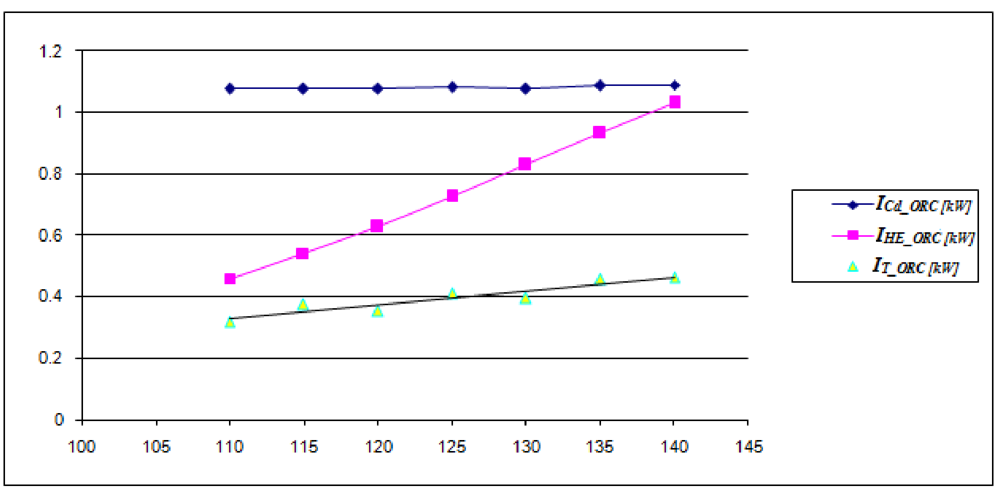

Fig. 2. Variation of I_{Cd_ORC}, I_{HE_ORC} and I_{T_ORC} versus t_H.

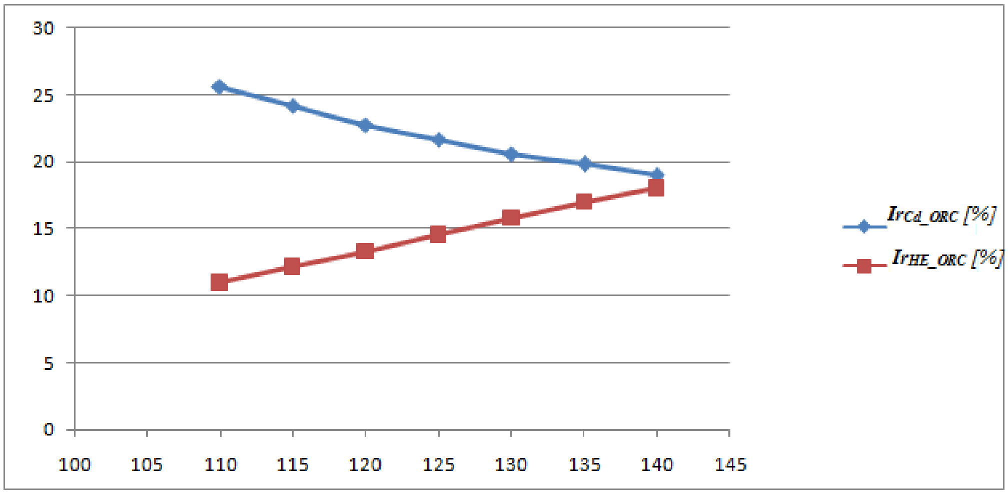

Fig. 3. Variation of Ir_{Cd_ORC} and Ir_{HE_ORC} versus t_H.

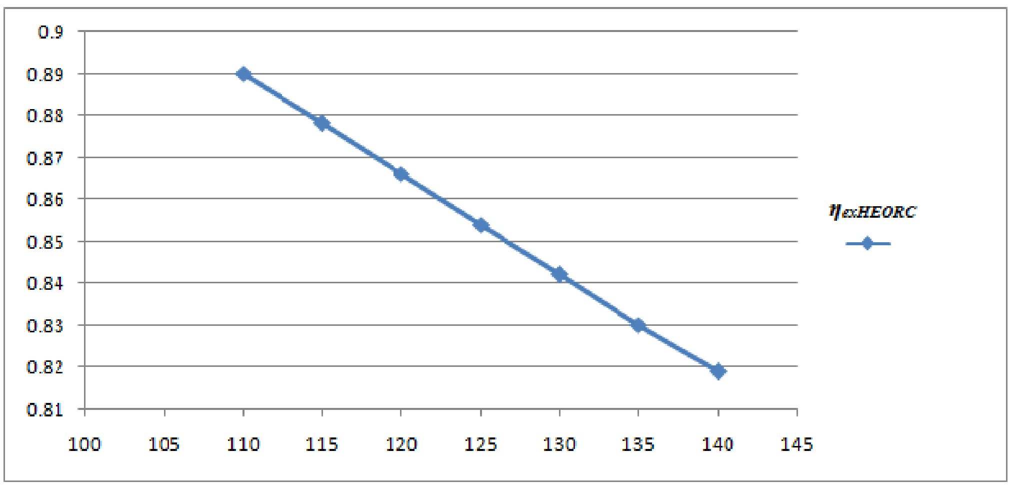

Fig. 4. Variation of $\eta_{exHEORC}$ versus t_H.

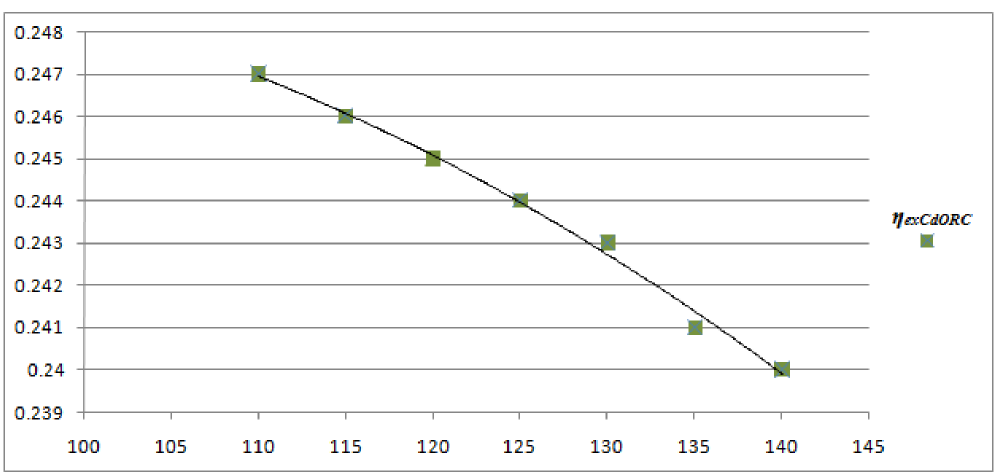

Fig. 5. Variation of $\eta_{exCdORC}$ versus t_H.

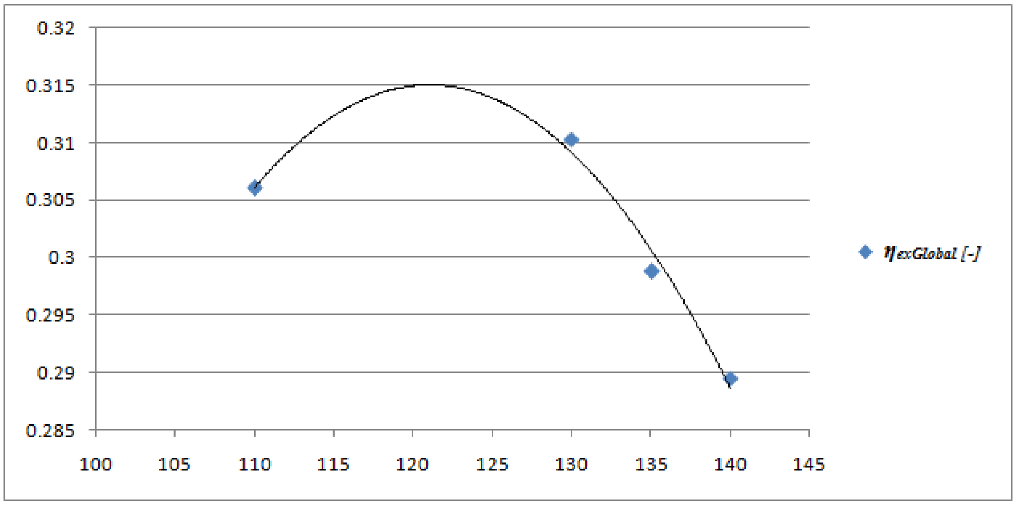

Fig. 6. Variation of $\eta_{exGlobal}$ versus t_H.

The increase of t_{GACS} has as result the decrease of the exergy efficiency of the Absorption Cooling System (η_{ex_ACS}) because of the increasing of the exergy destruction rate in every component. Considering a constant cooling power, the value of the COP_{ACS} is increasing with the increasing of the generator temperature until an optimal value around 0.78. Further increasing the temperature over 88°C, the COP_{ACS} does not increase, but the irreversibility in each component as the heat losses is increasing.

The difference between the generator temperature and the inlet temperature of the solution LiBr from the vapour generator is increasing and implies an increase in the irreversibility for ACS sub-system (Table 4).

Reduced irreversibility, I_r, was introduced on table 5, parameter that indicates the irreversibility of each component of the system, relative to the starting potential exergy at the beginning of the process, that means the exergy of the heat provided by the generator of the Absorption Cooling System and by the heat exchanger of the ORC).

The energy and exergy losses in the ORC pump are insignificant compared to those on the other components from ORC system. Only the components with high exergy destruction rates are presented in table 4, 5 and 6.

Table 6 shows that the exergy efficiency for every component is decreasing with the increasing of the temperature because of the increasing of the irreversibilities. The exergy efficiency of the ORC turbine seems to be constant, about 86 %.

The increasing irreversibility in the condenser does not an important impact therefore the exergy efficiency is decreasing because the Product of this component is constant as its Fuel has a small increasing (figures 2 and 5). Or, it is interesting to remark that, as it is shown on figure 3, reduced irreversibility on condenser $\boldsymbol{Ir_{Cd\_ORC}}$ is decreasing even if the exergy efficiency and local irreversibility are increasing. This means that the increasing of the exergy flow rate on the ORC heat exchanger is more important than the increasing of irreversibility in condenser and finally it is interesting to increase the high temperature of the system from this point of view.

Variation on heat exchanger irreversibility is significant; the exergy efficiency is decreasing having as consequence a growth of relative irreversibility in a range 10 and 18% (figures 3 and 4). Figure 6 indicates that $\boldsymbol{\eta_{exGlobal}}$ has an optimal value for t_H value between 115 and 120°C

5. CONCLUSION

A low-temperature solar Organic Rankine Cycle using R245fa as working fluid coupled with a LiBr/H_2O absorption refrigeration system is studied. In this paper the exergy analysis was applied by using the first and the second law of thermodynamics in order to indicate an optimum design and operation of this installation.

The ORC efficiency could be improved by increasing the maximum temperature in the cycle, but in the same time it is shown that irreversibilities occurring in each component are increasing.

For improving the performances of the ORC system, it is necessary to recover the condensation heat in the system because the most destruction rates are occurring in the condenser, table 4. It can be improved the Organic Rankine Cycle.

The global exergy efficiency decrease is due also to the increase of the irreversibilites at higher temperatures of the generator (Absorption Cooling System). Irreversibilites of absorber, generator and the ORC condenser represent a large share of total exergy loss.

The simulation model of this installation shows that performance of an ORC system using R245fa as working fluid is correct and can be improved by introducing internal heat exchangers and by a better heat transfer between hot and cold fluids of the system, which is the aim of a further work using the pinch analysis.

Nomenclature

A	heat exchanger area,[m^2]
a_1, a_2	thermal loss coefficient,[W/m^2K]
COP	coefficient of performance, [-]
Cb	fuel (resource), [kW]
c_p	specific heat at constant pressure, [$J\ kg^{-1}K^{-1}$]
ex	specific exergy, [$kJkg^{-1}$]
$\dot{E}x$	exergy flow rate, [kW]
f	solution flow ratio, [-]
h	specific enthalpy, [$kJkg^{-1}$]
I	exergy destruction, [kW]
Ir	reduced exergy destruction, [-]
L	heat exchanger length
$\dot{m}$	mass flow rate, [$kg\ s^{-1}$]
p	pressure, [bar]
P	product, [kW]
q	heat capacity, [$kJkg^{-1}$]
$\dot{Q}$	heat flow rate, [kW]
s	specific entropy, [$kJkg^{-1}K^{-1}$]
t	temperature, [°C]
T	temperature, [K]
U	heat transfer coefficient[$Wm^{-2}K^{-1}$]
$\dot{W}$	mechanical power, [W]
X	solutions concentration in refrigerant, [%]

Greek letters

Δp	pressure losses, [bar]
Δt	temperature difference, [°C]
η	efficiency, [-]

Subscripts

Ab	absorber
Abe	cooling water exit from absorber
Abi	cooling water inlet to absorber
ACS	absorption cooling system
Cd	condenser
Cde	cooling water exit from condenser
Cdi	cooling water inlet to condenser
Ev	evaporator
Eve	evaporator exit of the cooled water
Evi	evaporator inlet of the cooled water
Eg	cooled water
ef	water as refrigerant
ex	exergy
G	Generator
Ge	water inlet for solar collector
Gi	water exit from solar collector
H	high temperature/pressure
HE	heat exchanger
m	log mean temperature
ORC	Organic Rankine Cycle
P	weak solution in refrigerant
P	pump
R	strong solution in refrigerant
sol	solar fluid
t	turbine
T.V.	throttling valve
w	water
0	environmental state
1-6	state points
TOT	total

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