A comparison of using organic Rankine and Kalina cycles as bottom cycles in a solar-powered steam Rankine cycle

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Abstract
This study has attempted to compare two thermodynamic cascade cycles, which in this paper are presented as Systems A and B. System A was consisted of a steam Rankine cycle (SRC) as a top cycle and an organic Rankine cycle (ORC) as a bottom cycle. System B was consisted of the same SRC as the top cycle and a Kalina cycle as the bottom cycle. The comparison of both systems has been made by changing a number of parameters. The chosen parameters for this comparison were bottom cycle mass flow rate, aperture area of the collector, top cycle condensing pressure, and bottom cycle turbine inlet and outlet pressures. Also, a short economic evaluation has been made between two proposed cascade cycles (Systems A and B). The result indicated that the Kalina cycle shows superiority over the ORC as the bottom cycle in a solar-driven SRC. Furthermore, it was determined that the most effective parameters on the overall efficiency of the systems are the condensing pressure of the top cycle and the outlet pressure of the bottom cycle turbine. Also, among the chosen parameters, the outlet pressure of the bottom cycle turbine had the highest effect on the efficiency of the bottom cycles. Considering the economic aspect, the results showed that the levelized costs of energy for both systems are quite equal at 0.011 ($/kWh).

Keywords
Kalina cycle, ORC cycle, solar-driven power cycle, steam Rankine cycle

1 | INTRODUCTION

The global demand for energy is growing considerably. In 2018 energy consumption had its fastest jump of 2.13%. CO₂ emission rises as well. Undoubtedly, fossil fuels are not a suitable way to meet this demand regarding their environmental threats. Therefore, renewable energies could play a crucial role in producing clean energy. After wind energy, solar energy is the most...
common source of energy in renewables. It has a wide variety of applications, making it one of the most promising energy sources.\textsuperscript{9–11} According to the IEA (International Energy Agency), generating electricity by solar thermal energy has increased significantly over the past decades. Also, during this time, the number of concentrating solar power (CSP) plants has risen considerably.\textsuperscript{12} Among different types of solar thermal collectors, parabolic trough collectors are known as the most popular. Also, they are the most established solar thermal technology for power generation.\textsuperscript{13} Therefore, PTC has been chosen for this study.

Obtaining thermal energy with solar collectors comes with tremendous amounts of effort, and it is quite costly. Thus, it is vital to utilize every proportion of this energy as much as possible.\textsuperscript{14} Cascade cycles are an appropriate and effective way to exploit thermal energy as they employ the rejected heat from the top cycle to generate power in the bottom cycle.\textsuperscript{15} The rejected heat from the top cycle in cascade cycles is usually low-grade heat, which is not suitable to power a conventional steam Rankine cycle (SRC). Therefore, low-grade heat cycles such as the organic Rankine cycle (OR) and Kalina cycle would be the proper options.\textsuperscript{16,17} Moreover, recycling waste heat energy from the top cycle can reduce greenhouse gas release and improve energy efficiency.\textsuperscript{18}

The Kalina cycle was introduced in the early 1980s using a mixture of ammonia and water as the working fluid. Kalina cycle had remarkable improvements in the matter of thermal power plant design, and it can be considered as a rival to the ORC.\textsuperscript{19,20} Ashouri et al.\textsuperscript{21} investigated the performance of a solar powered system from the thermodynamic and economic aspects. The system consisted of a small-scale PTC, a Kalina cycle, a storage system, and an auxiliary heater. The system was assessed throughout a year and also compared with a fuel-driven Kalina cycle. A study assessed and optimized exergoeconomic parameters of solar driven Kalina cycle. According to the result of this study, exergoeconomic performance of the system is not highly affected by the solar collectors.\textsuperscript{22} Ghorbani et al.\textsuperscript{23} deployed a Kalina cycle driven by flat plate collectors and an auxiliary heater with a multistage desalination process in a cogeneration system. The result indicates that the collectors and auxiliary heater are the main sources of the exergy destruction. A study presented a cogeneration system, which used PTC and liquefied natural gas regasification. System outputs were power and cooling, which were generated by a Kalina-based cooling and power generation cycle and a gas turbine.\textsuperscript{24} Ghaedi and Rostamzadeh\textsuperscript{25} used an SGSP (salinity-gradient solar pond) to power a cogeneration system integrated with an ORC and a Kalina cycle with a reverse osmosis unit for power and freshwater production. Feng et al.\textsuperscript{26} performed a triple optimization on the Kalina cycle by changing heat transfer area distribution, in which the overall output power, thermal efficiency, and ecological function were improved 17.27%, 5.79%, and 1.05%, respectively.

Number of studies investigated the cascade cycles. Delgado-Torres and García-Rodríguez\textsuperscript{27} ran a desalination unit by using a solar-driven double-cascade ORC. Al-Sulaiman\textsuperscript{28} compared the size of the solar field of a SRC with a binary vapors cycle and an SRC with vacuum condensing pressure. The result showed that the smallest solar field belonged to the SRC with vacuum condensing pressure. Li et al. investigated the thermodynamic, economic, and energy analyses of a novel cascade cycle integrated with hybrid solar collectors (concentrating and non-concentrating).\textsuperscript{29} A study\textsuperscript{30} simulated a double cascade cycle with an SRC as the top cycle and ORC as the bottom. Also, a screw expander was deployed in the SRC. The cascade cycle was coupled with parabolic trough collectors (PTC) as the power source. In this system, steam was directly generated in the collectors and expanded in the screw expander. Li et al.\textsuperscript{31} investigated the efficiency of a cascade thermodynamic cycle, which included a screw expander and a phase change material storage tank coupled with PTC in different conditions. Also, an optimization was conducted to determine the optimum evaporation temperatures of the SRC and ORC in different solar irradiations. Habibi et al.\textsuperscript{32} presented a novel configuration consisting of a partial evaporation Rankine cycle, an ORC, and a storage system. The system was driven by PTC and was investigated from the energy and exergoeconomic aspects. In this novel study, a screw expander was employed instead of the conventional steam turbines in the partial evaporation Rankine cycle. Moreover, the effect of different parameters such as solar intensity, aperture area of the collector, and condensing temperature on the thermoeconomic performance of the system was assessed via conducting a parametric study. In addition, a multi-objective optimization was conducted. Yang et al.\textsuperscript{33} presented a solar-powered cascade system in which a supercritical carbon dioxide Brayton cycle and an SRC were used as the top and bottom cycles, respectively. To generate high pressure and temperature steam to drive the SRC, the high-temperature exhaust gas was deployed to transfer heat to a sCO\textsubscript{2}/water heat exchanger. Furthermore, a comparison was conducted between a simple sCO\textsubscript{2} system and the presented cascade system. In this regard, Wu et al.\textsuperscript{34} optimized a cascade Kalina-ORC cycle in which Kalina cycle was as top cycle and ORC was as the bottom cycle. The optimization was performed based on maximizing the
overall power output of the proposed system on heat exchangers’ tube diameter, which led to enhancing net power up to 16.17%.

Some studies compared the Kalina cycle and ORC. Ozahi et al.35 carried out a thermodynamic and thermo-economic analysis of combined cycles, a gas turbine-ORC, and a gas turbine-Kalina cycle. Ambriz-Diaz et al.36 presented a comparative study in which the performances of a Kalina cycle and an ORC were compared thermodynamically and economically. The cycles were coupled to a polygeneration system powered by low-grade temperature geothermal energy. Another study used structural parametric thermo-economic optimization to choose the best technology for waste heat recovery for a Brazilian diesel engine power plant. The compared technologies were a conventional Rankine cycle, an ORC, and a Kalina cycle.37 The articles, which were close to the subject of this study, are reviewed above. The purpose of the present work is described in the following.

In this study, Kalina cycle and ORC have been compared as the bottom cycle in a solar-powered SRC. In this regard, two cascade thermodynamic cycles, which are called systems A and B (in this paper), have been designed, simulated, and then compared. Furthermore, a sensitivity analysis was carried out by choosing five different parameters. Then the chosen parameters have been changed to determine the effect of each parameter on the efficiency of each individual bottom cycle (Kalina and ORC). Moreover, the effect of the chosen parameters on the overall efficiency of systems A and B has been assessed as well. The chosen parameters were bottom cycle mass flow rate, aperture area of the collector, top cycle condensing pressure, and bottom cycle turbine inlet and outlet pressures. Also, the economic performance of both systems has been analyzed and compared briefly. In most of the previous studies, ORC and Kalina cycle were assessed separately in terms of sensitivity analysis, economic aspect, and performance in a solar cascade cycle. To the best of the author’s knowledge, rare works have been done on the comparison of both cycles in an equivalent condition. This study has attempted to compare both cycles in an equivalent condition. The findings and thermodynamic analyses of this study are presented in the following.

2 | SYSTEM DESCRIPTION

Since in SRCs, a considerable amount of energy is rejected and wasted, the utilization of this rejected heat is vital. Although this low-grade heat could not be used to generate power in a conventional steam cycle, it could be recycled to run a low-grade heat thermodynamic cycle. Therefore, cascade thermodynamic cycles in which a low-grade heat cycle works as the bottom cycle are suitable for utilizing this amount of rejected heat. This study has attempted to compare two systems in which Kalina and ORC cycles are considered as the bottom cycles and an SRC as the top cycle. Systems are called System A and B in this study, and their description is presented below.

2.1 | System A description

In System A, an SRC is considered as the top cycle, and the rejected heat from the condenser of the top cycle is used to run the bottom cycle, which is an ORC. To use the ORC as the bottom cycle in cascade cycles, refrigerants could be good options due to their low boiling point temperature. Among different working fluids, R134a seems to have good performance compared to the others.28 Also, because of its low boiling point temperature (−26.3°C), it is an appropriate option to be used to compare the ORC and Kalina cycle. Therefore, it has been chosen as the working fluid of the ORC in this system. Moreover, the layout of this system, which is simulated in Aspen HYSYS 11.0, is illustrated in Figure 1. According to Figure 1, the outlet stream of the SRC pump (stream number 1) enters the SRC-PTC heat exchanger (E-101) to be superheated. Then the superheated vapor (stream number 2) enters the SRC turbine (K-100) to generate power. Afterward, the outlet stream of the SRC turbine enters the SRC-ORC heat exchanger (E-101) to produce subcooled liquid (stream number 4) and run the ORC simultaneously. The thermodynamic process in the ORC is the same as in the SRC; however, the working fluid in the ORC has a much lower boiling point temperature compared to water. The organic fluid (O1) enters the SRC-ORC heat exchanger (E101) to be superheated. Then, the superheated vapor (O2) goes toward the ORC turbine to generate power. The dead steam (O3) enters the ORC condenser to be liquefied (O4) to be pumped by the ORC pump (P-101) and complete the process of the ORC.

2.2 | System B description

In system B, the SRC is considered as the top cycle, is the same as system A, and the thermodynamic processes of both SRCs are the same. The main difference between these systems is that the bottom cycle in system B has been considered to be a Kalina cycle. Since the Kalina cycle is suitable for low-grade heat, it could be a good
option to be compared with an ORC, which uses a working fluid with a low boiling point temperature such as R134a. Furthermore, the layout of system B, which has been simulated in Aspen HYSYS 11.0, is illustrated in Figure 2. Kalina cycle uses a binary fluid with a certain ratio of water and ammonia as the working fluid. The efficiency of a Kalina cycle depends on the ammonia’s mass fraction in the basic solution. In this study, the mass fraction of ammonia has been considered to be 82%, since it showed superiority over other mass fractions in the survey at Ref. [38].

The process in the Kalina cycle is a bit different from the ORC. According to Figure 2, the binary fluid (K1) enters the SRC-Kalina heat exchanger (E101) to produce steam, then the steam and liquid mixture (K2) goes to the separator (V100). After entering the separator, the steam leaves from the top (K3), and the liquid goes from the bottom (K5). The top stream (K3) has more ammonia content, because of the lower boiling point temperature of ammonia compared to the water. Then the top stream passes through the turbine to generate power, then the dead steam stream (K4) leaves the turbine. It is possible to use normal back-pressure turbines because the molecular weights of the ammonia and water are close. Also, no special material is needed for the water and ammonia mixture. The outlet liquid stream of the separator (K5) enters the high-temperature recuperator (E102) to preheat the SRC-Kalina heat exchanger inlet stream (K1). Afterward, the rich ammonia stream (K4) and lean ammonia stream solutions (K6) are merged in the mixer. The outlet stream of the mixer (K7) enters the low-temperature recuperator (E103) to preheat the pumped liquid exiting the Kalina pump (K10). Moreover, high temperature (HT) and low temperature (LT) recuperators have been used to enhance the energy efficiency of the proposed system.

2.3 | Input heat source

It should be noted that both systems have been powered by parabolic trough collectors in which a single one has a
width of 5.76 m and a length of 12.27 m. Also, Therminol-VP1 is chosen as the heat transfer fluid of the PTC. The solar beam radiation is assumed to be constant at 750 W/m². The oil leaves the collector at a temperature of 321.9°C (595.05 K). The solar field consists of 68 collectors in series and 55 collectors in parallel, which is able to produce 150 MW of power. The discussion about the modeling of the PTC and the validation of the model will be mentioned in the subsection of “Modeling” with more details.

### 3 | MODELING

In this subsection, the modeling of the PTC and two systems (A, B) are presented. Modeling and simulation have been done by considering a number of assumptions, which are presented below.

- Pressure drop of the equipment has been neglected.
- Heat losses have not been considered in heat exchangers.
- Both systems operate in a steady-state condition.
- Solar beam radiation has been considered to be constant.
- The isentropic efficiencies of the pumps and turbines have been considered constant and specified.
- Mechanical losses in pumps and turbines have been neglected.
- Potential and kinetic energy changes have been neglected.

#### 3.1 | PTC modeling

The mathematical modeling of the PTC, which has been done by the Engineering Equation Solver (EES) software, is based on the article in Ref. [28]. The input parameters for modeling the single PTC are listed in Table 1. It has been assumed that the collector is located in the north-side direction with the constant solar beam radiation of 750 (W/m²). Moreover, the PTC has been analyzed in the steady-state condition.

| Parameters                          | Unit | Value |
|-------------------------------------|------|-------|
| Collector width                     | m    | 5.76  |
| Collector length                    | m    | 12.27 |
| Size of the receiver tube           | m    | 0.066 |
| Thickness of the receiver tube      | m    | 0.004 |
| Thickness of the cover glass        | m    | 0.006 |
| Size of the cover glass             | m    | 0.115 |
| Emittance of the cover glass        | -    | 0.86  |
| Emittance of the receiver           | -    | 0.15  |
| Mirror reflectance                  | -    | 0.94  |
| Intercept factor                    | -    | 0.93  |
| Glass cover transmittance           | -    | 0.96  |
| Absorbance of the receiver          | -    | 0.96  |
| Modifier of incidence angle         | -    | 1     |
| Number of collector in series       | -    | 68    |
| Number of collector in parallel     | -    | 55    |
| Flow rate of each row               | kg/s | 6     |
| Ambient temperature                 | °C   | 25    |
| Solar beam radiation                | W/m² | 750   |
| Type of fluid                       |      | Therminol-VP1 |

The aperture area is defined as

\[ A_{ap} = A_r \left[ S \right] \left( \frac{A_r}{A_{ap}} \right) \left( T_l - T_{amb} \right), \] (2)

where \( A_{ap} \) is the aperture area of the PTC, and \( A_r \) is the receiver area. The heat absorbed, which is gained by the receiver, is determined from the following equation:

\[ S = G_b \eta_r. \] (3)

The \( G_b \) is the solar beam radiation. The receiver efficiency could be calculated from the following equation:

\[ \eta_r = \rho_c \gamma \tau \alpha, \] (4)

where \( \rho_c, \gamma, \tau, \alpha, \) and \( K_y \) are the mirror’s reflectance, intercept factor, the transmittance of the glass cover, the absorbance of the receiver, and the incidence angle modifier, respectively, which their values are stated in Table 1. The aperture area is defined as

\[ Q_u = \dot{m} \left( C_p, T_o - C_p, T_l \right). \] (1)

The \( \dot{m}, C_p, \) and \( T \) are the PTC flow rate, the specific heat of the hot thermal fluid, and the temperature, respectively. Also, the useful energy could be calculated from the following equation

\[ Q_u = A_{ap} F_R \left[ S - \frac{A_r}{A_{ap}} U_L (T_l - T_{amb}) \right]. \] (2)
\[ A_{ap} = (W - D_{co})L, \]  
(5)

where \( W, D_{co}, \) and \( L \) are related to the size of the PTC, which are mentioned in Table 1. The heat removal factor could be determined from the following equation:

\[ F_R = \frac{m_C P}{A_T U_L} \left[ 1 - \exp \left( \frac{-A_r U_l F_i}{m_C P} \right) \right], \]  
(6)

where \( F_i \) is the factor of collector efficiency, which is obtained from the following equation:

\[ F_i = \frac{U_R}{U_L}. \]  
(7)

The PTC heat loss coefficient between the surroundings and receiver is defined as

\[ U_L = \left[ \frac{A_r}{(h_{c,ca} + h_{t,ca}) A_c} + \frac{1}{h_{r,cr}} \right]^{-1}. \]  
(8)

The PTC heat loss coefficient related to the radiation between the surroundings and the cover glass is calculated from the following equation:

\[ h_{t,ca} = \varepsilon_{cv} \sigma (T_h + T_o) (T_o^2 + T_c^2), \]  
(9)

where \( \varepsilon_{cv} \) is the emittance of the cover glass, and \( \sigma \) is the constant of the Stefan–Boltzmann. The PTC heat loss coefficient related to the radiation between the receiver and the cover glass is determined from the following equation:

\[ h_{r,cr} = \frac{\sigma (T_c + T_{r,av}) (T_c^2 + T_{r,av}^2)}{\frac{1}{\varepsilon_r} + \frac{A_r}{A_c} \left( \frac{1}{\varepsilon_{cv}} - 1 \right)}, \]  
(10)

where \( \varepsilon_r \) is the emittance of the receiver. The PTC heat loss coefficient related to the connection between the surroundings and the cover glass is obtained from the following equation:

\[ h_{c,ca} = \left( \frac{Nus_k_{air}}{D_{c,co}} \right). \]  
(11)

The \( Nus \) and \( k_{air} \) represent the Nusselt number and thermal conductivity of the air, respectively. The overall heat loss coefficient between the ambient and the hot thermal fluid is obtained as follows:

\[ U_o = \left[ \frac{1}{U_L} + \frac{D_{t,co}}{h_{c,r,i} D_{r,i}} + \left( \frac{D_{t,co}}{2k_r} \ln \left( \frac{D_{t,co}}{D_{r,i}} \right) \right) \right]^{-1}, \]  
(12)

where \( h_{c,r,in} \) is the heat loss coefficient between the cover glass and receiver, which is determined from the following equation:

\[ h_{c,r,in} = \frac{Nus_k_r}{D_{r,i}}. \]  
(13)

The temperature of the cover glass is calculated from the following equation:

\[ T_c = \frac{h_{r,cr} T_{r,a} + \frac{A_r}{A_c} (h_{c,ca} + h_{r,ca}) T_{amb}}{h_{r,cr} + \frac{A_r}{A_c} (h_{c,ca} + h_{r,ca})}. \]  
(14)

Finally, the collector performance is obtained from the following equation:

\[ \eta_{Collector} = \frac{Q_u}{G_b A_{ap}}. \]  
(15)

### 3.2 Thermodynamic simulation

The thermodynamic simulation of the systems has been conducted by Aspen Hysys 11.0. The input data for the SRC, ORC, and Kalina cycle are given in Tables 2–4, respectively. The following equations have been used to assess the performance of the systems. The produced power by the steam turbine is defined as

\[ W_{st} = m_{st} (h_2 - h_3), \]  
(16)

### Table 2 Steam Rankine cycle input data

| Parameter                     | Value       |
|-------------------------------|-------------|
| Turbine isentropic efficiency | 75%         |
| Pump isentropic efficiency    | 75%         |
| Mass flow rate                | 61 kg/s     |
| Turbine inlet pressure        | 100 (Bar)   |
| Turbine outlet pressure       | 1 (Bar)     |
| Turbine inlet temperature     | 594.92 K    |
| Condensing temperature        | 372.74 K    |
| Input heat from the collector | 150 MW      |
where $h$ and $\dot{m}_{st}$ indicate enthalpy and steam mass flow rate. The net power produced by the SRC is described in the following equation:

$$W_{\text{net,src}} = \dot{W}_{st} - \dot{W}_{sp}, \quad (17)$$

where the subscripts $st$ and $sp$ indicate the SRC turbine and SRC pump. The required work for the steam Rankine pump is defined in the following equation:

$$\dot{W}_{sp} = \dot{m}_{sp} (h_1 - h_4). \quad (18)$$

The net power generated by the ORC is described in the following equation:

$$W_{\text{net,orc}} = \dot{W}_{ot} - \dot{W}_{op}, \quad (19)$$

The efficiency of the systems, A and B, is defined as

$$\eta = \frac{W_{\text{net}}}{Q_{\text{solar}}}. \quad (23)$$

3.3 | Economic equations

A short economic evaluation has been performed to evaluate the cost-effectiveness of the proposed systems. The levelized cost of energy (LCOE) has been calculated from the following equation:

$$LCOE = \frac{\sum_{i=0}^{n-1} C_i + C_o}{W_{\text{net}}}, \quad (24)$$

where $C_c$, $n$, $r$, and $C_o$ are the capital cost investment, system’s lifetime, discount rate, and the operating and maintenance cost, respectively.

The net present value (NPV) and the amount of overall income over a year have been determined as follows:

$$NPV = \sum_{i=0}^{n-1} \frac{C_i}{(1 + r)^i}, \quad (25)$$

3.3.1 | Organic Rankine cycle input data

| Parameter                        | Value          |
|----------------------------------|----------------|
| Turbine isentropic efficiency    | 75%            |
| Pump isentropic efficiency       | 75%            |
| Mass flow rate                   | 569 kg/s       |
| Turbine inlet pressure           | 30 Bar         |
| Turbine outlet pressure          | 6.6 Bar        |
| Turbine inlet temperature        | 359.07 K       |
| Condensing temperature           | 293.03 K       |
| Cooling water input temperature  | 286.15 K       |
| Cooling water output temperature | 292.85         |

3.3.2 | Kalina cycle input data

| Parameter                        | Value          |
|----------------------------------|----------------|
| Turbine isentropic efficiency    | 75%            |
| Pump isentropic efficiency       | 75%            |
| Mass flow rate                   | 158 kg/s       |
| Turbine inlet pressure           | 30 Bar         |
| Turbine outlet pressure          | 6.6 Bar        |
| Turbine inlet temperature        | 366.42 K       |
| Condensing temperature           | 281.15 K       |
| Ammonia mass fraction in solution| 82%            |
| Cooling water input temperature  | 278.15 K       |
| Cooling water output temperature | 281.15         |
\[ C_t = C_{sv} + C_E - C_0 - C_v, \]  

(26)

where \( C_{sv} \) and \( C_E \) denote the salvage value and yearly electrical income, respectively. The economic parameters and assumptions, which have been used in this study, are listed in Table 5.

### 3.4 Validation

In this subsection, the validations of the PTC, Kalina cycle, and the SRC-ORC cascade cycle have been presented. To assess the accuracy of the simulated systems, mean absolute percentage error has been calculated for each validation, and the formula is presented below:

\[ \text{Err} = \frac{1}{n} \sum_{i=1}^{n} \left| \frac{R_v - C_v}{R_v} \right|, \]  

(27)

where \( \text{Err} \) represents the mean absolute percentage error (MAPE), also \( R_v \) and \( C_v \) are the reference value and the calculated value, respectively.

#### 3.4.1 PTC validation

The output results of the PTC collector have been compared with the experimental data of Ref. [42], which is used the properties of the LS-2 PTC. The thermal efficiency comparison between the experimental work and this study, which is shown in Figure 3, demonstrates a good modeling accuracy. The MAPE of the PTC validation has been calculated about 1.56%.

#### 3.4.2 Kalina cycle validation

The output results of the proposed Kalina cycle have been compared with Ref. [38]. In this matter, the electrical efficiency of the proposed Kalina cycle has been compared with “case 1” of the Ref. [38] as the turbine inlet pressure of the cycle was increased. As it is clear in Figure 4, the simulation of the proposed Kalina cycle is accurate. Also, the Mean Absolute Percentage Error, calculated from Equation (27), is approximately 1.33%.

| TABLE 5 | The economic parameters and assumptions\textsuperscript{15,41} |
|----------|-------------------|
| **Parameters** | **Value** |
| PTC cost ($/m^2) | 280 |
| Operating and maintenance cost ($/year) | 2% of equipment cost |
| Electricity price ($/kWh) | 0.21 |
| Salvage value ($) | 20% of equipment cost |
| System lifetime (years) | 25 |
| Inflation rate (%/year) | 4.4 |
| Discount rate (%/year) | 6 |
| ORC turbine cost ($) | \( 1,476,000 \left( \frac{a}{m^2} \right)^0.5 \left( \frac{SP}{25} \right)^1.1 \) |
| Generator cost ($) | \( 2447(kW)^{0.49} \) |
| Condenser cost ($) | \( 597(kW)^{0.68} \) |
| Heat exchanger cost ($) | \( 235(kW)^{0.75} \) |
| SRC turbine cost ($) | \( 31093(kW)^{0.41} \) |

Abbreviations: ORC, organic Rankine cycle; PTC, parabolic trough collector; SRC, steam Rankine cycle.

**FIGURE 3** The comparison of results of PTC modeling in this study with the experimental data of Ref. [42]. PTC, parabolic trough collector.
3.4.3 Validation of SRC-ORC cascade cycle

As it has been shown in Figure 5, net electrical power versus collected heat from the solar field for the proposed system has been calculated and compared with Ref. [28]. The Mean Absolute Percentage Error calculated from Equation (27) is about 1.91%, which indicates that the simulation of the cascade cycle is reliable.

4 RESULTS AND DISCUSSIONS

In this simulation, firstly, the modeling of the PTC has been conducted in the EES software. The solar radiation has been assumed to be constant (750 W/m²). The solar collectors are able to produce 150 MW thermal energy. The calculated solar energy has been given to Hysys Aspen 11.0 as the input heat to generate power through the SRC. Then the rejected heat from the SRC condenser is deployed to generate power through either the ORC or the Kalina cycle (Systems A and B).

Furthermore, the effect of various operational parameters has been investigated on the overall performance of the systems. Also, the performance of the bottom cycles (ORC and Kalina) has been assessed separately as well. The influence of the top cycle condensing pressure, bottom cycles mass flow rate, bottom cycles turbine inlet and outlet pressures, and total aperture area of the PTCs have been evaluated to compare the performance of the systems. It should be noted that in each subsection, to have a clear assessment of the effect of parameters on the performance of the systems, only the investigated parameter has been changed, and other parameters have been remained constant, and their values are given in Tables 2–4. For instance, in Section 4.1, only the condensing pressure of the top cycle has been changed,
and the bottom cycles mass flow rate, bottom cycles turbine inlet and outlet pressures, and total aperture area of the PTCs have been remained constant, and their values are equal to the values given in Tables 2–4. The results of this comparison are presented as follows:

4.1 The effect of the condensing pressure of the top cycle

In this subsection, the impact of the top cycle condensing pressure on the overall performances of the systems (A and B) is evaluated.

The condensing pressure of the SRC has been changed from 1 to 7 Bar. This increment of the condensing pressure led to higher heat duty in the heat exchanger that combines the top and the bottom cycles (E101). In other words, increasing top cycle condensing pressure boosts the amount of heat that is transferred to the bottom cycles.

Figure 6 shows the net power output of the bottom cycles at various SRC condensing pressure. According to Figure 6, the power produced by the bottom cycles increases, due to the increase of the heat exchanger duty. Increasing the SRC condensing pressure from 1 to 7 Bar increases the net power production of systems B and A bottom cycles (Kalina and ORC) by 8.71% and 16.25%, respectively. This shows that the ORC is more affected by the amount of input heat, which comes from the heat exchanger.

The efficiency of the bottom cycles versus the condensing pressure is illustrated in Figure 7. Although the net power produced by the Kalina cycle increases when the condensing pressure of the SRC rises, the Kalina cycle efficiency decreases. In contrast, the efficiency of the ORC increases when the condensing pressure of the SRC rises. However, the efficiency of the Kalina cycle is higher than the ORC and remains higher in the range of 1–7 Bar.

**Figure 6** Net power output of the bottom cycles at various SRC condensing pressures. SRC, steam Rankine cycle

**Figure 7** The efficiency of the bottom cycles at various SRC condensing pressures. SRC, steam Rankine cycle
Nevertheless, by increasing the condensing pressure of the top cycle, the overall efficiencies of both systems are reduced (Figure 8). The overall efficiencies of systems A and B have been (calculated by Equation 23) reduced by 24.75% and 25.17%, respectively. Since both systems are more efficient at lower condensing pressures of the top cycle, in the further investigations, the condensing pressure of the SRC has been considered to be 1 Bar to gain the maximum efficiency.

4.2 The effect of the bottom cycle mass flow rate

In this subsection, the impact of the bottom cycle mass flow rate on the overall performances of the systems and the performances of the bottom cycles are shown.

Since the mass flow rates of the ORC and the Kalina cycle are far from each other, it would not be a quite correct comparison to change their mass flow rate with equal amounts and analyze the results. Therefore, both their mass flow rates have been increased by 10% to observe the impact of changing the mass flow rate on the bottom cycles’ performance and the overall performance of the systems. Figure 9 shows the result of this examination. As it is clear in Figure 9, the efficiency of system B increases when the mass flow rate of the Kalina cycle increases. On the other hand, the efficiency of system A reduces as the mass flow rate of the ORC increases.

In system A, as the mass flow rate of the ORC increases, the temperature of stream O2 decreases from 99.43 to 85.92°C, yet the vapor fraction of the stream remains equal to 1. By decreasing the temperature of stream O2, the enthalpy of this stream reduces. Moreover, the reduction of
the temperature in stream O2 results in the reduction of the temperature in stream O3 from 41.52 to 24.8°C. Although the temperature difference between streams O2 and O3 increases, the difference between the enthalpy of these two streams decreases by approximately 12%. According to Equation (16), the produced power is related to both enthalpy difference and the mass flow rate. In this situation, the enthalpy difference has been reduced; the mass flow rate has been increased. However, the increase of the mass flow rate is not able to compensate for the reduction in the enthalpy difference, so the power produced by the ORC decreases.

In System B, the changes are a bit different. As the mass flow rate of the Kalina cycle rises, the temperature and the vapor fraction of stream K2 decrease. Although the vapor fraction of stream K2 reduces, the amount of steam that leaves the separator rises. This means a higher amount of steam enters the turbine. As the mass flow rate of the Kalina cycle rises by 10%, the vapor fraction of stream K2 decreases from 0.6 to 0.56. Although the vapor fraction of stream K2 falls from 0.6 to 0.56, the mass flow rate of the K3 stream (turbine inlet stream) increases from 79.03 to 81.79 kg/s which means more power is produced in the Kalina cycle turbine. Additionally, the mass flow of stream K5 (liquid content of stream K2) increases from 53.96 to 64.2 kg/s. Furthermore, when the mass flow rates of streams K3 and K5 rise, the duties of the HT and LT recuperators rise as well, which leads to growth in the temperatures of streams K11 and K1. In other words, stream K9 receives more amount of heat in the preheating process when it passes through LT and HT recuperators. So, the input stream of E101 (K1) becomes hotter. Therefore, the power produced by the Kalina cycle increases.

Considering the changes in the net power produced by the bottom cycles, as the mass flow rate of the bottom cycles increased by 10%, the net power produced by the Kalina cycle rose by 1.42%. In contrast, the net power produced by the ORC was reduced by 4.41%, which evidently shows that the ORC is more sensitive to changes in the mass flow rate as compared with the Kalina. Also, system B has a better performance as compared to system A.

4.3 The effect of the bottom cycle turbine inlet pressure

In this subsection, the impact of the bottom cycle turbine inlet pressure on the overall performance of the systems and the performance of the bottom cycles are examined. Due to the close operating pressure range of both the ORC and Kalina cycle, the turbine inlet pressure for both of these cycles has been increased from 30 to 35 Bar.

The changes in the efficiency of the Kalina cycle and ORC as the inlet turbine pressure of the bottom cycles increases are shown in Figure 10. Also, the comparison between the overall efficiency of systems A and B is illustrated in Figure 11. In system A, as the turbine inlet pressure rises by 16.66% (30–35 Bar), the power produced by the ORC increases by 6.93%. On the other hand, the ORC pump required work rises by 21.14%. The ORC's efficiency and system A's overall efficiency increases by 5% and 1.18%, respectively.

In system B, as the turbine inlet pressure increases by 16.66% (30–35 Bar), the power produced by the Kalina cycle rises about 6.99%. At the same time, the Kalina pump required work increases by 21.36%. The Kalina cycle's efficiency and system B's overall efficiency increased about 6.1% and 1.66%, respectively. This comparison shows that the Kalina is more sensitive to the turbine inlet pressure, as clearly can be seen in Figure 10. Also, system B has a better performance compared to system A.
4.4 The effect of the bottom cycle turbine outlet pressure

In this subsection, the impact of the bottom cycle turbine outlet pressure on the overall performances of the systems and the performances of the bottom cycles is shown.

The outlet pressure of the bottom cycle turbine has been changed from 5.6 to 9.6 Bar to assess the effect of the proposed parameter on the overall performances of the systems and on the efficiency of the bottom cycles. Figure 12 illustrates the changes in the efficiency of the bottom cycle when the outlet pressure of the bottom cycles increases. Also, the comparison between the overall efficiencies of the systems is shown in Figure 13.

In system A, as the turbine inlet pressure rises (5.6–9.6 Bar), the power produced by the ORC reduces by 33.83%. On the other hand, the ORC pump required work decreases by 16.39%. The efficiency of the ORC and the overall efficiency of system A reduce by 36.5% and 9.41%, respectively.

In System B, when the turbine outlet pressure increases, the power generated by the Kalina cycle reduces by 29.67%, and the Kalina pump required work decreases by 16.41%. The Kalina cycle's efficiency and System B's overall efficiency decline by 30.4% and 8.75%, respectively.

This comparison indicates that the ORC is more sensitive to the change of outlet turbine pressure. Furthermore, the changes in the pump required work of both bottom cycles are approximately equal. This means the power produced by the ORC turbine is more dependent on its outlet pressure.

Moreover, when the outlet pressure of the turbine increases, the difference between the generated powers by the ORC and Kalina cycle rises. When the turbine outlet pressure increases from 5.6 to 9.6 Bar, the difference between produced powers by the bottom cycles increases from 0.96 to 1.23 MW. Since the changes in power consumption of their
pumps are approximately equal, the difference between their efficiencies increases as well. In addition, System B has better performance compared to System A as well as the previous comparisons.

4.5 | The effect of the total aperture area of the collector

In this subsection, the impact of the total aperture area of the PTCs on the overall performances of the systems and the performances of the bottom cycles has been evaluated.

Figure 14 clearly shows the changes in the efficiencies of the Kalina cycle and ORC as the aperture area of the PTCs increases. Also, the comparison between the overall efficiencies of systems A and B is illustrated in Figure 15.

The total aperture area of the PTCs has been increased from 258,771 to 282,641 m², which results in an increase in the input heat of the systems from 150 to 162 MW. When the input heat from the PTCs reaches 162 MW, the heat transfer oil temperature reaches 388.36°C, which is close to the critical temperature of the oil (390°C).

When the aperture area of the PTCs expands, the temperature of stream 2 in the SRC increases, which results in the increase of the SRC turbine output work by 10.56%, and the increase of the heat duty of the E101 by 7.18%.

In system A, when the heat duty of E101 increases, more amount of heat is transferred to the ORC. The temperature of the entering stream to the ORC turbine (O2) increases from 85.92 to 94.77°C. The growth in stream O2 temperature leads to the increase of the produced power of the ORC by 9.35%. Moreover, system A’s efficiency and the ORC’s efficiency rise by 2.4% and 2.8%, respectively.

In system B, as the heat duty of E101 rises, the temperature and vapor fraction of stream K2 increases, which means more amount of steam goes through the
turbine (stream K3). The mass flow rate and temperature of stream K3 increased by 4.24% and 3.43%, respectively. These enlargements lead to a rise of 5.19% in power produced by the Kalina cycle. The notable result is that since the mass flow rate of the Kalina cycle is fixed and it has a constant value, by expansion in stream K3’s mass flow rate, stream K5’s mass flow rate decreases consequently. The decline in stream K5’s mass flow rate leads to the lower heat duty in E102 (HT recuperator). On the other hand, as the temperature of the stream K3 increases, the temperature of K7 stream increases as well, which leads to a rise in the heat duty of E103 (LT recuperator). Consequently, stream K10 receives more heat in the E103 (LT recuperator).

Although K10 receives more heat in the E103 (LT recuperator), the temperature of the stream which goes to E101 (K1) reduces, because the heat duty of E102 plays a more important role in the process of preheating, and most of the required heat for preheating comes from this heat exchanger. Therefore, when the heat duty of E102 falls, the growth in the heat duty of E103 is not able to provide the needed heat for the preheating process single-handedly. Another considerable result is that, when the aperture area of the PTCs increases, the power produced by the Kalina cycle increases by 5.19%, however, the efficiency of the Kalina cycle decreases by 1.56%. Because according to Equation (23), the efficiency is also affected by the input heat. In this case, the rate of climb in the Kalina’s output work is not as fast as the rate of climb in the input heat. This can be explained by the decrease in the E102 heat duty. In other words, when the temperature of the input stream of the E101 (K1) decreases, more amount of heat is spent on raising the temperature of this stream. Even though the efficiency of the Kalina cycle decreases, the efficiency of system B increases about 1.5%, which means the increase of the SRC efficiency could compensate for the drop in the Kalina efficiency.

As it is clear in Figure 14, the difference between the efficiencies of the bottom cycles reduces as the aperture area rises. Even though the efficiency of the Kalina decreases, it remains higher than ORC until the heat transfer oil reaches its critical temperature of 390°C.

### 4.6 Economic results

In this sub-section, the results of the economic evaluation are presented. It should be noted that the financial assessment has been performed based on the best cases of systems A and B from the energy point of view. Also, the best case of the systems has been obtained according to the sensitivity analysis, that means the value of the parameters has been selected somehow to lead to the highest energy efficiency.

As it is shown in Table 6, the economic results of both systems A and B are almost the same. Both systems’ LCOE and payback period are 0.011 $/kWh and 2 years, respectively. However, the amount of NPV for system B is higher than System A.

The capital cost share of each sub-section for systems A and B is illustrated in Figure 16. As it can be seen, the solar system has a major contribution to total capital cost investment. Its share is 84% and 83% for Systems A and B respectively.

![Figure 15: The efficiency of the bottom cycles as the total aperture area of the collector increases](image)

![Table 6: The economic results of systems A and B](image)
B, respectively. Also, the Kalina cycle and ORC achieve the second rank with 9% and 10% for Systems A and B, respectively.

5 | CONCLUSION

In this study, two cascade thermodynamic cycles have been compared, Systems A and B. System A is consisted of an SRC as the top cycle and an ORC as the bottom cycle. System B is consisted of the same SRC as the top cycle and a Kalina cycle as the bottom cycle. Moreover, both of the systems have been powered by parabolic trough collectors, and the solar radiation has been considered to be constant and equal to 750 W/m². The thermodynamic simulation of these systems has been conducted in Aspen Hysys 11.0. Furthermore, the effects of five different parameters have been examined on the systems’ performances to determine which system is more sensitive to the desired parameters. The top cycle condensing pressure, bottom cycles mass flow rate, bottom cycles turbine inlet and outlet pressures, and total aperture area of the collectors are the desired parameters. Some of the essential results are summarized as follows:

- It was determined that the most effective parameters on the overall efficiency of the systems are the condensing pressure of the top cycle and the outlet pressure of the bottom cycle turbine.
- The results showed that between the chosen parameters, the outlet pressure of the bottom cycle turbine has the highest effect on the efficiency of the bottom cycles.
- By increasing the top cycle’s condensing pressure, the ORC’s efficiency increases, but the Kalina cycle efficiency reduces. Also, the ORC is more sensitive to the change of the top cycle condensing pressure.
- As the mass flow rate of the bottom cycles increase, the efficiency of the Kalina cycle increases and also leads to an increase in the efficiency of System B. On the other hand, the efficiency of the ORC reduces, which results in the reduction of efficiency in system A.
- According to the results, ORC is considerably more sensitive to the change in the mass flow rate compared to the Kalina cycle.
- The results showed that the Kalina cycle is more sensitive to the change of the turbine inlet pressure as compared to the ORC. On the other hand, the ORC is more sensitive to the turbine outlet pressure.
- The enlargement of the total aperture area of the PTCs leads to the increase of the heat duty of E101 (the heat exchanger which combines top and bottom cycles). It results in boosting the efficiency of the ORC and reducing the efficiency of the Kalina cycle. Also, the ORC is more sensitive to the increase of the total aperture area.
- The economic evaluation showed that the LCOE for both Systems (A and B) are equal at 0.011 ($/kWh), and the payback period for both systems is 2 years.

NOMENCLATURE

SYMBOLS

- \( A \)  aperture area (m²)
- \( C_p \)  specific heat (kJ/kg·K)
- \( C_c \)  capital cost ($)
- \( C_o \)  operating and maintenance cost ($)
- \( C_{sv} \)  salvage value ($)  
- \( C_e \)  electrical income ($)
- \( C_v \)  calculated value
- \( D \)  diameter (m)
- \( \bar{E}rr \)  mean absolute percentage error
- \( F_1 \)  factor of collector efficiency
- \( F_R \)  heat removal factor
- \( G_b \)  solar beam radiation (kW/m²)
- \( h \)  heat transfer coefficient (kW/m²·K) and enthalpy (kJ/kg)
- \( k \)  conductivity (kW/m·K)
- \( L \)  length (m)
- \( m \)  mass flow rate (kg/s)
\( n \) system lifetime (years)
\( \text{NUS} \) Nusselt number
\( Q_u \) useful energy absorbed by PTC (kW)
\( r \) discount rate
\( R_v \) reference value
\( S \) heat absorbed by the receiver (kW)
\( T \) temperature (K)
\( U_L \) heat loss coefficient between surroundings and receiver (kW/K)
\( U_o \) overall heat transfer coefficient (kW/K)
\( W \) collector width (m)
\( \dot{W} \) power (kW)

GREEK SYMBOLS
\( \alpha \) absorbance of the receiver
\( \gamma \) intercept factor
\( \varepsilon_{cv} \) cover glass emittance
\( \eta_r \) receiver efficiency
\( \rho_c \) mirror reflectance
\( \sigma \) Stefan–Boltzmann constant (kW/m² K⁴)
\( \tau \) transmittance of the glass cover

SUBSCRIPT
\( \text{amb} \) ambient
\( \text{ap} \) aperture area
\( \text{av} \) average
\( c \) cover glass
\( \text{ca} \) between cover and ambient
\( \text{co} \) cover
\( \text{cr} \) between cover and receiver
\( i \) inlet
\( \text{kp} \) Kalina cycle pump
\( \text{kt} \) Kalina cycle turbine
\( o \) outlet
\( \text{op} \) ORC pump
\( \text{orc} \) organic Rankine cycle
\( \text{ot} \) ORC turbine
\( r \) radiation and receiver
\( \text{solp} \) solar field pump
\( \text{sp} \) Rankine cycle pump
\( \text{src} \) steam Rankine cycle
\( \text{st} \) Rankine cycle turbine

REFERENCES
1. Mahmoud M, Ramadan M, Olabi AG, Pullen K, Naher S. A review of mechanical energy storage systems combined with wind and solar applications. *Energy Convers Manage.* 2020; 210:210.
2. Kulkarni VV, Bhalla V, Garg K, Tyagi H. Hybrid nanoparticles-laden fluid based spiral solar collector: a proof-of-concept experimental study. *Renew. Energy.* 2021;179: 1360-1369.
3. Pourmoghadam P, Mehrpooya M. Dynamic modeling and analysis of transient behavior of an integrated parabolic solar dish collector and thermochemical energy storage power plant. *J Energy Storage.* 2021;42:103121.
4. Zhang S, Liu M, Zhao Y, Liu J, Yan J. Dynamic simulation and performance analysis of a parabolic trough concentrated solar power plant using molten salt during the start-up process. *Renew Energy.* 2021;179:1458-1471.
5. Aramesh M, Pourfayaz F, Kasaeian A. Transient heat extraction modeling method for a rectangular type salt gradient solar pond. *Energy Convers Manage.* 2017;132: 316-326.
6. Calise F, Cappiello FL, Dentice d’Accadia M, Vicidomini M. Thermo-economic optimization of a novel hybrid renewable trigeneration plant. *Renew Energy.* 2021;175: 532-549.
7. Aramesh M, Pourfayaz F, Kasaeian A. Numerical investigation of the nanofluid effects on the heat extraction process of solar ponds in the transient step. *Sol Energy.* 2017;157: 869-879.
8. Gholami A, Pourfayaz F, Saifoddin A. Techno-economic assessment and sensitivity analysis of biodiesel production intensified through hydrodynamic cavitation. *Energy Sci Eng.* 2021;9(11):1997-2018.
9. Aramesh M, Pourfayaz F, Haghir M, Kasaeian A, Ahmadi MH. Investigating the effect of using nanofluids on the performance of a double-effect absorption refrigeration cycle combined with a solar collector. *Proc Inst Mech Eng A J Power Energy.* 2019;234(7): 981-993.
10. Kasaeian A, Daneshzarian R, Pourfayaz F. Comparative study of different nanofluids applied in a trough collector with glass-glass absorber tube. *J Mol Liq.* 2017;234: 324-323.
11. Kaoed A, Abubakr M, Al-Oran O, Hassan MA. Performance analysis and particle swarm optimization of molten salt-based nanofluids in parabolic trough concentrators. *Renew Energy.* 2021;177:1045-1062.
12. Elbeh MB, Sleiti AK. Analysis and optimization of concentrated solar power plant for application in arid climate. *Energy Sci Eng.* 2021;9(6):784-797.
13. Rafiei A, Loni R, Ahmadi MH, et al. Sensitivity analysis of a parabolic trough concentrator with linear V-shape cavity. *Energy Sci Eng.* 2020;8(10):3544-3560.
14. Pourmoghadam P, Jafari Mosleh H, Karami M, Dynamic simulation of a solar ejector-based trigeneration system using TRNSYS-EES co-simulator. *Energy Sci Eng.* 2022;10(3): 707-725. doi:10.1002/ese3.1055
15. Pourmoghadam P, Farighi M, Pourfayaz F, Kasaeian A. Annual transient analysis of energetic, exergetic, and economic performances of solar cascade organic Rankine cycles

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integrated with PCM-based thermal energy storage systems. *Case Stud Therm Eng.* 2021;28:101388.

16. Feng H, Chen W, Chen L, Tang W. Power and efficiency optimizations of an irreversible regenerative organic Rankine cycle. *Energy Convers Manage.* 2020;220:113079.

17. Wu Z, Feng H, Chen L, Tang W, Shi J, Ge Y. Constructual thermodynamic optimization for ocean thermal energy conversion system with dual-pressure organic Rankine cycle. *Energy Convers Manage.* 2020;210:112727.

18. Chen W, Feng H, Chen L, Xia S. Optimal performance characteristics of subcritical simple irreversible organic Rankine cycle. *J Therm Sci.* 2018;27(6):555-562.

19. Zhang X, He M, Zhang Y. A review of research on the Kalina cycle. *Renew Sustain Energy Rev.* 2012;16(7):5309-5318.

20. Ghorbani B, Khatami Jouybari A. Development of a new CO2 liquefaction system using the Linde–Hampson process, Kalina power generation unit, and flat plate solar collectors with Al2O3/H2O nanofluid. *Energy Sci Eng.* 2021. doi:10.1002/esee.1034.

21. Ashouri M, Khoshkar Vandani AM, Mehrpooya M, Ahmadi MH, Abdollahpour A. Techno-economic assessment of a Kalina cycle driven by a parabolic Trough solar collector. *Energy Convers Manage.* 2015;105:1328-1339.

22. Boyaghchi FA, Sabaghian M. Multi objective optimisation of a Kalina power cycle integrated with parabolic trough solar collectors based on exergy and exergoeconomic concept. *Int J Energy Technol Policy.* 2016;12(2):154-180.

23. Ghorbani B, Mehrpooya M, sadeghzadeh M. Developing a trigeneration system of power, heating, and freshwater (for an industrial town) by using solar flat plate collectors, multi-stage desalination unit, and Kalina power generation cycle. *Energy Convers Manage.* 2018;165:113-126.

24. Ebrahimi A, Ghorbani B, skandarzadeh F, ziabasharhagh M. Introducing a novel liquid air cryogenic energy storage system using phase change material, solar parabolic trough collectors, and Kalina power cycle (process integration, pinch, and exergy analyses). *Energy Convers Manage.* 2021;228:228.

25. Ghaebi H, Rostamzadeh H. Performance comparison of two new cogeneration systems for freshwater and power production based on organic Rankine and Kalina cycles driven by salinity-gradient solar pond. *Renew Energy.* 2020;156:748-767.

26. Feng H, Qin W, Chen L, Cai C, Ge Y, Xia S. Power output, thermal efficiency and exergy-based ecological performance optimizations of an irreversible KCS-34 coupled to variable temperature heat reservoirs. *Energy Convers Manage.* 2020;205:112424.

27. Delgado-Torres AM, Garcia-Rodriguez L. Double cascade organic Rankine cycle for solar-driven reverse osmosis desalination. *Desalination.* 2007;216(1-3):306-313.

28. Al-Sulaiman FA. Energy and sizing analyses of parabolic trough solar collector integrated with steam and binary vapor cycles. *Energy.* 2013;58:561-570.

29. Li Y, Yuan J, Yang Y. Performance analysis of a novel cascade integrated solar combined cycle system. *Energy Proc.* 2015;75:540-546.

30. Li J, Li P, Pei G, Alvi JZ, Ji J. Analysis of a novel solar electricity generation system using cascade Rankine cycle and steam screw expander. *Appl Energy.* 2016;165:627-638.

31. Li P, Li J, Gao G, et al. Modeling and optimization of solar-powered cascade Rankine cycle system with respect to the characteristics of steam screw expander. *Renew Energy.* 2017;112:398-412.

32. Habibi H, Zoghi M, Chitsaz A, Shamsaee M. Thermo-economic performance evaluation and multi-objective optimization of a screw expander-based cascade Rankine cycle integrated with parabolic trough solar collector. *Appl Therm Eng.* 2020;180:180.

33. Yang H, Li J, Wang Q, et al. Performance investigation of solar tower system using cascade supercritical carbon dioxide Brayton-steam Rankine cycle. *Energy Convers Manage.* 2020;225:225.

34. Wu Z, Chen L, Feng H, Ge Y. Constructual thermodynamic optimization for a novel Kalina-organic Rankine combined cycle to utilize waste heat. *Energy Rep.* 2021;7:6095-6106.

35. Ozahi E, Abusoglu A, Tozlu A. A comparative thermo-economic analysis and optimization of two different combined cycles by utilizing waste heat source of an MSWW. *Energy Convers Manage.* 2021;228:228.

36. Ambriz-Díaz VM, Rubío-Mayá C, Chávez O, Ruiz-Casanova E, Pastor-Martínez E. Thermodynamic performance and economic feasibility of Kalina, Goswami and organic Rankine cycles coupled to a polygeneration plant using geothermal energy of low-grade temperature. *Energy Convers Manage.* 2021;243:243.

37. Morawski AP, deAraújo LR, Schiaffino MS, et al. On the suitable superstructure thermoeconomic optimization of a waste heat recovery system for a Brazilian diesel engine power plant. *Energy Convers Manage.* 2021;234:234.

38. Ogriiseck S. Integration of Kalina cycle in a combined heat and power plant, a case study. *Appl Therm Eng.* 2009;29(14):2843-2848.

39. Kalogirou SA. *Solar Energy Engineering: Processes and Systems.* Academic Press; 2013.

40. Duffie JA, Beckman WA. *Solar Engineering of Thermal Processes.* John Wiley & Sons; 2013.

41. Desai NB, Bandyopadhyay S. Thermo-economic comparisons between solar steam Rankine and organic Rankine cycles. *Appl Therm Eng.* 2016;105:862-875.

42. Dudley MA, Kolb GJ, Mahoney AR, et al. Test results: SEGs LS-2 solar collector (No. SAND94-1884). Sandia National Lab.

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