Performance Evaluation of Solar Energy-driven Binary Rankine Cycle under Egyptian Climatic Conditions

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Abstract

The importance of the Organic Rankine Cycle (ORC) comes from dealing with low-quality energy sources resulting from many thermodynamic processes. This type of low-level energy source permanently causes heat emissions, global warming, as well as climatic changes in which our world lives today. Re-exploitation of this type of energy contributes to the production of the required energy for various industrial purposes, as well as the production of electrical power. In this paper, the ORC is integrated with the Steam Rankine Cycle (SRC) as a bottoming cycle which can be called, the Binary Rankine Cycle (BRC) to generate 5MW net power. As there are two fluids used in the system, water in the topping cycle and organic fluid in the bottoming cycle. The BRC may utilize solar energy as a heating source. With the increasing need for efficient and low-cost energy solutions, it is essential to understand the thermal performance of a system built in different climates. The performance of the system is carried out and compared with different organic fluids R134a, R245fa, and R1234yf at different steam turbine inlet temperatures, pressures, and back pressures. The study reveals that the R245fa BRC has the best performance followed by the R134af BRC and then the R1234yf BRC. Also, the study aimed to determine the best solar collector area in Aswan, Hurghada, & Cairo.

Keywords:

Organic Rankine Cycle; (general keyword); Evaporator; Climatic conditions; Organic fluids.

1 Introduction

Many researchers around the world are interested in developing and improving ORC performance. Such studies had been conducted to provide a complete understanding of the capability and reliability of these types of low-grade energy cycles. Imran et al. **Error! Reference source not found.**, **Error! Reference source not found.** and

Wang et al. **Error! Reference source not found.** compared different configurations of ORC systems under constant heat source conditions using a thermo-economic optimization study. The results revealed that R245fa was the best working fluid under prescribed conditions. Also, Imran et al. **Error! Reference source not found.** studied the geometrical parameters of the evaporator of the ORC for low-temperature geothermal heat sources to optimize the cost and the pressure drop of the evaporator. Besides, the heat source temperature and the pump speed were investigated experimentally to assess the performance of a small-scale ORC with R245fa as a working fluid [5]. The different operating conditions of R245fa used in ORC were investigated to evaluate the thermal and hydraulic performance of the brazed plate heat exchanger **Error! Reference source not found.**. In addition, the performance of the diaphragm pump of an ORC was tested experimentally with different types of working fluids under various conditions. The highest isentropic efficiency for the diaphragm pump was achieved with R245fa [7]. Jung et al. [8] demonstrated, experimentally and numerically, the viability of using the zeotropic mixture as a working fluid through a 1 kW ORC. Another study evaluated experimentally a new zeotropic mixture of R245fa 60% R134a 40% (molar concentration) on the performance of a 1 kW ORC [9]. For the same output power of 1 kW ORC, Muhammad et al. [10] performed an experimental study using R245fa as a working fluid. In the same way, A 3 kW ORC had been studied experimentally using an open-drive scroll expander and R245fa [11]. In addition, Kim et al. [12] studied and evaluated the performance of an ORC that generates less than 10 kW using R245fa and a scroll expander as the power generation device. A regenerative 11 kW ORC was tested experimentally with two different working fluids to assess the effect of the screw expander performance on the behavior of such a system at nominal and part-load conditions [13]. Abadi and Kim [14] studied different zeotropic fluid mixtures made of two or three refrigerants to enhance the performance of the ORC [14]. Besides, different designs of ORC systems had been applied for waste heat recovery of light-duty vehicles [15] and truck diesel engines [16], [17], and [18]. A one-dimensional analysis method simulated the off-design conditions of the radial-inflow turbine and the heat exchanger of an ORC system for industrial waste heat recovery [19 and 20].

According to the review of the previous research, few researchers have studied the effect of the ORC on the performance of the BRC. As a result, the present work aims to investigate the impact of the ORC assisted by solar energy on the performance of the BRC. This is accomplished by testing various types of refrigerants and their effects on the BRC's performance.

2 System Description and Methodology

The main objective of this study is to evaluate the performance of the thermal power system assisted by parabolic trough solar collectors (PTC) which are integrated with the BRC. The BRC is a combination of two cycles with two different fluids, one in a lower-temperature zone and the other in a high-temperature zone. The proposed BRC consists of an SRC as a topping cycle and an ORC as a bottoming cycle. A heat exchanger will be used in a BRC to transfer the thermal energy from the steam to the organic working fluid. The organic fluid receives heat from the steam to drive the turbine of the ORC cycle before being condensed and returned to the evaporator by the feed pump. Generally, the system consists of the solar field which will be parabolic through the solar collector, the Primary cycle will be the SRC, and the second cycle will be the ORC as shown in Fig. 1.

The working fluids and their thermodynamic properties used in ORC are illustrated in Table 1.

Table 1: Thermodynamic properties of the working fluids used in the ORC

Refrigerant	C_P (kJ/kg.K)	h_{fg} (kJ/kg)	M_W (kg/kmol)	T_C (℃)	P_C (kPa)
R134a	1.451	166.7	102	101	4059
R245fa	1.338	183.4	134	154	3651
R1234yf	1.416	135.6	114	94.7	3382

The principle of operation is that the parabolic trough collects the rays of solar radiation and directs them to the thermal heat transfer fluids such as molten salt and then pumped to the heat exchangers to produce superheated steam. Thermal energy in the steam cycle will be converted into electrical energy. The outlet heat from the steam turbine will be used to heat the organic fluid in the ORC to produce electricity. The general objective of the present work can be broken down into the following specific objectives that would together achieve the overall goal of the research as follows;

1. Developing a computer model using EES software to investigate the effect of the various operating parameters (maximum pressure, maximum temperature, and back pressure) on the performance of the steam Rankin cycles to obtain the optimum values of these parameters,

2. Comparing the performance of the proposed system using different organic working fluids to choose the best one.

To achieve the mentioned objectives, some assumptions are considered such as the steady-state operation of the system, no pressure drop in pipes and connections, the pressure changes only through the expansion devices and pumps, heat losses from/to the system are ignored and internal irreversibility is ignored [22].

2.1 Energy analysis of the proposed system

The equations for energy and mass balance (Eq. 1&2) are written for each component in the proposed system which is considered as control volume.

$$\sum m_{in} = \sum m_{out} \tag{1}$$

$$Q - W + \sum m_{in} h_{in} - \sum m_{out} h_{out} = 0 \tag{2}$$

The performance of the proposed system can be evaluated by many parameters such as the efficiencies of SRC and ORC cycles which are by the ratio of net power output to the heat supplied for each one as written in Eq. 3&4.

$$\eta_{SRC} = \frac{W_{ST} - W_{SP}}{Q_{in,SRC}} \tag{3}$$

$$\eta_{ORC} = \frac{W_{OFT} - W_{OFP}}{Q_{in,ORC}} \tag{4}$$

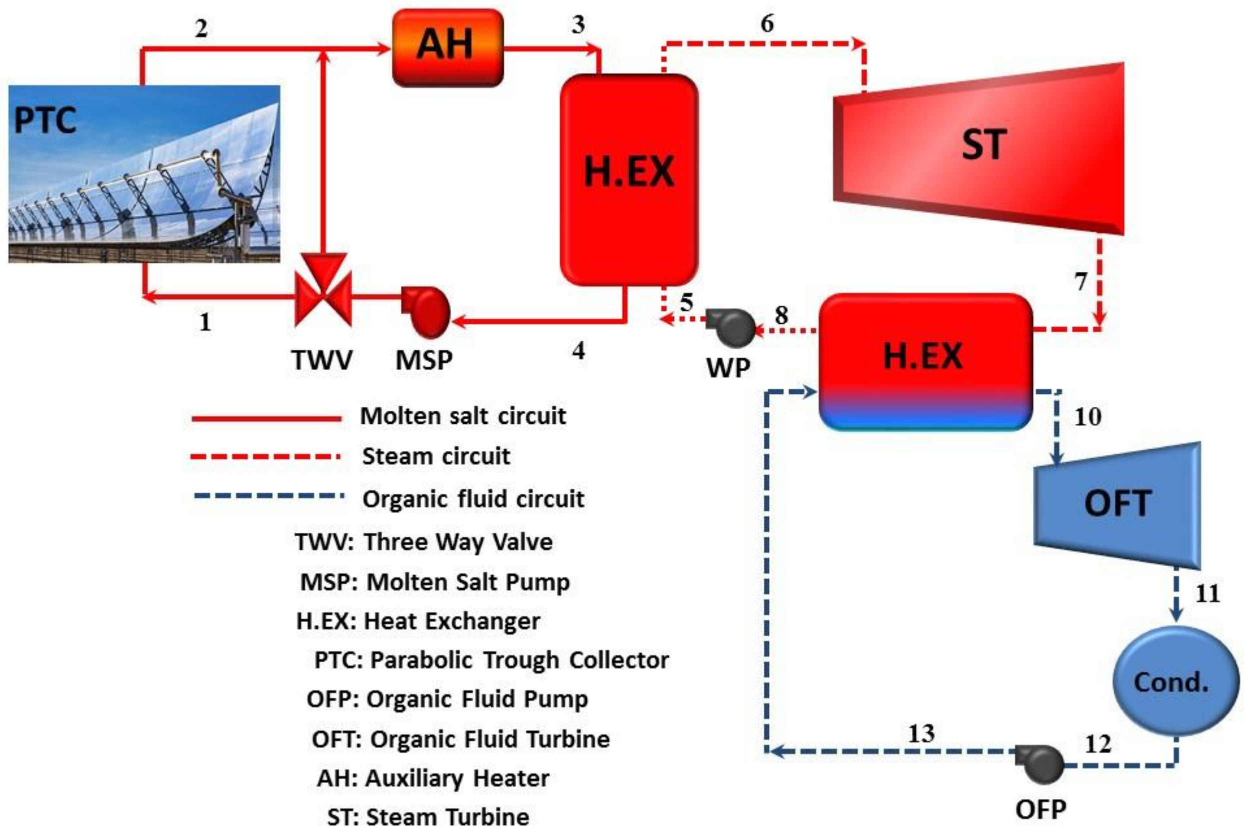

Fig. 1 Schematic diagram of the proposed system.

2.2 Exergy analysis of the proposed system

Exergy is the minimum work that could be achieved by bringing the system into thermodynamic equilibrium with its environment. The specific useful exergy of each state in the proposed system can be determined using eq. 5.

$$e = h - h_0 - \left(T_0(S - S_0)\right) \text{ [23]} \qquad (5)$$

Where h is the enthalpy, S is the entropy, and subscript 0 refers to the standard condition taken by 298K and 1atm.

Exergy loss can be calculated for each component in the proposed system as defined in the following equation:

$$exe_{loss} = (e_{in} - e_{out}) - w \text{ [23]} \qquad (6)$$

Where e_{in} & e_{out} show the outlet and inlet exergy of each component, w refers to the work done by that component (for all heat exchangers assumed that the w equals zero).

The exergy efficiency had been calculated by using the following equation:

$$\eta_{exe} = 1 - \frac{exe_{loss,tot}}{e_{i,tot}} \quad (7)$$

The input parameters used in the current study are listed as shown in T

Table *2*: input parameters used in the study [21]

Input Data	**Values**
Steam turbine Rankine cycle efficiency	0.85
Steam cycle pump efficiency	0.80
The power output of the plant	5.00 MW
Baseline steam turbine inlet temperature	380.00℃
Baseline steam turbine inlet pressure	12.00 MPa
Electrical generator efficiency	0.96
Electrical motor efficiency	0.96

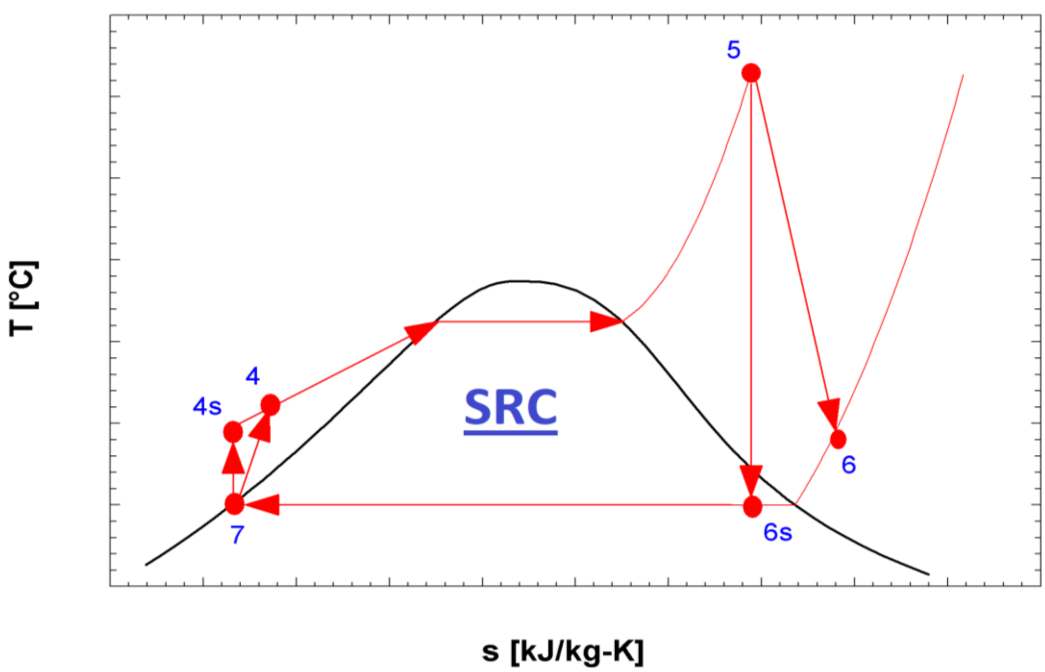

Fig. 2 T-s Diagram for SRC.

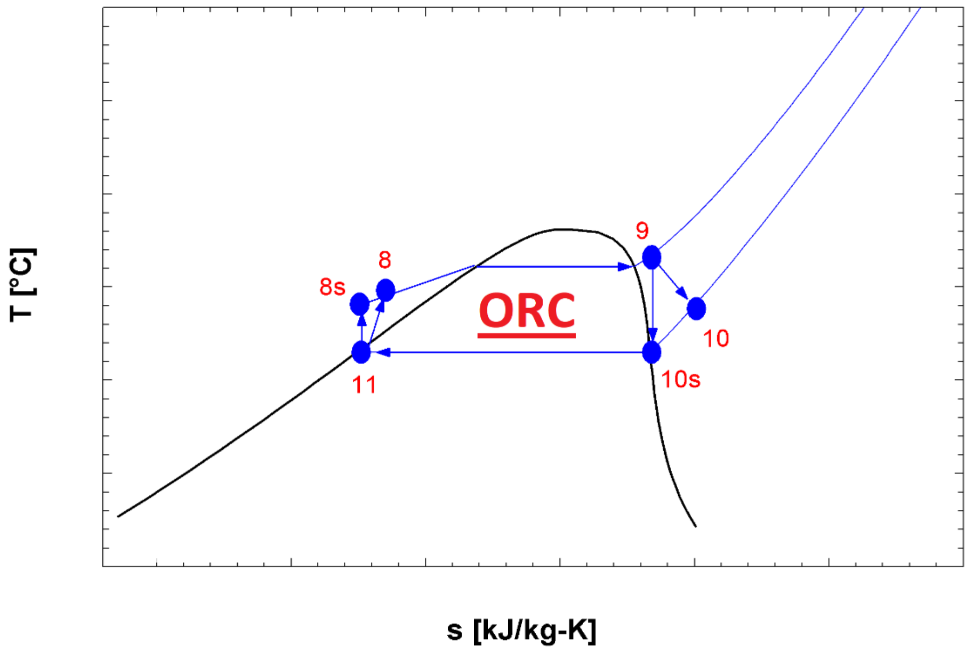

Fig. 3 T-s Diagram for ORC.

2.3 Model Validation

The available data from the literature [21] was used to validate the created models for the parametric simulation of the Binary Rankine Cycle. The simulation results listed in Table 3 compared with those documented in the literature. As can be observed, the results of the parameters computed in the current work and those reported in earlier literature are in good accord.

Table 3: Comparison between obtained results and data in the literature [21].

State Number	Mass Flow Rate (kg/s)		Pressure (kpa)		Temperature (℃)		Enthalpy (kJ/kg)	
	Present Work	Reference	Present Work	Reference	Present Work	Reference	Present Work	Reference
4	61.48	61.5	12,000.00	12,000.00	101.20	101.18	433.00	433.00
5	61.48	61.5	12,000.00	12,000.00	379.90	380.00	2,979.00	2,977.20
6	61.48	61.5	100.00	100.00	99.61	99.63	2,279.00	2,278.20
7	61.48	61.5	100.00	100.00	99.61	99.63	417.50	417.50

8	590.50	591.8	2,660.00	2678.94	36.39	36.40	102.80	102.80
9	590.50	591.8	2,660.00	2678.94	89.61	89.63	296.60	296.10
10	590.50	591.8	887.50	887.47	42.77	42.33	277.40	277.00
11	590.50	591.8	887.50	887.47	35.00	35.00	100.90	100.90

3 Results & Discussion

The effect of the inlet temperature of SRC will be demonstrated through the range of (300 - 500°C). Besides, the effect of the inlet & back pressure of SRC through the ranges of (6 - 15 kPa) and (50 110 kPa) respectively. This study will be done in 3 different cities in Egypt, Aswan, Hurghada, & Aswan using 3 different working fluids R245fa, R134a, & R1234yf.

The back pressure of the steam turbine is a crucial design consideration for the SRC and ORC. Consequently, it was investigated in this study. Fig. 4 depicts the influence of steam turbine back pressure on the thermal efficiency of the considered systems. Fig. 4 depicts the effect of steam turbine back pressure over the range of (50 to 110 kPa) on the overall thermal efficiencies of the systems under consideration. The value of the SRC inlet pressure is fixed at 12,000 kPa. At a backpressure value of 50 kPa, the efficiency of the SRC is 27.71% and the efficiencies of the ORC for refrigerants (R134a, R1234yf, and R245fa) are 7.089%, 6.808%, and 7.554% respectively. Consequently, the energy level at the relevant saturation temperature is extremely low to drive the ORC.

On the other hand, At the backpressure of 100 kPa, the efficiency of the SRC is 25.74% and the efficiencies of the ORC for refrigerants (R134a, R1234yf, and R245fa) are 9.056%, 8.434%, and 10.190% respectively. As the back pressure rises, the thermal energy transferred to the ORC increases, resulting in a rise in the thermal efficiency of the ORC. Nonetheless, the drop in the pressure ratio of the steam turbine as a result of an increase in backpressure reduces the SRC's efficiency. Accordingly, the thermal efficiency of the BRC improves as the ORC thermal efficiency increases. The R245fa BRC has the best thermal efficiency among the examined BRCs because it has a high latent heat compared to other refrigerants as mentioned in Table 1. While the R1234yf BRC has the lowest thermal efficiency. At 100 kPa, the efficiency of the R1234yf BRC is 31.91%, compared to 25.74% for the SRC; this is an increase of 19.33% due to the ORC.

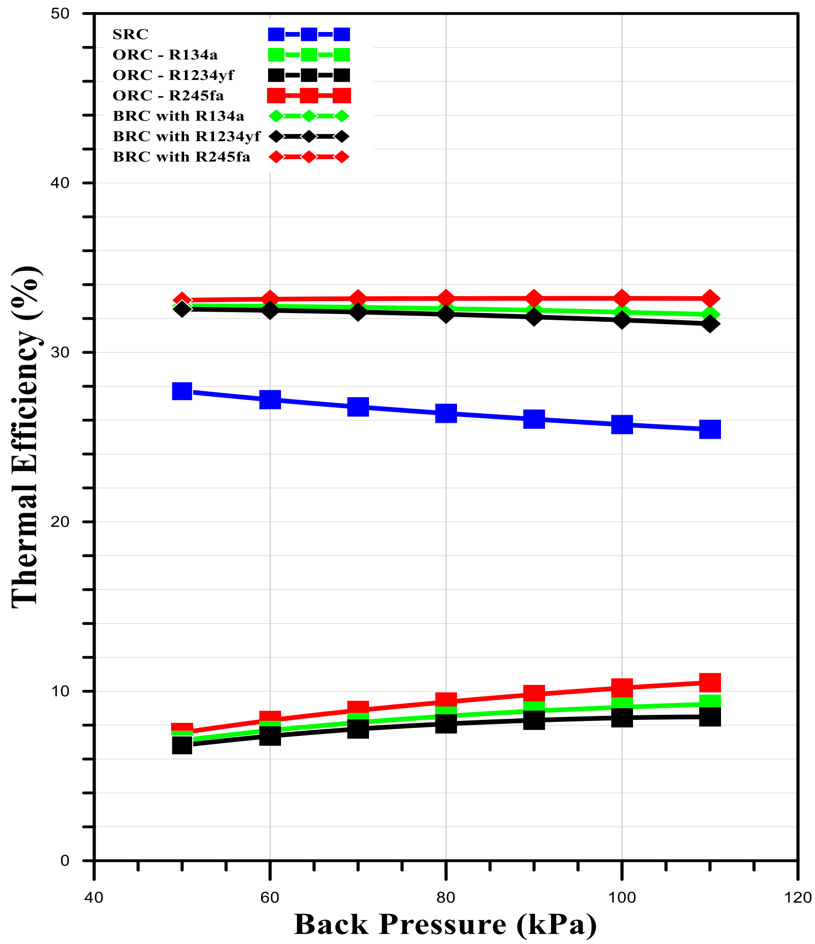

Figure 4: Effect of the back pressure of the steam turbine on the thermal efficiency

Fig. 5 depicts the effect of the steam turbine's back pressure on the BRC exergy efficiency. Fig. 5 shows that the BRC exergy efficiency is inversely proportional to the back pressure of the steam turbine. At the backpressure of 100 kPa, the exergy efficiency of the BRC for refrigerants (R134a, R1234yf, and R245fa) are 35.96%, 25.74%, and 36.88% respectively. In addition, at 110 kPa, the exergy efficiencies of the BRC for refrigerants (R134a, R1234yf, and R245fa) are 35.82%, 25.46%, and 36.86%, respectively.

As shown in Fig. 4, the R245fa BRC has the highest thermal efficiency because it has high latent heat when compared to the other refrigerants listed in Table 1. The

R245fa BRC also has the highest exergy efficiency. Furthermore, the exergy efficiency of the R1234yf BRC is the lowest.

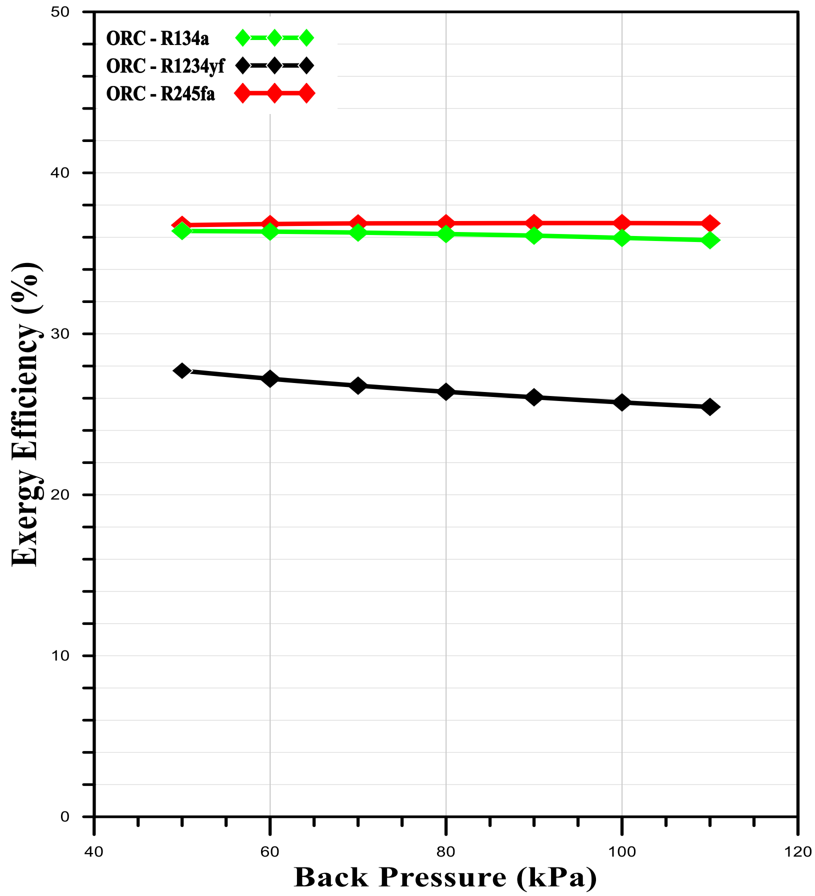

Figure 5: Effect of the back pressure of the steam turbine on BRC exergy efficiency

The effect of the steam turbine's inlet pressure on the thermal performance of the systems under deliberation is shown in Fig. 6 depicts the influence of steam turbine inlet pressure within the range (6,000-15,000 kPa) on thermal efficiency. The value of SRC backpressure is maintained at 100 kPa. As is well known, as the inlet pressure rises, the SRC's thermal efficiency rises, while the ORC’s_thermal efficiency remains constant. The SRC's efficiency is 23.4% and the ORC's efficiencies for the refrigerants R134a, R1234yf, and R245fa are 9.063%, 8.429%, and 10.180%, respectively, at an inlet pressure value of 6000 kPa due to thermophysical properties of the working fluids which mentioned in Table 1.

On the other hand, at an inlet pressure of 15,000 kPa, the efficiency of the SRC is 26.22% and the efficiencies of the ORC for refrigerants (R134a, R1234yf, and R245fa) are constant.

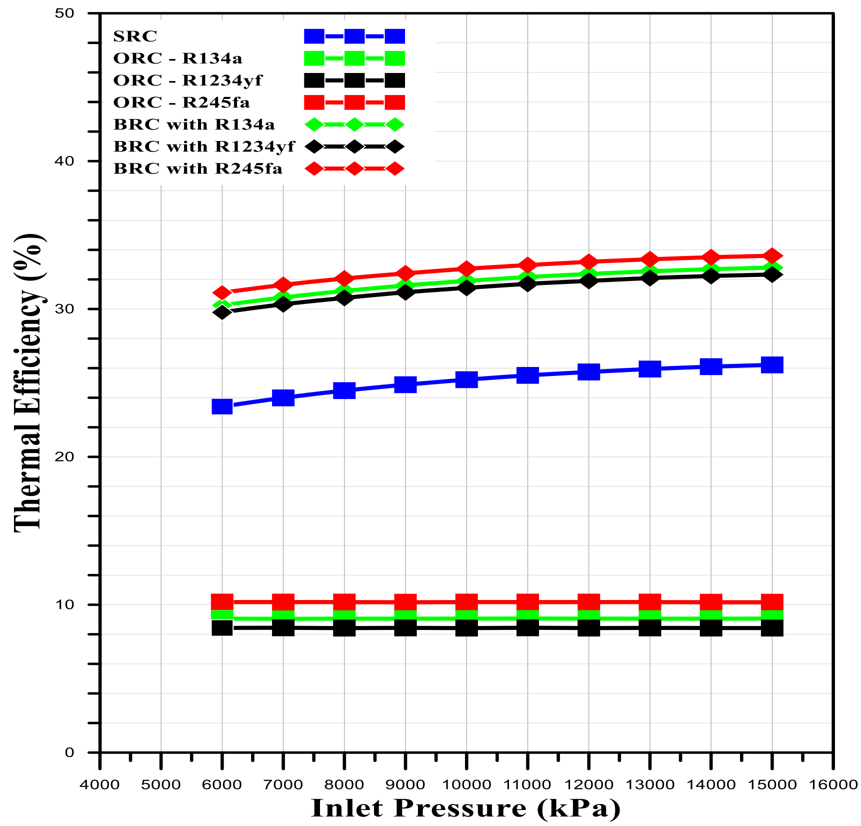

Figure 6: Effect of the inlet pressure of the steam turbine on thermal efficiency and specific work

The impact of the steam turbine's inlet pressure on the BRC's exergy efficiency is shown in Fig. 7. The figure shows that the inlet pressure of the steam turbine affects the BRC's exergy efficiency. At 6000 kPa inlet pressure, the BRC exergy efficiencies for refrigerants (R134a, R1234yf, and R245fa) are 33.62%, 33.08%, and 34.55%, respectively due to its thermodynamic properties. Moreover, at 15,000 kPa the exergy efficiencies of the BRC for refrigerants (R134a, R1234yf, and R245fa) are 36.45%, 35.93%, and 37.35%, respectively. As previously stated, the R245fa BRC has the highest thermal efficiency. Also, the R245fa BRC has the highest exergy efficiency.

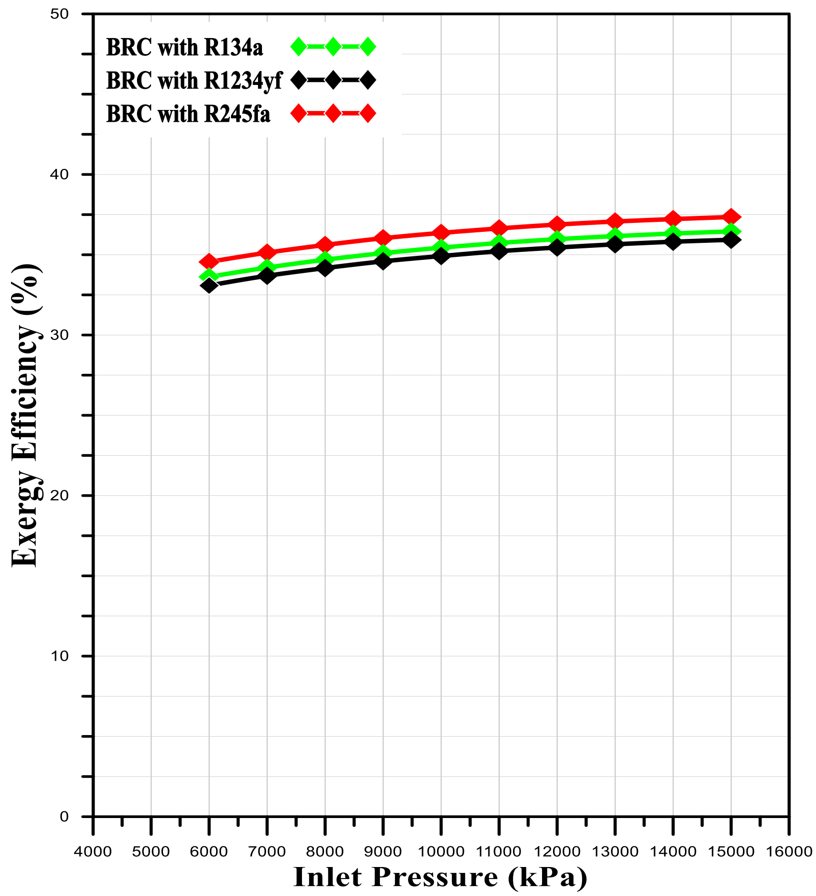

Figure 7: Effect of the inlet pressure of the steam turbine on BRC exergy efficiency

Fig. 8 describes the impact of steam turbine inlet temperature on the thermal efficiency of the systems under concern. Fig. 8 illustrates the effect of steam turbine inlet temperatures between 300 and 550 ºC on thermal efficiency. The inlet pressure is maintained at 12000 kPa, and the SRC backpressure is kept at 100 kPa. The SRC's thermal efficiency increases as the inlet temperature rises, while the ORC's efficiency remains constant. At an inlet temperature of 300 ºC, the SRC efficiency is 16.31% and the ORC efficiency is 9.051%, 8.425%, and 10.180% for the refrigerants R134a, R1234yf, and R245fa, respectively. Alternatively, at an inlet temperature of 550 ºC, the SRC efficiency is 28.08% and the ORC efficiency for refrigerants (R134a, R1234yf, and R245fa) is 9.049%, 8.425%, and 10.180%, respectively because of its thermodynamic characteristics. The SRC efficiency is increased, leading to a rising in the BRC's thermal efficiency. The R245fa & R134a BRCs have the highest thermal efficiency as shown in Fig. 8.

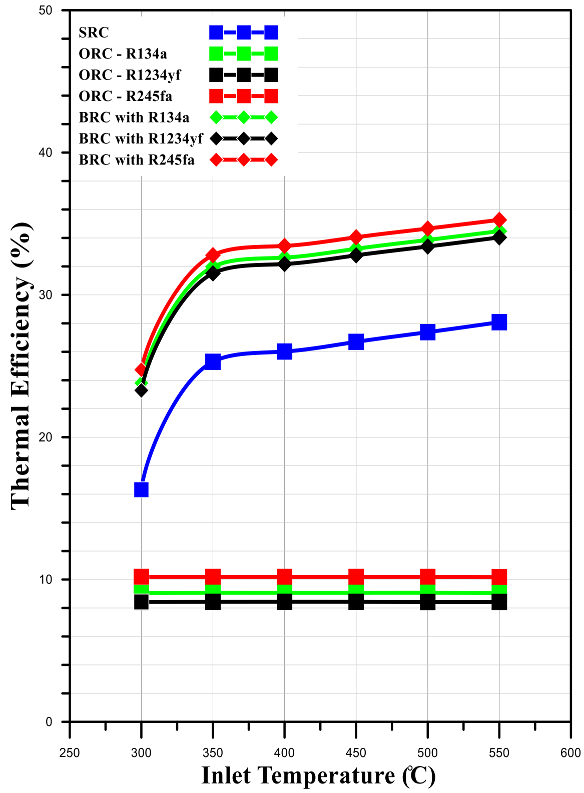

Figure 8: Effect of the inlet temperature of the steam turbine on thermal efficiency and specific work

Fig. 9 shows the effect of the steam turbine's inlet temperature on the BRC's exergy efficiency. Fig. 9 represents an increase in BRC exergy efficiency for all refrigerants as the inlet temperature rises from 300 to 550 ºC. The BRC exergy efficiencies for refrigerants (R134a, R1234yf, and R245fa) are 16.90%, 16.53%, and 17.56%, respectively at 300 ºC. Furthermore, at 350 ºC, the BRC's exergy efficiencies for refrigerants (R134a, R1234yf, and R245fa) are 31.95%, 31.48%, and 32.76%, respectively. At 550 ºC, the exergy efficiency of the BRC for refrigerants (R134a, R1234yf, and R245fa) are 47.17%, 46.57%, and 48.26%, respectively.

The highest thermal efficiency is found in the R245fa BRC, as was previously mentioned. On the other hand, the R245fa BRC continues to have the highest exergy efficiency.

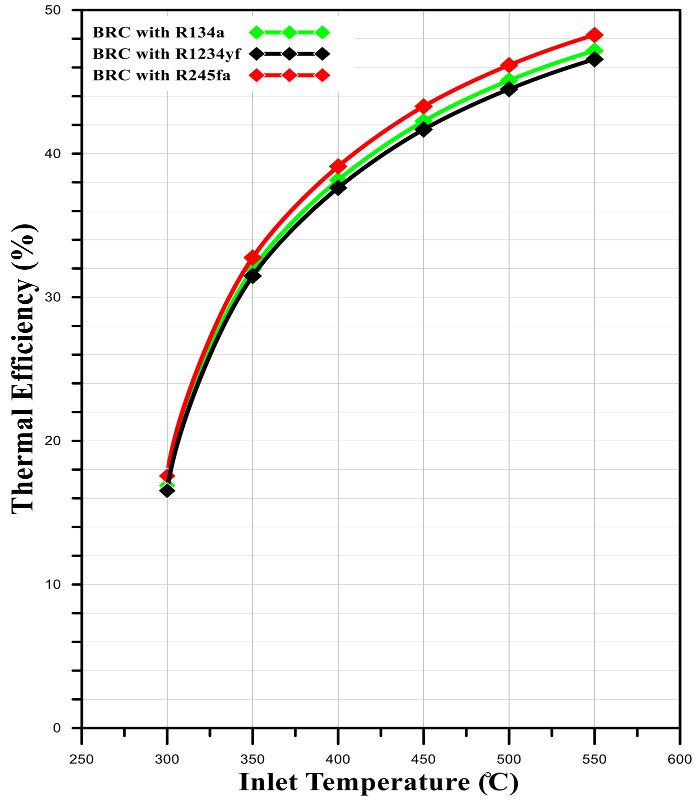

Figure 9: Effect of the inlet temperature of the steam turbine on BRC exergy efficiency

Another important input parameter, which is air dry bulb temperature, should be considered as it affects the system performance. Dry bulb temp as a function of local solar time is shown in Fig. 10. Dry bulb temperature increases gradually and then decreases with local solar time (LST). This behavior depends directly on solar radiation variation as it is the source of heat. The data indicate the dry bulb temperature of ambient air within Aswan, Cairo, & Hurghada cities. The minimum temperature in Aswan is around 10°C at night in winter and the maximum temperature is about 45°C at mid-day in summer. For spring and autumn, the dry bulb temperature reaches 30-35 °C approximately. Whereas in Cairo and Hurghada, the temperature varies from 35.50 to 37.50 °C in summer and 19.00 to 20.00 °C in winter.

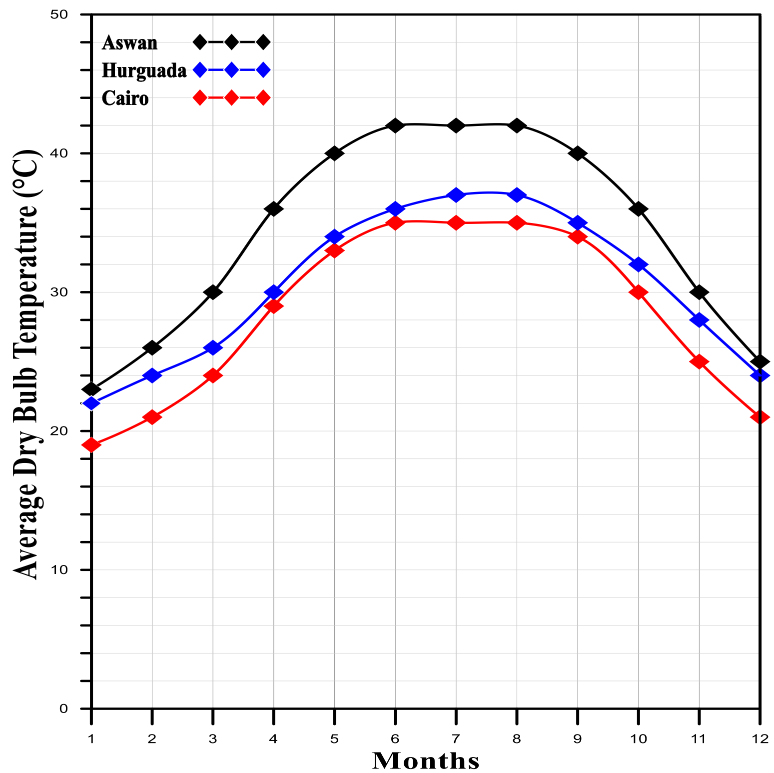

Figure 10: Dry bulb temperature in Aswan City (hourly).

Fig. 11, 12, & 13 represent the effect of the inlet pressure on the area of the solar collector in three cities, Aswan, Hurghada, and Cairo respectively at an inlet temperature of 380 ºC & back pressure of 100 kPa to produce net power 5.00MW. The most expensive part of the system is the solar field so, reducing the size of the field will have a significant impact on the overall cost of the plant. Fig. 11 shows that the area of solar collectors in Aswan City varies from 11,800 m^2 to 12,800 m^2 for refrigerant R134a, from 11,500 to 12,500 for refrigerant R245fa, and 12,000 m^2 to 13,000 m^2 for refrigerant R1234yf. Observation shows that the total area decreases when the inlet pressure of the steam turbine increases. From the below figure, we can conclude that in Aswan city the lowest area among the 3 refrigerants is R245fa. At 6,000 kPa, R245fa requires 12,505 m^2, R134a requires 12,857 m^2, & R1234yf requires 13,063 m^2. At 15,000 kPa, R245fa requires 11,571 m^2, R134a requires 11,859 m^2, & R1234yf requires 12,027 m^2.

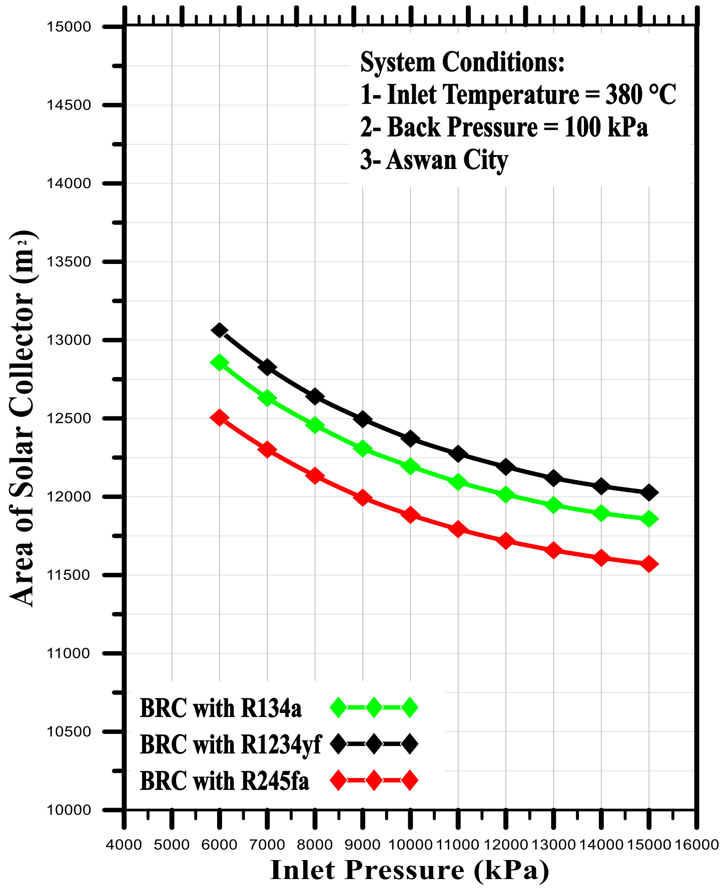

Figure 11: Effect of the inlet pressure on the area of solar collector in Aswan.

Fig. 12 illustrates that the area of solar collectors in Hurghada City varies from 12,000 m^2 to 13,000 m^2 for refrigerant R134a, from 11,600 to 12,600 for refrigerant R245fa, and 12,200 m^2 to 13,200 m^2 for refrigerant R1234yf. Observation shows that the total area decreases when the inlet pressure of the steam turbine increases. From the below figure, also, we can conclude that in Hurghada city the lowest area among the 3 refrigerants is R245fa. At 6,000 kPa, R245fa requires 12,674 m^2, R134a requires 13,031 m^2, & R1234yf requires 13,239 m^2. At 15,000 kPa, R245fa requires 11,727 m^2, R134a requires 12,020 m^2, & R1234yf requires 12,190 m^2.

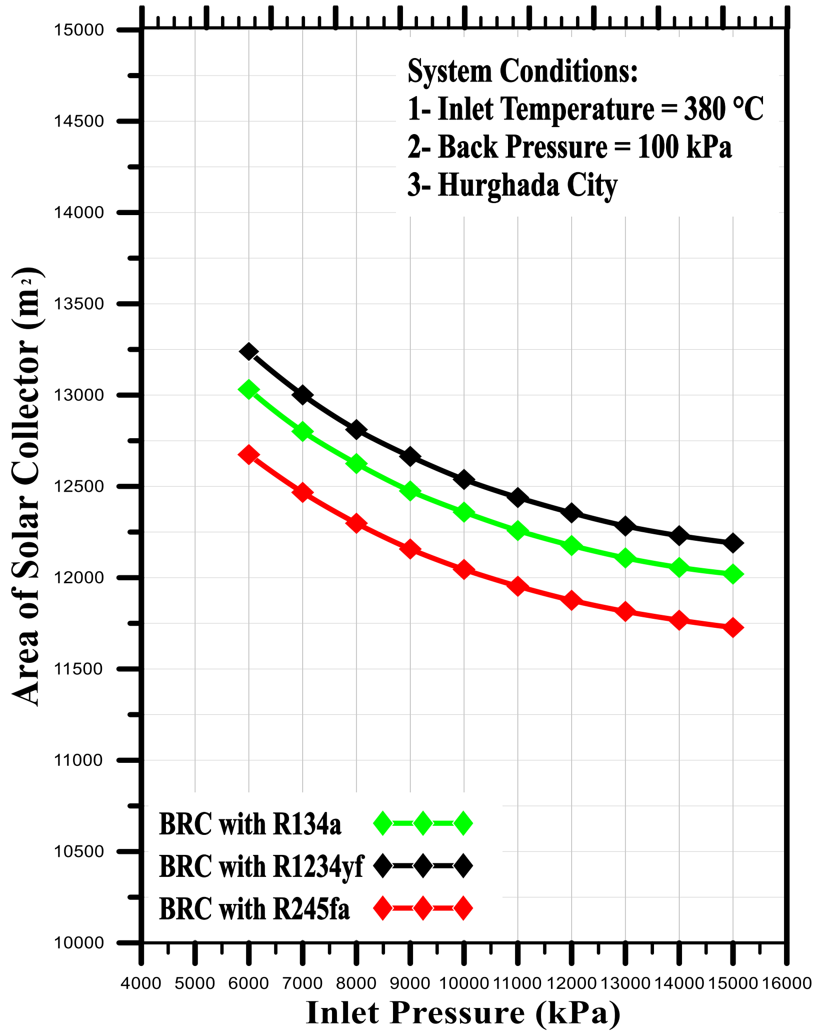

Figure 12: Effect of the inlet pressure on the area of solar collector in Hurghada.

Fig. 13 shows that the area of solar collectors in Cairo City varies from 13,300 m^2 to 14,300 m^2 for refrigerant R134a, from 13,000 to 14,000 for refrigerant R245fa, and 13,500 m^2 to 14,600 m^2 for refrigerant R1234yf. Observation shows that the total area decreases when the inlet pressure of the steam turbine increases. From the below figure, also, we can conclude that in Cairo city the lowest area among the 3 refrigerants is R245fa as well. At 6,000 kPa, R245fa requires 13,981 m^2, R134a requires 14,374 m^2, & R1234yf requires 14,605 m^2. At 15,000 kPa, R245fa requires 12,936 m^2, R134a requires 13,259 m^2, & R1234yf requires 13,447 m^2.

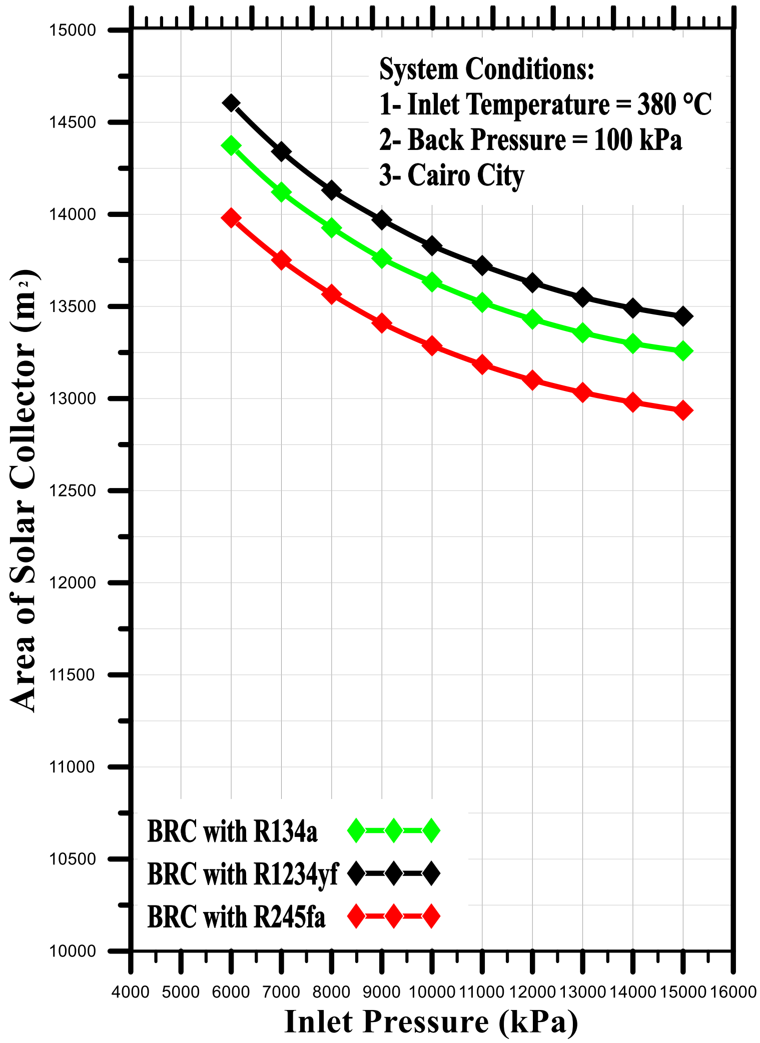

Figure 13: Effect of the inlet pressure on the area of solar collector in Cairo.

4 Conclusion

Nowadays, power generation from low-grade energy sources is gaining interest and the organic Rankine cycle (ORC) is seen as one of the most suitable technologies for such applications. In this paper, the performance of ORC working with different working fluids under Egyptian climatic conditions has been investigated. A computer model using EES software was performed to investigate the effect of the various operating parameters on the cycle performance and compare the performance of the cycle with different organic working fluids to choose the best one. Also, determine the best solar collector area, and the optimum mass flow rate of the hot fluid of the solar field under the climatic conditions of Aswan. It can be concluded from the work that:

- At a backpressure value of 50 kPa, the efficiency of the SRC is 27.71% and the efficiencies of the ORC for refrigerants (R134a, R1234yf, and R245fa) are 7.089%, 6.808%, and 7.554% respectively.
- R245fa has the highest thermal efficiency compared with the other working fluids used in this study followed by the R134a BRC while the R1234yf has the lowest thermal efficiency.
- R245fa has the best exergetic efficiency while R1234yf has the lowest exergetic efficiency.
- Under climatic conditions of Aswan, the required collector area for R245fa, R134a, and R1234yf are 12,505 m^2, 12,857 $m^{2,}$ and 13,063 m^2, respectively.

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NOMENCLATURE	
C_p	Specific Heat (kJ/kg.k)
e	Exergy (kJ/kg)
exe_{loss}	exergy loss (kJ/kg)
h	Enthalpy (kJ/kg)
h_{fg}	Enthalpy of vaporization (kJ/kg)
MW	Molecular Weight (kg/kmol)
m_s	The mass flow rate of steam (kg/s)
m_{wf}	The mass flow rate of organic fluid (kg/s)
P	Pressure (kPa)
P_C	Critical Pressure (kPa)
Pr	Pressure ratio
Q	Heat input (kW)

S	Entropy (kJ/kg.C)
T_C	Critical Temperature (°C)
W_P	Power of SRC pump (kW)
W_T	Power of steam turbine (kW)
$W_{P.ORC}$	Power of ORC pump (kW)
$W_{T.ORC}$	The output power of the ORC turbine (kW)
Greek symbol	
η	thermal efficiency of SRC (–)
η_{exe}	Exergy efficiency (–)
η_{ORC}	Thermal efficiency of ORC (–)
η_p	Mechanical efficiency of the SRC pump (–)
$\eta_{p.ORC}$	Mechanical efficiency of the ORC pump (–)
η_T	Mechanical efficiency of the SRC turbine (–)
$\eta_{T.ORC}$	Mechanical efficiency of the ORC turbine (–)
Abbreviations	
AH	Auxiliary Heater
BRC	Binay Rankine Cycle
H. Ex	Heat Exchanger
MSP	Molten Salt Pump
OFP	Organic Fluid Pump
OFT	Organic Fluid Turbine
ORC	Organic Rankine Cycle
PTC	Parabolic Trough Solar Collectors
SRC	Steam Rankine Cycle
ST	Steam Turbine
TWV	Three Way Valve
WP	Water Pump