Research Article
Performance Analysis and Optimization of an Organic Rankine Cycle Coupled to a Fresnel Linear Concentrator for Various Working Fluids

[1]Sidi Bouhamady, [1]Ababacar Thiam, [2]El Bah Menny and [1]Dorothé Azilinon
[1]Laboratoire d'Energétique Appliquée, Ecole Supérieure Polytechnique de Dakar, BP 5085 Sénégal
[2]Faculté des Sciences et Techniques, Université de Nouakchott, BP 880, Nouakchott, Mauritanie

Abstract: The aim of this study is to model, optimize and analyze the performance of a micro power solar of 3 kW in order to meet the priority needs of Mauritania rural people in electric power and contribute to the reduction of greenhouse gases. The proposed system is composed of a Fresnellinear concentrator which is coupled to an Organic Rankine cycle with a regenerator. Several working fluids (R134a, R152a, R290 and R717) were used for better optimizing the system. Thus, many thermodynamic and physical parameters which are influent on the performance of the overall system were analyzed. This analysis shows that the R152a and R134a fluids are the best candidates for the applications of ORC solar at low temperatures. Indeed, a 5°C superheating, the overall system efficiency, the energy and exergy efficiency of the ORC cycle has been respectively improved to 7.7%, 16.1% and 14.4%. The Optimization made on the overall system allowed a 26% and 12% reduction of the surface of the concentrator and the volume flow out of the micro-turbine respectively. The minimum surface and volumetric flow rate required to produce 3 kW is respectively 21.25 m^2 and 21.421 m^3/h, this last result is achieved from the operating conditions: working fluid R152a, evaporation temperature which is 90°C and the direct normal radiation 1800 W/m^2.

Keywords: Areas rural, EES (engineering equation solver), energetic efficiency, fresnel linear concentrator, optimisation, organic rankine cycle, working fluids

INTRODUCTION

The lack of energy means the fact of not having access to necessary energy services. Most people in developing countries are affected by this tragic phenomenon especially people from rural areas. Actually, the basic needs (Niez, 2010) are lighting, cooking energy, heating and cooling system. According to the Niez (2010), about 22% of the world's population have no access to electricity and 85% of these people live without electricity in rural developing countries areas and the majority of these people are in sub-Saharan Africa and South Asia. The same way as the African countries, Mauritanian rural area was only 3.9% electrified in 2015. However, there are several renewable energy solutions such as the solar power, geothermal power, biomass and the rejected industrial heat, all these can be exploited to meet the priority needs of these populations and also reduce emissions of greenhouse gas. According to Solar Gis Mauritania has respectively a global horizontal and direct normal sunshine average of 2300 kWh/m^2 and 2000 kWh/m^2 and this proves once again that solar powers plants are good opportunities to increase access to electricity in

these remote areas. According to the works of Kuo *et al*. (2009) the efficiency of photovoltaic modules commercially available is lower of 10 to 15% than that of solar thermal modules which means that CSP technology is more competitive than photovoltaic technology. The conversion of these solar resources into electricity is not economically profitable by the conventional Rankine cycle because of their low temperature heat (Chen *et al*., 2010). Among the various energy conversion cycles at low temperature, the organic Rankine cycle is however, less complex and easily maintained with a competitive cost of investment contrary to the other cycles (Chen *et al*., 2010).

The performance and economic profitability of an organic Rankine cycle solar are related to the thermo-physical properties of the working fluid (Bruno *et al*., 2008; Stijepovic *et al*., 2011). Analyzed the relationship between the properties of working fluids and the economic and thermodynamic profitability of an ORC solar cycle from a theoretical and analytical point of view. Their results showed that poor choice could lead to a less efficient and costly cycle. Several authors have studied the performance of different working fluids in order to select the optimum working fluid for the

Corresponding Author: Sidi Bouhamady, Laboratoire d'Energétique Appliquée, Ecole Supérieure Polytechnique de Dakar, BP 5085 Sénégal

organic Rankine cycle solar (Wang *et al.*, 2011). Analyzed the performance of different working fluids as a function of the operating conditions, in particular the evaporation pressure and the condensation temperature. Chen *et al.* (2010) summarized the selection criteria for working fluids and studied the influence of these properties on the performance of the ORC solar (Bertrand *et al.*, 2008). Analyzed the thermodynamic characteristics and performance of 20 working fluids of a low temperature ORC cycle. Fluids with a critical temperature above 75°C were studied (Wang *et al.*, 2012). Studied 13 working fluids for an ORC cycle in order to optimize the total exchange surface of the exchangers. Studied the performance of an ORC solar system for electricity production using working fluids with low Global Warming Potential (GWP).

This study was interested in standard ORC solar system i.e., without recuperator. In recent years ORC solar system with recuperator have been developed in order to increase the performance of ORC systems. Several authors have studied the performance of different working fluids for this type of system (Darvish *et al.*, 2015). Compared the energy and exergy performance of several working fluids for an ORC cycle with recuperator (Desideri *et al.*, 2016). Evaluated experimentally and compared the performance of different working fluids in an ORC solar system with recuperator. Fluids with a critical temperature greater than 150°C were studied. To our knowledge, the performances of working fluids whose critical temperature is less than 150°C of an ORC solar system with recuperator have never been studied. The aim of this work is to study the performance and optimization of a microcentral solar power plant of 3 with recuperator kW,working with critical temperature working fluids ranging from 80 °C to 130 °C.
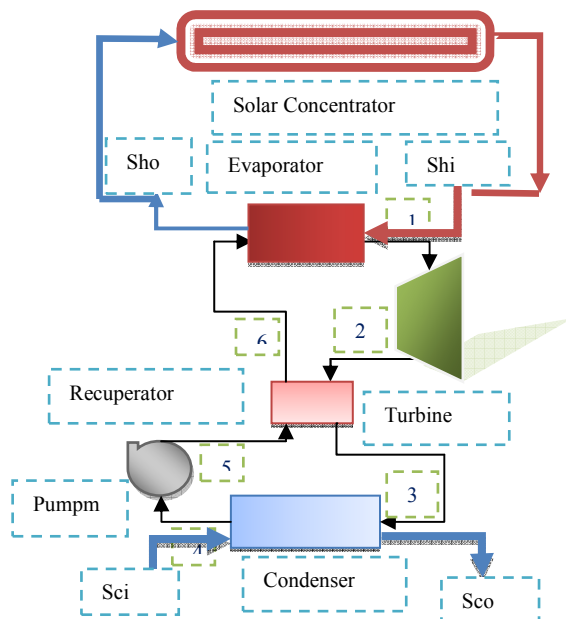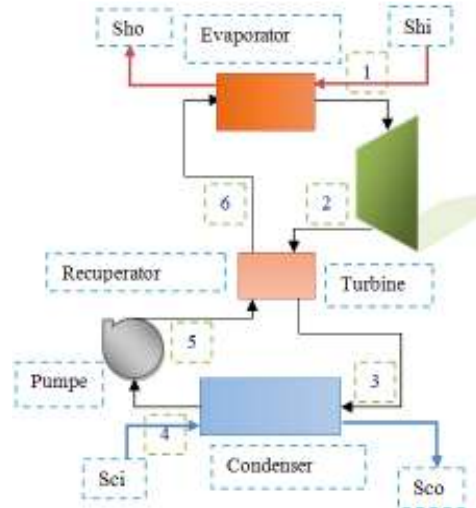

Fig. 1: ORC solar system

Fig. 2: ORC system

MATERIALS AND MODELS

Materials:

System description: This present ORC solar system is made of a linear Fresnel concentrator, a diaphragm type pump 4-5 and three heat exchangers: Evaporator (6-1), condenser (4-3) and recovery (3-2) and a micro-turbine/Scroll compressors, as shown in Fig. 1. Indeed, the micro-turbine illustrated here is similar to that proposed in the study of Quoilin *et al.* (2008) and Lemort (2006). The functioning scheme is as following: the pump supplies the working fluid to the evaporator where it is heated and vaporized by the heat transfer fluid, which is synthetic oil, coming from the solar collector. The enthalpy of the high-pressure steam produced in the micro-turbine is then converted into work. The Low pressure steam at the outlet of the micro-turbine is directed to the condenser where it is liquefied for starting a new cycle. The whole process (Fig. 2) described above is shown in the diagram T-s in Fig. 3 and 4. The thermodynamic modeling of the ORC cycle was performed using the energy balance and mass for four shortlisted working fluids. The theoretical performances and the thermodynamic and environmental properties of different preselected fluids were evaluated and compared under certain assumptions. This work also studies the performance of the ORC model coupled to a Fresnel linear concentrator.

Models:

Model of the ORC cycle: The various components of the ORC cycle are open systems that exchange matter permanently. Each cycle component is associated with a corresponding control volume. The application of the principle of conservation of mass and energy of a volume control between time t to t + Δtcan be reduced to the following equations:
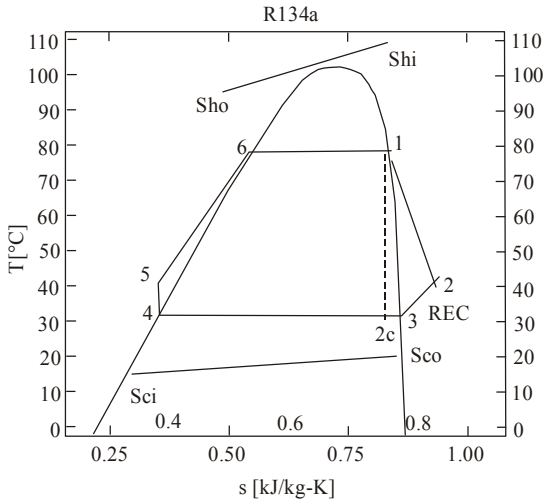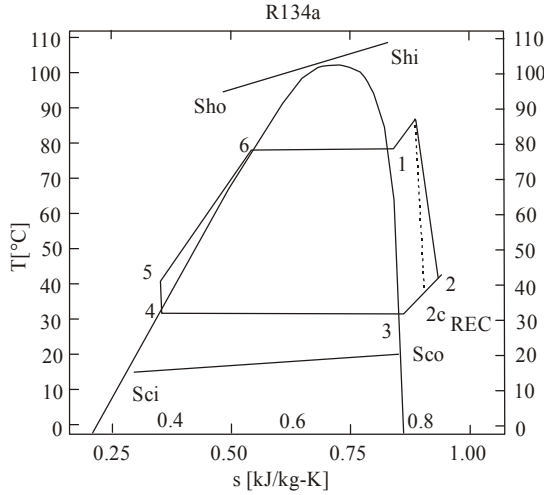
Fig. 3: ORC cycle without overheating

Fig. 4: ORC cycle with overheating

$$\begin{cases} \sum_e \dot{m}_e + \sum_s \dot{m}_s = 0 \\ \dot{Q} + \dot{W} + \sum_{e,s} \dot{m}_{e,s}\left(h + \frac{1}{2}V^2 + gz\right) = 0 \end{cases} \quad (1)$$

Most of the thermal machines in permanent diet with unidirectional flow and changes in kinetic and potential energy are often negligible; the Eq. (1) is simplified yet:

$$\dot{Q} + \dot{W} + \dot{m}(h_e - h_s) = 0 \quad (2)$$

Equation (2) is particularly used in the study of ORC cycle. Indeed, with the components that exchange only heat with the outside environment, $\dot{W} = 0$ and (2) become:

$$\dot{Q} = \dot{m}(h_s - h_e) \; si \; \dot{W} = 0 \quad (3)$$

Contrary to that, for the adiabatic transformations where there will be production of mechanical energy, $\dot{Q} = 0$ and (3) become:

$$\dot{W} = \dot{m}(h_s - h_e) \; si \; \dot{Q} = 0 \quad (4)$$

The equation of the destruction of exergy flow or(flow of irreversibility) ORC cycle for a steady state can be expressed as follows:

$$\dot{I} = \dot{T}_0\left(\sum \dot{m}_e \dot{s}_e - \sum \dot{m}_\iota \dot{s}_\iota - \sum \frac{\dot{Q}_k}{T_k}\right) \quad (5)$$

For the supposed adiabatic components, this equation can be reduced to:

$$\dot{I} = \dot{T}_0(\sum \dot{m}_e \dot{s}_e - \sum \dot{m}_\iota \dot{s}_\iota) \quad (6)$$

Energy and exergy equations of ORC cycle: For both adiabatic components considered here, (the) pump and the turbine, work exchanged with the outside which is given by Eq. (4).

Relaxation in the micro-turbine:

$$\dot{W}_t = \dot{m}(h_1 - h_2)\eta_{mt} \quad (7)$$

Flow of irreversibility (exergy destroyed in the micro-turbine):

$$\dot{I}_t = T_0 \dot{m}(s_2 - s_1) \quad (8)$$

Mechanical pump power:

$$\dot{W}_p = \dot{m}(h_5 - h_4)/\eta_p = \dot{m}v_4(P_5 - P_4)/\eta_p \quad (9)$$

Flow irreversibility (Exergy destroyed in the pump):

$$\dot{I}_p = T_0 \dot{m}(s_5 - s_4) \quad (10)$$

For both heat transfer components, there is no work exchange with the outside so that the equation is used (3) for the amount of heat exchanged with the outside:

Thermal power received by the working fluid (evaporator):

$$\dot{Q}_e = \dot{m}(h_1 - h_5) \quad (11)$$

Flow of irreversibility (The energy conveyed between the inlet and the outlet of the evaporator):

$$\dot{I}_e = T_0[\dot{m}_h(sh_0 - sh_i) - \dot{m}(s_5 - s_1)] \quad (12)$$

Thermal power rejected the cold sink (condenser):

$$\dot{Q}_c = \dot{m}(h_3 - h_4) \quad (13)$$

Flow of irreversibility (The energy conveyed between the inlet and the outlet of the condenser):

$$\dot{I}_e = T_0[\dot{m}_c(sc_0 - sc_i) - \dot{m}(s_4 - s_3)] \quad (14)$$

Table 1: Correction factor of the optical performance of NOVA-1 sensor

$\Theta_{\perp \text{ où } //}$	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°
Transverse $K_\perp$	1.00	0.98	0.95	0.95	0.91	0.86	0.7	0.48	0.24	0.00
Longitudinal $K_{//}$	1.00	0.98	0.92	0.83	0.7	0.53	0.33	0.12	0.01	0.00

Thermal power regained in recovery:

$$\dot{Q}_r = \dot{m}(h_2 - h_3) \tag{15}$$

Flow of irreversibility (The energy conveyed between the inlet and the outlet of the recuperator):

$$\dot{I}_r = T_0 \dot{m}[(s_6 + s_3) - (s_2 + s_5)] \tag{16}$$

Determination of ORC cycle performance: ORC system performance is determined by the following equations:

Mechanical power:

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_p \tag{17}$$

Thermal efficiency (performance under the first law of thermodynamics):

$$\eta_I = \frac{\dot{W}_{net}}{\dot{Q}_e} \tag{18}$$

The total reversibility of the cycle:

$$\dot{I}_{tot} = \sum_\iota \dot{I}_\iota = \dot{I}_t + \dot{I}_e + \dot{I}_c + \dot{I}_p + \dot{I}_r \tag{19}$$

Performance under the second law of thermodynamics:

$$\eta_{II} = \frac{\eta_I}{(1 - {T_c}/{T_h})} \tag{20}$$

$T_0, T_c \, et \, T_h$ Are the reference temperatures the cold source and hot source respectively. The flow volume of output of the turbine Vt2 determines the size of the turbine and the influence on the system cost. Therefore, the working fluids with low volume flow are preferred for economic reasons:

$$V_{t2} = \frac{\dot{m}}{\rho_2} \tag{21}$$

Here, m˙ and ρ2 are the mass flow rate and volumetric mass at the point of state two.

Model optical of concentrator and the linear thermal receiver: The solar collector considered for this study was modeled by a performance-based model NOVA-1 developed by the company Solar Novatec. The overall performance of the sensor is the product of the optical concentrator performance by the thermal efficiency of linear receiver:

$$\eta_c = \eta_{opt}.\eta_{th} \tag{22}$$

The model of the optical performance, which is defined as follows, models the optical concentrator:

$$\eta_{opt} = \eta_0.K_\perp(\Theta_\perp).K_\parallel(\Theta_\parallel) \tag{23}$$

η_0 is the maximum optical efficiency of the concentrator. In this hub, this yield is 0.67. $K\perp$ and $K \, //$ are factors of corrections, the first is called transversal and the other one is called longitudinal. These factors are respectively based on a transverse angle ($\Theta\perp$) and a longitudinal angle ($\Theta \, //$) and are determined from Table 1.

The linear receiver is modeled as follows:
The thermal efficiency of the linear receiver is defined by the Eq. (20):

$$\eta_{th} = 1 - \frac{P_{loss}}{DNI.\eta_{opt}} \tag{24}$$

P_{loss} is the dissipated power by the receiver and DNI is the direct normal radiation.
The dissipated heat power Ploss is calculated by the following formula:

$$P_{loss} = u_0.\Delta T + u_1 \Delta T^2 \tag{25}$$

With u_0 and u_1 are coefficients which give respectively to 5.6 $10^{-2} W.m^{-2}.K^{-1}$ 2.13 $10^{-4} W.m^{-2}.K^{-2}$ and DT is the temperature difference between the coolant temperature and the temperature of outside air.

The global model of the Solar Organic Rankine Cycle (SORC): The coupling of solar collector model and the model of the ORC machine allows one hand, to calculate the thermal power transported by the coolant and on the other hand, to evaluate the surface of the hub while meeting the operating conditions nominal ORC machine.
Thermal power carried by the coolant fluid:

$$Q_{th} = DNI.S_c.\eta_c \tag{26}$$

Here, Sc is the total area of concentrator mirrors and the DNI isthe direct normal radiation.
By neglecting losses the load between the receiver and the evaporator, we can write:

$$Q_{th} = Q_e \tag{27}$$

From Eq. (22) and (23) the total area of the hub is determined by the formula:

Table 2: Physical properties, environmental data of working fluids pre-selected

Fluid	T_{cri} (°C)	P_{cri} (MPa)	P_{max} (MPa)	P_{min} (MPa)	PR	Security Group *	ODP	GWP (100 ans)
R290	96.68	4.247	2.85	1.218	2.34	A3	0	~20
R152a	113.3	4.520	2.108	0.794	2.65	A2	0	124
R717	132.3	11.333	3.709	1.351	2.74	B2	0	<1
R134a	101	4.059	2.366	0.887	2.66	A1	0	1430

Table 3: ASHRAE classification of the working fluid* (Facão and Oliveira, 2009)

Flammable increase		A3		B3
		A2		B2
		A1		B1
		Increased toxicity		

Table 4: Performance of ORC solar system without and with overheating for various working fluids

Fluid	Overheating (°C)	P_{max} (MPa)	$\dot{m}$ (kg/s)	η_I (%)	η_{II} (%)	Rg (%)	I_{tot} (kW)	Q_e (kW)	V_{t2} (m³/h)	Sc (m²)
R290	0	2.85	0.155	5.52	9.81	3.80	13.4	54.34	23.33	34.3
R152a		2.108	0.181	5.53	9.86	3.81	13.06	54.17	30.48	34.19
R717		3.709	0.045	5.77	10.27	3.98	12.52	51.93	16.41	32.78
R134a		2.366	0.296	5.40	9.61	3.72	13.56	55.49	28.31	35.02
R290	5	2.85	0.143	6.17	10.98	4.04	12.6	48.58	20.66	32.25
R152a		2.108	0.165	6.26	11.14	4.04	11.88	47.86	26.59	31.99
R717		3.709	0.040	6.62	11.75	4.27	11.25	45.37	14.14	30.51
R134a		2.366	0.269	6.05	10.77	3.96	12.6	49.51	24.81	32.92

$$S_c = \frac{Q_e}{DNI.\eta_c} \qquad (28)$$

Overall performance of the solar ORC model: The overall yield of the ORC solar systemis calculated as follows:

$$\eta_{SORC} = \eta_{ORC}.\eta_c \qquad (29)$$

η_{SORC} is the performance of the first law of thermodynamics the ORC cycle. For simplicity reasons, several reasonable assumptions were implemented by thermodynamic modeling. These assumptions are listed as following:

- Evaporation temperature 80°C
- Condensing temperature 30°C
- Mechanical micro-turbine Yield 63%
- Isentropic efficiency of the micro-turbine 70%
- Pump Efficiency 80%
- Reference temperature 25°C

The Synthetic oil at 80°C is provided by a Fresnel linear concentrator. The condenser is cooled by air ambient in a surrounding area and the whole is supposed to be located in a hot rural area in Mauritania where the monthly average ambient temperature is about 30°C. The source temperature may vary between 60°C and 110°C (Table 2 and 3).

RESULTS AND DISCUSSION

In total, four pure organic working fluids were chosen as potential candidates, as shown in Table 1.

Table 4 lists the performance results of the ORC solar system for a power of 3 kW.

Overall performance of the ORC solar system: Thermal efficiency characterizes the performance of the ORC machine. Figure 5 shows that when the inlet temperature of the turbine increases the thermal efficiency also increases. From Table 4 it varies between 5.4% (R134a) to 5.77% (R717). Figure 6 shows that when the inlet temperature of the turbine increases the energy efficiency also increases. The exergy efficiency of the ORC cycle assesses the potential to recover the noble energies and the level of irreversibility of the different components. A maximum exergetic efficiency of 10.27% was obtained by using
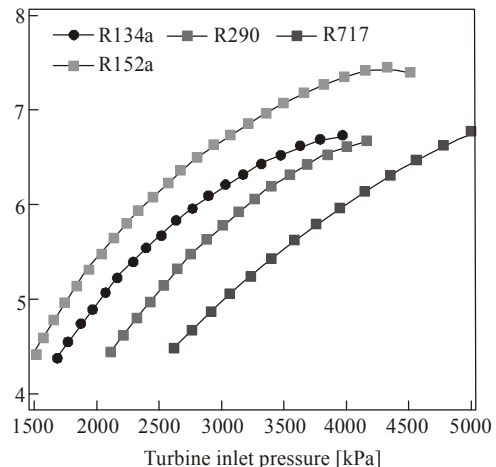

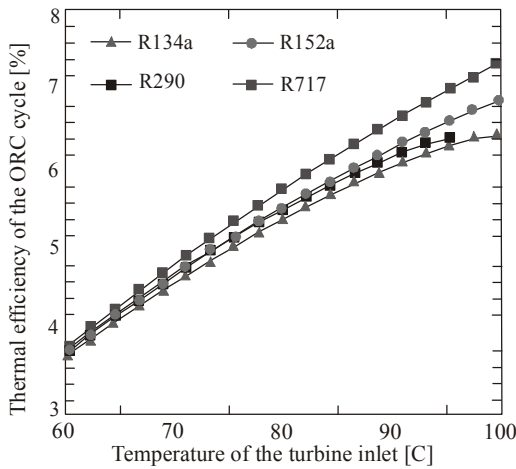Fig. 5: Thermal efficiency vs the turbine inlet pressure

Fig. 6: Thermal efficiency vs the turbine inlet temperature for various working fluid
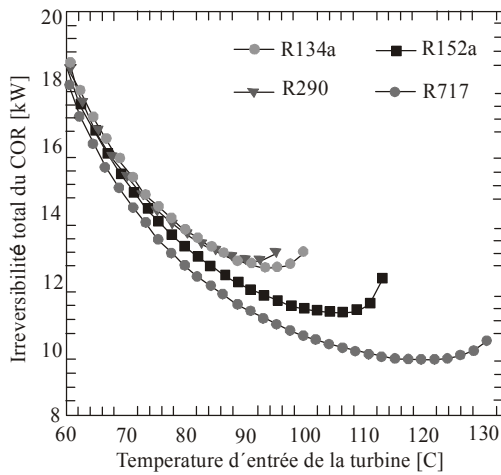

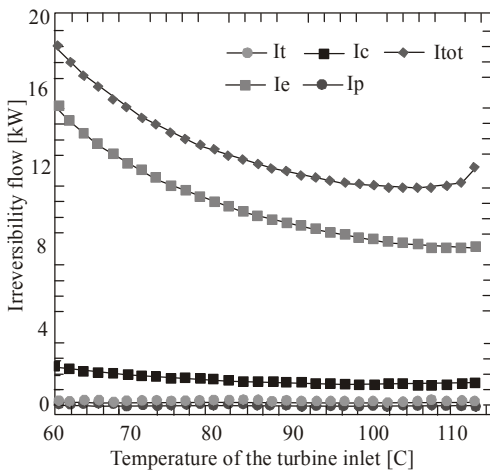Fig. 7: Variation of irreversibility flow vs the turbine inlet temperature

Fig. 8: Variation of irreversibility flow by component vs the turbine inlet temperature

R717 and 152a (9.86%). The R134a has the lowest exergy efficiency (9.61%) this is due to its high share of
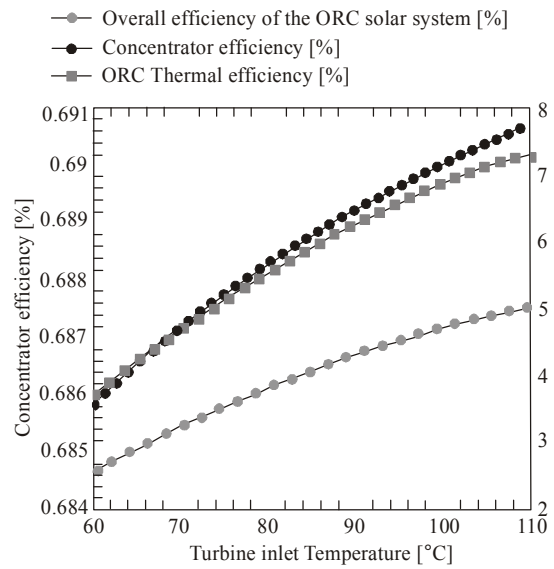

Fig. 9: Overall efficiency, from the collector and thermal vs turbine inlet temperature

irreversibility, which is 13.56 kW. The irreversibility and exergy efficiency are inversely proportional in ORC cycle.

Figure 7 shows that when the inlet temperature of the turbine increases the total rate of irreversibility decreases and beyond the critical temperature of each fluid, the latter begins to lose these thermo physical characteristics and this entails a increased irreversibility rates. The evaporator and the condenser contribute 78% and 14% of the total irreversibility of the cycle, respectively, followed by the turbine (Fig. 8).

The overall yield of the ORC solar system varies from 3.72% (R134a) to 3.98% (R717) this is due to the thermal efficiency of the ORC cycle which is low, ranging from 5.4% (R134a) at 5.77% (R717), compared to the solar concentrator efficiency varying in a range of 68.6% to 69.1% (Fig. 9). Figure 10 shows that when the inlet temperature of the turbine increases Overall Performance also increases From Fig. 9 it is clear that the inlet temperature of the turbine significantly influence the thermal efficiency and (the) exergy efficiency and consequently the overall performance. So it is always necessary to work with higher temperatures (in order) to ensure maximum performance of the cycle, without forgetting that these operating temperatures should be around 10-15°C below the working fluid critical temperature (Delgado-Torres and Garcia-Rodriguez, 2007)

Influence of ambient temperature: The ambient temperature influences strongly the overall performance (Fig. 11). Beyond 30°C the overall efficiency decreases with (the) increasing (of) ambient temperature if the ORC cycle uses as working fluid R290 or R134aFor R152a and R717 those fluids overall performance retains its performance up to 48°C. In hot areas the
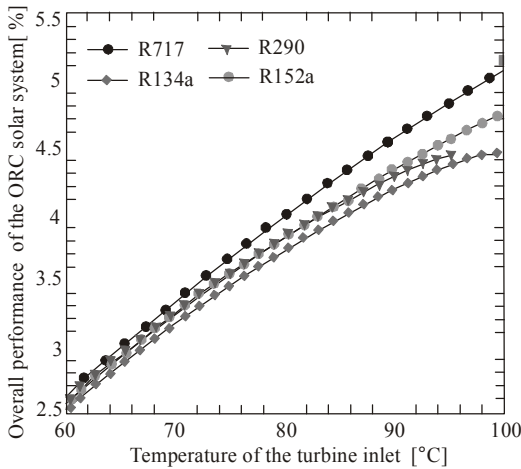
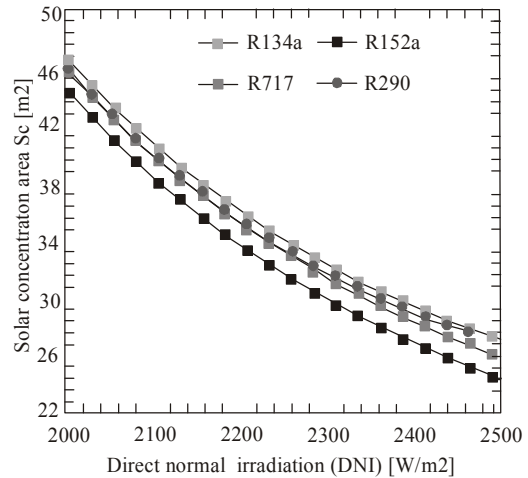Fig. 10: Overall Performance vs the inlet temperature for various fluids work to Tc = 30 C
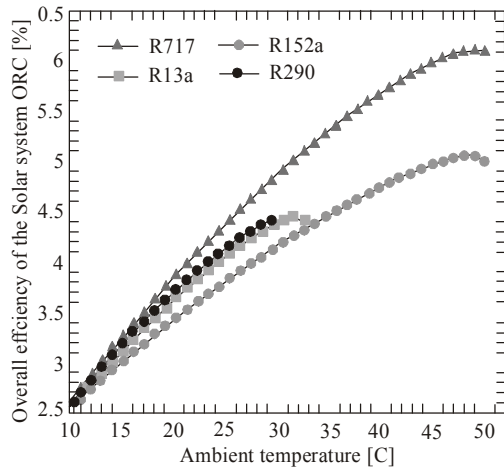

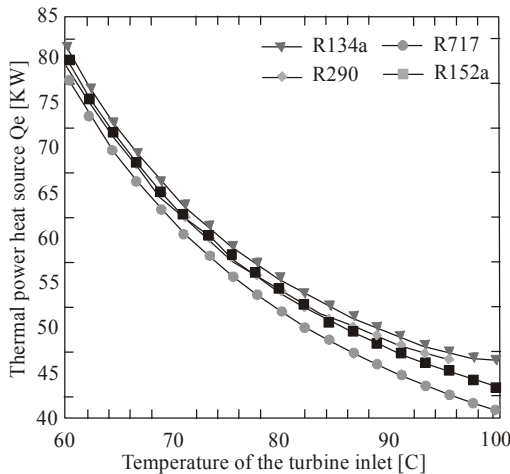Fig. 11: Global efficiency vs ambient temperature for various working fluid

Fig. 12: Variation of the heat input vs the turbine inlet temperature

operation of the ORC solar system with R290 and R134a working fluid is unfavorable.

Fig. 13: Variation of the sensor surface and the direct normal radiation

Influence of heat input from the heat source: Heat input in an ORC solar system is important. It determines the surface of the solar concentrotor and defines a substantial portion of the cost of the system. Therefore, the fluids which have minimum amount of heat are preferred. From Table 4 the heat input required to produce 3 kW varies 51.01 kW (R7171) to 54.95 kW (R134a). Figure 12 shows that the heat input requirement is reduced when the inlet temperature of the turbine increases.

Influence the implantation site on the solar system ORC: The implantation site of a solar system ORC greatly influences the performance and cost of the system. The heat input required to produce a net working 3 kW depends exclusively on the direct normal radiation received at site. When DNI is high at the implantation site, the surface of the solar collector decreases (Fig. 13). This causes a reduction in the cost of the overall system and the cost of the space occupied by the system. From Table 2, the area needed to produce 3 kW varies from 32.78 m^2 (R717) 35.02 m^2 (R134a).

Influences the operating pressures and the turbine flow rate: High pressure in an ORC cycle requires a robust system and increases the cost of the components. To overcome this problem, it would be wise to use two micro-turbines in series where parallel to divide the pressure at the inlet of the turbine. This is advantageous since the power system will be higher. In return, the system becomes expensive due to the increase of micro turbines numbers. From the Table 4 we see that the fluid R152a and R134a have low-pressure values in the condenser while the fluids R290 and R717 have the highest pressures in the evaporator greater than 2.5 MPa (Maizza and Maizza, 1996).
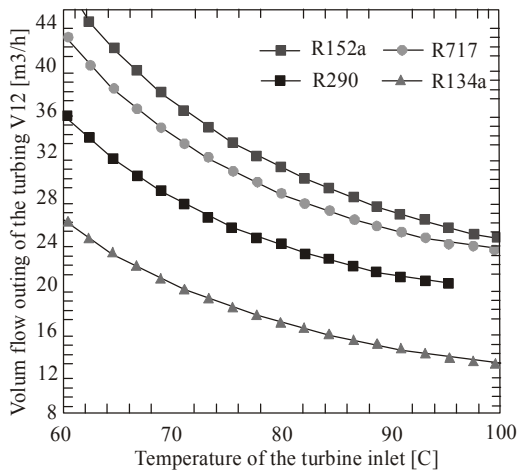
Fig. 14: Variation of the volume flow vs the turbine inlet
temperature

From the economic point of view, the volume flow
leaving the turbine is important because it determines
the size and cost of the microturbine. As shown in
Table 4, the fluid R717 and R152a have the lowest
rates. A low volume flow rate also reduces the size of
the pump and reduces losses of the connecting pipes
loads. In general, the volume flow leaving the turbine is
inversely proportional to the turbine inlet temperature,
which can be illustrated in Fig. 14.

Security and environmental impact: Some
substances, mostly refrigerants deplete the ozone layer
and / or contribute to global warming. Because of their
adverse effects, there is a need to choose those that
have less harmful effects on the environment. The
fluids R717 and R290 are low GWP R152a and R134a
monitoring whose GWP is a little high (Table 2).
According to the classification ASHAR (Table 3)
R134a is the best candidate (A1) non-flammable nor
toxic followed by R152a (A2) flame retardant and

nontoxic. The R290 (A3) more flammable than R152a
and nontoxic. R717 (B2) bit flammable and toxic.

**Performance of the ORC solar system with and
without overheating of 5°C:** With an overheating of
5°C, the solar ORC overall performance improved
slightly (Table 4). It reached its maximum of 4.27%
(R717) with overheating compared to 3.96% (R717) in
the case without overheating (corresponding to an
increase of 7.7%). Despite the increase in overall yield
of the solar ORC cycle thermal and exergy efficiency
improved by 16.1% and 14.4% respectively. This
results in a decrease in the amount of heat required (Qe)
of 51.93 kW to 45.37 kW (Corresponds to a 14.4%
reduction) and therefore the surface of the concentrator
required to produce a net work of 3 kW is lowered
32.78 m^2 to 30.51 m^2 (7.4% reduction). Beyond the
critical temperature of each working fluid, with or
without overheating the thermal efficiency and exergy
decrease. The reason for this decrease is due to the
growth of irreversibility flow (I tot) and the amount of
heat (Q e) of the high evaporation temperature Te cycle
(Fig. 7 to 13).

**Optimization of some operating parameters of the
solar system ORC:** The aim is to optimize the surface
of the solar concentrator and the volume flow of the
outlet turbine for economic reasons.

The optimization of the surface of the solar
concentrator will reduce the cost of the optical system
and in turn a decrease in overall area occupied by the
solar system ORC. On Fig. 15 to 19 the arrows are
directed towards areas of unfavorable operating while
the optimal operating zones are identified by the points.
Figure 15 shows that the areas where the solar yield
ORC is maximum (5.041%) correspond to a surface of
the concentrator ranging from 20 m^2 to 26 m^2 over
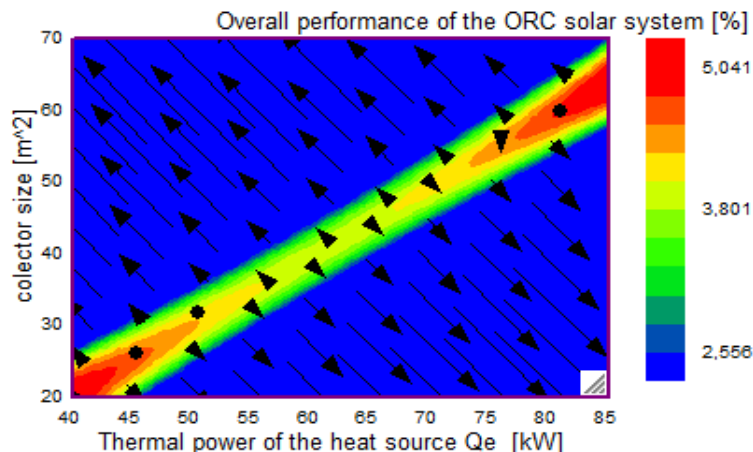30.51 m^2 to 32.78 m^2 before optimization (26%

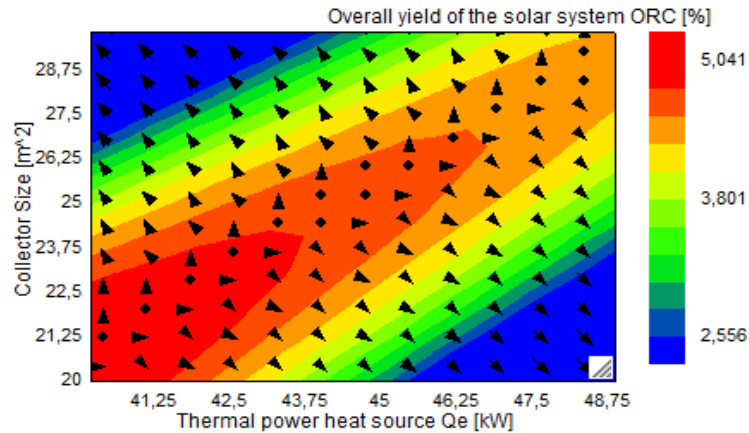Fig. 15: Optimization of the surface of the concentrator vs thermal power heat source
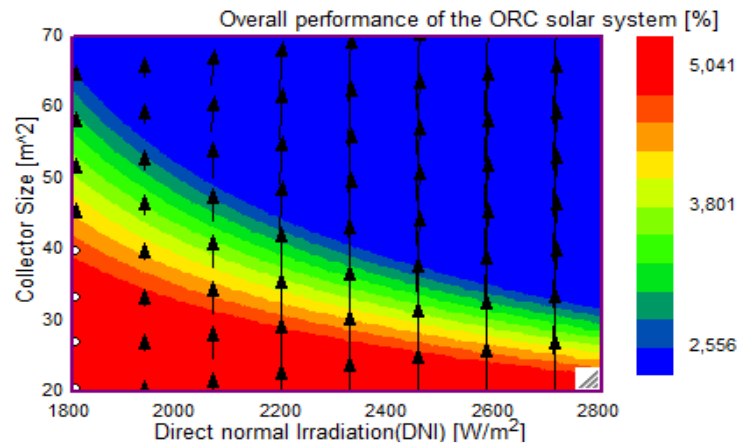
Fig. 16: Optimum operating regions

Fig. 17: Optimum operating regions

reduction). This is caused by decreased mass flow 0.18 kg/s to 0.16 kg/s at an operating temperature of 85°C. The optimum area to maximize the overall efficiency and minimize the surface of the concentrator is observed at a 21.25 m^2 surface and a minimal amount of heat 41.25 kW (Fig. 16). This amount of heat was obtained with a direct normal insolation of 1800 W/m^2 (Fig. 17). The volume flow out of the turbine determines the cost of the micro turbine. In order to optimize it we must on the one hand, minimize the mass flow rate of the cycle and secondly to ensure that the temperature of the turbine inlet does not exceed the critical temperature of the working fluid. By minimizing the mass flow we at the same time maximize the overall performance and the parallel volume flow is also optimized. The maximum overall efficiency is obtained with mass flow and volume ranging from 0.16 kg/s to 0.14 kg/s and 24 m^3/h respectively (Fig. 18). The most favorable area to minimize the volume flow and maximize the overall performance is observed to 21.424 m^3/h (Fig. 19) with a minimum rate of 0.14 kg/s at 90°C (Fig. 20).

CONCLUSION

This study focuses on the analysis of performance and the optimization of an organic Rankine cycle coupled to a Fresnel linear concentrator. The thermodynamic modeling of organic Rankine cycle was obtained by using the energy and mass balance for four preselected working fluids with operating temperatures between 80°C to 130°C. The model of the solar collector used for this study is a commercial model developed by the company Novatec Solar.

The influence of several thermodynamic and physical parameters on the performance of the overall system were analyzed for different working fluids.

Following this analysis the R152a fluids and R134a appears as potential candidates for solar applications at low temperature ORC, followed by the fluids R290 and R717 that offer excellent performance but require safety precautions, because of their flammability and toxicity respectively.

The evaporation temperature and ambient strongly influence the overall performance, thermal and Exergy system efficiency. Parallel to the increase of the
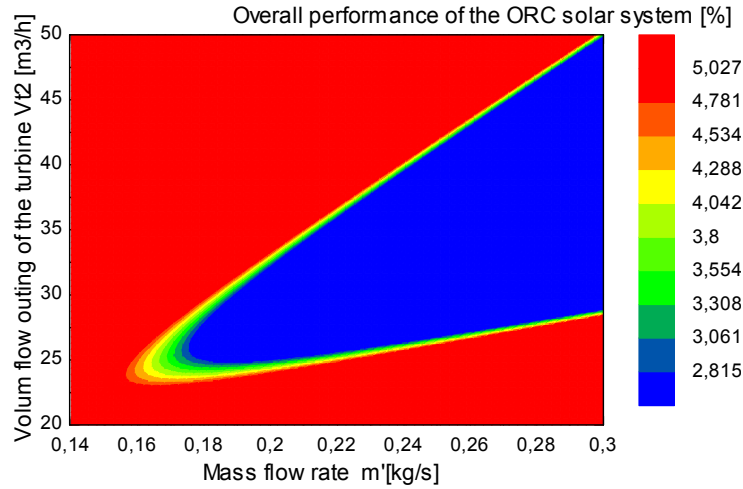
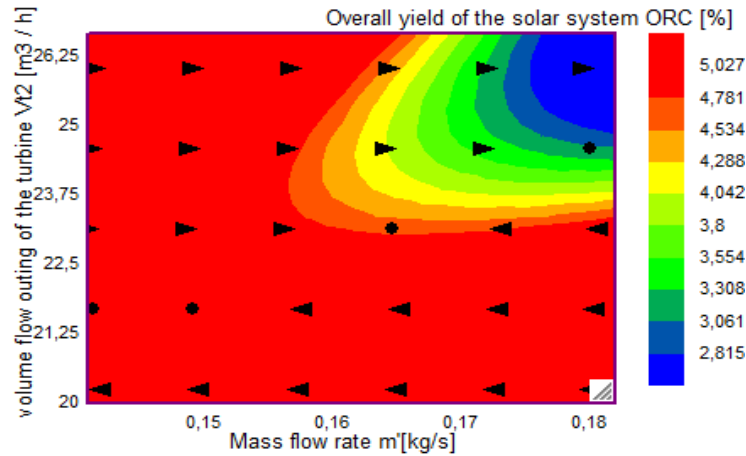Fig. 18: Optimization of the volume flow out of the turbine
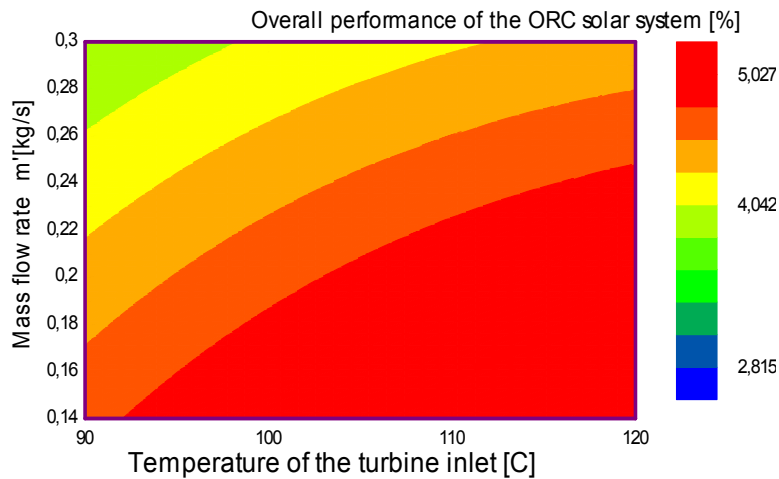

Fig. 19: Optimum operating regions

Fig. 20: Overall yield vs the turbine inlet temperature and the mass flow rate

evaporating temperature, the system yields also increase and once it approaches the critical temperature of the working fluid yields it begins to decline, following a deterioration of the thermo physical properties beyond this temperature. Beyond ambient temperatures of 30°C for R134a and R290 fluids and

48°C for R152a R717 fluids the overall system performance degrades.

The performance of the ORC solar system without and with overheating has been studied for various working fluid. With an overheating of 5°C the overall efficiency of the solar system ORC, energy and exergy efficiency of the ORC cycle has been improved to 7.7, 16.1 and 14.4%, respectively.

The optimization of the surface of the concentrator and the volume flow of micro-turbine output have led to a reduction of 26 and 12%, respectively. This is very advantageous to minimize the investment cost of the overall system. The surface and the minimum flow rate acquired to produce 3 kW is 21.25 m^2 and 21.424 m^3/h respectively. This is achieved for the operating conditions: working fluid R152a and the evaporation temperature is 90°C and the direct normal radiationis 1800.

REFERENCES

Bertrand, T.F., G. Papadakis, G. Lambrinos and A. Frangoudakis, 2008. Criteria for working fluids selection in low-temperature solar organic Rankine cycles. Proceeding of the 1st International Conference on Heating, Cooling and Buildings. Lisbon, pp: 513-526.

Bruno, J.C., J. Lopez-Villada, E. Letelier, S. Romera and A. Coronas, 2008. Modelling and optimisation of solar organic Rankine cycle engines for reverse osmosis desalination. Appl. Therm. Eng., 28(17-18): 2212-2226.

Chen, H., D.Y. Goswami and E.K. Stefanakos, 2010. A review of thermodynamic cycles and working fluids for the conversion of low-grade heat. Renew. Sust. Energ. Rev., 14(9): 3059-3067.

Darvish, K., M.A. Ehyaei, F. Atabi and M.A. Rosen, 2015. Selection of optimum working fluid for organic rankine cycles by exergy and exergy-economic analyses. Sustainability, 7(11): 15362-15383.

Delgado-Torres, A.M. and L. Garcia-Rodriguez, 2007. Preliminary assessment of solar organic Rankine cycles for driving a desalination system. Desalination, 216(1-3): 252-275.

Desideri, A., S. Gusev, M. van den Broek, V. Lemort and S. Quoilin, 2016. Experimental comparison of organic fluids for low temperature ORC (organic Rankine cycle) systems for waste heat recovery applications. Energy, 97: 460-469.

Facão, J. and A.C. Oliveira, 2009. Analysis of energetic, design and operational criteria when choosing an adequate working fluid for small ORC systems. Proceeding of the International Mechanical Engineering Congress and Exposition. Lake Buena Vista, Florida, USA, 6: 175-180.

Kuo, C.T., H.Y. Shin, H.F. Hong, C.H. Wu, C.D. Lee, I.T. Lung and Y.T. Hsu, 2009. Development of the high concentration III-V photovoltaic system at INER, Taiwan. Renew. Energ., 34(8): 1931-1933.

Lemort, V., 2006. Testing and modeling scroll compressors with a view to integrate them as expanders into a Rankine cycle. DEA Thesis, University of Liege.

Maizza, V. and A. Maizza, 1996. Working fluids in non-steady flows for waste energy recovery systems. Appl. Therm. Energ., 16(7): 579-590.

Niez, A., 2010. Comparative Study on Rural Electrification Policies in Emerging Economies: Keys to Successful Policies. OECD/IEA, Paris, France, pp: 118.

Quoilin, S., M. Orosz and V. Lemort, 2008. Modeling and experimental investigation of an organic rankine cycle using scroll expander for small scale solar applications. Proceeding of the 1st International Conference on Solar Heating, Cooling and Building. Lisbon.

Stijepovic, M.Z., P. Linke, A.I. Papadopoulas and A.S. Grujic, 2011. On the role of working fluid properties in organic rankine cycle performance. Appl. Therm. Eng., 36: 406-413.

Wang, E.H., H.G. Zhang, B.Y. Fan, M.G. Ouyang, Y. Zhao and Q.H. Mu, 2011. Study of working fluid selection of organic Rankine cycle (ORC) for engine waste heat recovery. Energy, 36(5): 3406-3418.

Wang, Z.Q., N.J. Zhou, J. Guo and X.Y. Wang, 2012. Fluid selection and parametric optimization of organic Rankine cycle using low temperature waste heat. Energy, 40(1): 107-115.