Abstract
In this study, a parametric analysis was performed of a supercritical organic Rankine cycle driven by solar parabolic trough collectors (PTCs) coupled with a vapour-compression refrigeration cycle simultaneously for cooling and power production. Thermal efficiency, exergy efficiency, exergy destruction and the coefficient of performance of the cogeneration system were considered to be performance parameters. A computer program was developed in engineering equation-solver software for analysis. Influences of the PTC design parameters (solar irradiation, solar-beam incidence angle and velocity of the heat-transfer fluid in the absorber tube), turbine inlet pressure, condenser and evaporator temperature on system performance were discussed. Furthermore, the performance of the cogeneration system was also compared with and without PTCs. It was concluded that it was necessary to design the PTCs carefully in order to achieve better cogeneration performance. The highest values of exergy efficiency, thermal efficiency and exergy destruction of the cogeneration system were 92.9%, 51.13% and 1437 kW, respectively, at 0.95 kW/m² of solar irradiation based on working fluid R227ea, but the highest coefficient of performance was found to be 2.278 on the basis of working fluid R134a. It was also obtained from the results that PTCs accounted for 76.32% of the total exergy destruction of the overall system and the cogeneration system performed well without considering solar performance.
Graphical Abstract

Performance comparison of solar driven supercritical organic Rankine cycle coupled with vapor compression refrigeration cycle

Key trade off:
- Thermal and exergy efficiency
- COP of system
- Working fluid selection

Results:
- Highest exergy efficiency 34.21%
- Highest thermal efficiency 68.58%
- COP of system 2.27
- Best working fluid R227ea

Introduction

Energy consumption used for cooling and power generation has increased drastically in recent years [1]. The energy consumed in air-conditioning and cooling equipment is large [2]. The vapour-compression refrigeration system is a widely used cooling system in which the compressor is the highest energy-consuming component because this energy is used to increase the coolant pressure. However, more energy use in the cooling system increases carbon-dioxide emissions, leading to higher global temperature and greenhouse effects. Therefore, renewable-energy technology is now used for refrigeration purposes. Such renewable energy sources are geothermal, solar, biomass, etc. [3]. The justification for using renewable energy for cooling is to reduce emissions of carbon dioxide and power consumption. Moreover, the cogeneration system of the organic Rankine cycle and vapour-compression cycle (ORC–VCC) is a heat-driven power and refrigeration system. The heat-driven cooling system is a clean technology and free from pollution. Solar energy is one of the most promising renewable heat sources because of its low costs, noise-free operation and abundance in nature [4]. Solar energy is the most suitable for cooling, heating and power generating, among other renewable energy sources. Solar collectors are being used to harvest solar energy where heat-transfer fluid (HTF) circulates for absorbing the heat of the Sun, which is used to drive the different types of thermodynamic cycles [5].

Nowadays, solar parabolic trough collectors (PTCs) are being widely used as heat sources to run combined as well as simple cycle electricity-generation systems. In this direction, a few studies were conducted using PTCs for trigeneration and cogeneration applications, e.g. Al-Sulaiman et al. [6] carried out a study on the PTC-driven ORC for utilizing waste heat for a cogeneration process. They found that during trigeneration, the energy efficiency was increased from 15% to 94%. Al-Sulaiman [7] carried out a thermodynamic analysis of the PTC-integrated steam Rankine cycle with ORC and R134a as the best working fluid among the other selected working fluids because it provided the highest electrical efficiency of 26%. The exergy and energy analysis of the PTC-integrated supercritical organic Rankine cycle (SORC) system was examined by Singh and Mishra [8]. Exergetic metrics including the fuel-depletion ratio, improvement potential and irreversibility ratio were found to be 0.579, 11 859 kW and 0.9296, respectively. Among the other working fluids tested, R600a was the best. Singh and Mishra [9] carried out a study on the PTC-driven supercritical carbon-dioxide cycle (sCO₂–ORC). They discovered that solar irradiation increased the thermal and exergy efficiency of the combined cycle. R407c was chosen as the best working fluid, with a combined system thermal
and exergy efficiency of 43.49% and 78.07%, respectively, at 0.95 kW/m². A PTC-driven partial heating scCO₂ cycle combined with ORC was studied by Khan and Mishra [10]. They discovered that solar irradiation enhanced the thermal and exergy efficiency, while the solar incidence angle reduced performance. Al-Zahrani and Dincer [11] performed exergy and energy analysis of the PTC-driven scCO₂ cycle to convert solar heat into power. They considered the parameters of the PTC as design parameters such as solar irradiation, receiver emittance and solar-beam incidence angle. The exergy and energy efficiency of the PTC were observed as 66.35% and 38.51%, respectively.

The SORC is better technology for the conversion of waste heat to power. For example, Song et al. [12] analysed the SORC to recover waste heat at low temperature. They concluded that the SORC was an appropriate method for industrial waste-heat recovery and the R152a was observed as the best-performing working fluid and produced the maximum net power and thermal efficiency. Kalra et al. [13] demonstrated that the operating fluid pressure was above its critical pressure in the SORC and found that the SORC had various benefits over its subcritical ORC because the heat-source cooling curve in the SORC matched the working-fluid heating curve. A parametric analysis of the SORC system for the medium-temperature geothermal heat source was undertaken by Moloney et al. [14]. They came to the conclusion that the SORC was more efficient than the ORC for low-temperature heat sources. Saadon and Islam [15] performed a thermal analysis of the SORC and discovered that it had a higher thermal efficiency and power output than the conventional ORC with a preheater.

Also several studies were conducted on the combined ORC–VCC system in recent years. Pektezel and Acar [16] analysed the combined ORC and single and double evaporator VCC systems for exergy and energy. They discovered that R600a was the most suitable fluid for the combined system. They also discovered that, with a single evaporator, the coefficient of performance (COP) and exergy efficiency of the combined cycle were higher than with a dual evaporator. The ORC-integrated VCC system was subjected to an exergy and energy study by Saleh [17]. He came to the conclusion that R602 was a good working fluid for system performance and environmental concerns. Finally, he concluded that the highest COP, exergy efficiency and pressure ratio of the turbine of the system and the corresponding total mass flow rate of the working fluid using R602 were 0.99, 53.8%, 12.2 and 0.005 kg/s-kW, respectively, at 25°C condenser temperature and maintaining other parameters as constant. Javanshir et al. [18] performed thermal and exergoeconomic analyses on the regenerative ORC–VCC system. Their results indicated that among other selected working fluids, R134a showed the lowest exergy and thermal efficiencies while R143a and R22 showed the highest exergy and energy efficiency. They also observed that the total unit cost of the product was found to be $60.7/GJ. The maximum exergy destruction was found in the boiler followed by the turbine. Hu et al. [19] investigated the performance of an ORC–VCC system operated by solar power. They concluded that the ice production and the cooling power depended on the condensation and generation temperature. R245fa was selected as an appropriate working fluid for this purpose. Using R245fa, cooling power and ice production per unit square collector area were 126.44 W/m² and 7.61 kg/m²·day, respectively. Moles et al. [20] proposed a model of the low-temperature heat-activated combined ORC–VCC system. They concluded that the computed thermal and electrical COP of the ORC–VCC system varied between 0.30–1.10 and 15–110, respectively. Additionally, R1234ze(E) was considered an acceptable working fluid for enhanced efficiencies. Li et al. [21] performed a working-fluid selection study for the combined ORC–VCC system. They concluded that butane was chosen as the better working fluid for the ORC–VCC system, with temperature variations ranging from 60°C to 90°C, −15°C to 15°C and 30°C to 55°C at boiler exit, evaporation and condensation temperatures varying, respectively. The COP of the complete system was obtained as 0.47 based on butane at a boiler exit temperature of 90°C.

It has been observed from the above-mentioned literature survey that several studies have been conducted on the ORC–VCC cogeneration system. However, no study was conducted on the SORC–VCC cogeneration system driven by PTCs. The impacts of the PTC design parameters on the cogeneration system have also not been observed in previous research. This demonstrates the novelty of the current study. The main objective of the present study is to examine the exergetic and energetic performance of the cogeneration SORC–VCC system simultaneously for the cooling and power generation driven by PTCs. The system-performance parameters were considered to be exergy and thermal efficiency, the COP of the cogeneration system and exergy destruction. The effects on the system performance of different parameters such as solar irradiation, the solar-beam incidence angle and the HTF velocity in the absorber tube, condenser temperature and evaporator temperature, and turbine inlet pressure were examined. Finally, the performance of the cogeneration system was compared with and without PTCs.

1 System description

Fig. 1 shows the schematic diagram of the SORC–VCC system driven by the PTC. A PTC field, a regenerating SORC and a VCC are all part of the system. Because there is a lot of heat energy available at the turbine outlet, regeneration of the SORC is used in this study. With the help of a recuperator, this heat is utilized to warm the incoming steam to the heat exchanger. Also, the SORC is beneficial for maximum power production [13]. This system uses two fluids: a single HTF in the solar field and another fluid in both the SORC and VCC subsystems. However, the SORC and VCC subsystems have different mass flow rates. There is a common condenser for both subsystems. For controlling the mass flow rate to both of the subsystems, two flow regulators and a common condenser for both cycles are used. To control the mass flow rate to both cycles, two flow
regulators are used. The fluid flow directions are shown by the arrows. The main advantage of this combined SORC–VCC system is that it produces cooling when power is not needed or vice versa. The advantage of using a combined SORC–VCC system produces cooling when refrigeration is not needed. Therefore, this system is beneficial throughout the year [17]. A solar field with a few hundred PTCs is used. The type of collector and other input data are given in Table 1. The T-s diagram has been displayed in Fig. 2.

### Table 1: Input conditions for simulation of the proposed model

| Parabolic trough collector | LS-3 [22] |
|-----------------------------|-----------|
| Maximum exit temperature of HTF | 390°C [7] |
| Maximum exit pressure of HTF | 10 MPa |
| HTF mass flow rate | 0.575 (kg/s) |
| Atmospheric pressure | 1.013 bar |
| Atmospheric temperature | 25°C |
| Effectiveness heat exchanger | 0.95 [7] |
| SORC turbine’s isentropic efficiency | 0.87 [18] |
| Compressor’s isentropic efficiency | 0.8 [18] |
| SORC pump's isentropic efficiency | 0.8 [18] |
| SORC’s mass flow rate | 1.5 kg/s |
| VCC’s mass flow rate | 1 kg/s |
| Turbine inlet pressure | 50 bar |
| Outlet pressure to the compressor/turbine | 6.018 bar |
| Condenser temperature | 35°C [17] |
| Effectiveness of recuperator | 0.95 [7] |
| Evaporator temperature | 0°C [17] |

A schematic of the PTC-driven SORC–VCC cogeneration system is shown in Fig. 1. The T-s diagram has been displayed in Fig. 2.

### 2 Thermodynamic analysis

The mathematical modelling of the SORC–VCC system has been conducted in this section. Modelling of the PTC system has already been conducted by Al-Sulaiman [7].

Modelling of the system was performed using the exergy and energy balance in each component. Results are calculated with help of the Engineering Equation Solver (EES) [23] software.

#### 2.1 Thermal modelling of the SORC–VCC system

While solving equations, the following assumptions were considered: (i) steady-state conditions are assumed for all thermodynamic processes; (ii) friction and pressure losses are negligible in each component and pipes; (iii) heat loss to the surroundings is negligible in each component except the condenser unit; (iv) the working-fluid conditions at the entry to the compressor and exit to the condenser in the VCC subsystem are saturated vapour and saturated liquid, respectively; (v) the PTCs have 50 collectors in series in a single row and there are a total of seven rows and the length of each collector is 12.27 m [7].

Modelling equations for the cogeneration (SORC–VCC) system are adapted from the previous studies [13, 18].

The total solar heat available in the SORC–VCC cogeneration system is determined as:

\[ \dot{Q}_{HX} = \dot{m}_{\text{SORC}} \cdot (h_4 - h_{10}) \]  

where \( \dot{m}_{\text{SORC}} \) is the working-fluid mass flow rate in the SORC subsystem.

The work done by the turbine is given as:

\[ \dot{W}_{\text{Turbine}} = \dot{m}_{\text{SORC}} \cdot (h_4 - h_{15}) \cdot \eta_{\text{Turbine}} \]  

The heat recovered in the in the recuperator is given by the heat balance equation:

\[ \dot{m}_{\text{SORC}} \cdot (h_6 - h_5) = \dot{m}_{\text{SORC}} \cdot (h_{10} - h_9) \]
If the same working fluid is circulating having the same mass flow rate on the hot and cold sides of the recuperator, then the effectiveness of the recuperator is given as:

\[ \varepsilon = \frac{(T_5 - T_6)}{(T_5 - T_9)} = \frac{\text{Actual heat transfer}}{\text{Maximum heat transfer}} \]  \hspace{1cm} (4)

The rejection of heat by the condenser unit is given as:

\[ \dot{Q}_{\text{cond}} = (\dot{m}_{\text{SORC}} + \dot{m}_{\text{VCC}}) \cdot (h_8 - h_7) \]  \hspace{1cm} (5)

where \( \dot{m}_{\text{VCC}} \) is the mass flow rate of the working fluid in the VCC subsystem.

Applying the energy-balance equation in the mixer:

\[ (\dot{m}_{\text{SORC}} + \dot{m}_{\text{VCC}}) \cdot h_7 = \dot{m}_{\text{SORC}} \cdot h_6 + \dot{m}_{\text{VCC}} \cdot h_{14} \]  \hspace{1cm} (6)

The compressor work is given by:

\[ \dot{W}_{\text{comp}} = \frac{\dot{m}_{\text{VCC}} \cdot (h_{14s} - h_{13})}{\eta_{\text{comp}}} \]  \hspace{1cm} (7)

where \( h_{14s} \) is the enthalpy at state 14 when the isentropic work is to be done.

Furthermore, when the system is brought to dead conditions (298.15 K temperature, 1.013 bar pressure), the maximum theoretical work obtained is termed as exergy. On the basis of the control volume, the exergy balance equation is determined as:

\[ \sum (1 - \frac{T_0}{T_{e}}) Q_{Q} - W_{c.v} - \sum (m_i Ex_i) - \sum (m_e Ex_e) - Ex_{\text{destruction}} = 0 \]  \hspace{1cm} (13)

where subscript Q denotes the thermal property numerically available at a given stage and \( Ex_{\text{destruction}} \) refers to the exergy-destruction rate. Furthermore, \( Ex \) signifies the specific system exergy. Without considering the chemical exergy, potential energy and kinetic energy, the physical exergy can then be assessed as [7, 25]:

\[ Ex_{\text{ph}} = (h - h_0) - T_0(s - s_0) \]  \hspace{1cm} (14)

The exergy inlet to the system and obtained from solar radiation by Petela’s formula [26] is described as:

\[ Ex_s = A_p \cdot G_0 \cdot \left[ 1 - \frac{1}{3} \left( \frac{T_0}{T_{su}} \right)^4 - \frac{4}{3} \left( \frac{T_0}{T_{su}} \right) \right] \]  \hspace{1cm} (15)

where \( T_{su} \) is the temperature of the Sun, i.e. 5800 K [7].

\( Ex_{\text{evap}} \) can be defined as [18]:

\[ Ex_{\text{evap}} = Q_{\text{evap}} \cdot \frac{T_0}{T_e} - 1 \]  \hspace{1cm} (16)

where \( T_e \) is the evaporator temperature.

The exergy efficiency of the SORC–VCC cogeneration system is defined as [18]:

\[ \eta_{\text{ex}} = \frac{\dot{W}_{\text{net}} + Ex_{\text{evap}}}{Ex_s} \]  \hspace{1cm} (17)

The exergy efficiency of the overall system (SORC–VCC–PTC) is shown as [24]:

\[ \eta_{\text{ex,overall}} = 1 - \frac{Ed_{\text{Total}}}{Ex_s} \]  \hspace{1cm} (18)
where $\text{Ed}_{\text{Total}}$ is the total exergy destruction of the overall system.

Another performance parameter to be defined is the COP of the SORC-VCC cogeneration system ($\text{COP}_S$). This can be defined as [17]:

$$\text{COP}_S = \eta_{\text{SORC}} \cdot \text{COP}_{\text{vcc}} = \frac{\dot{Q}_{\text{evap}}}{\dot{Q}_{\text{HX}} + \dot{W}_{\text{SORC pump}}} \tag{19}$$

where $\eta_{\text{SORC}}$ and $\text{COP}_{\text{vcc}}$ are the efficiency of the SORC and the COP of the vapour-compression subsystem, respectively.

### 2.2 Working-fluid selection

Working-fluid selection is challenging because it affects the performance and economic feasibility of the system. It is good if the critical temperatures of the working fluid are near the temperature of the heat source and the temperature of the waste [17]. The HTF selected in the current study is Therminol-VP1 in the absorber tube of the solar collector to absorb the heat of the Sun because this synthetic oil has excellent heat-transfer properties as well as good temperature stability compared with other HTFs such as molten salt and a maximum working temperature of 400°C [27]. Its thermophysical properties were discussed in the study of Mwesigye et al. [27]. The working fluids in the SORC system should be non-toxic, environmentally friendly, economical, able to use heat energy from the source of waste heat and also, at optimum pressure and temperature, the thermophysical and thermodynamic properties should be optimal [28]. Due to these reasons, fluids such as R227ea, R236fa, R245fa, R1234ze and R134a were selected. The thermophysical properties of the considered working fluids are shown in Table 2.

### 2.3 Verification of the model

The current model was validated using previous studies. The PTC as well as the cogeneration SORC–VCC system has been separately verified. Verification with the change in heat loss in the absorber tube with the temperature change above the ambient temperature and exit temperature of the PTC have been shown in Tables 3 and 4, respectively, at the same baseline situations. There is <1% deviation from the experimental results. It is a better agreement than in the previous theoretical study by Al-Sulaiman [7] and the experimental study by Dudley et al. [22]. Apart from this, the thermal efficiency of the SORC system has been validated by the study by Le et al. [29] under the same baseline conditions as given in Table 5. Table 6 shows the COP variation of

| Working fluids | Weights (g/mole) | Critical temperature (°C) | Critical pressure (MPa) | ODP | GWP |
|----------------|------------------|--------------------------|-------------------------|-----|-----|
| R227ea         | 170.03           | 101.8                    | 2.925                   | 0   | 3580|
| R236fa         | 152.04           | 124.92                   | 3.2                     | 0   | 680 |
| R245fa         | 134.05           | 154.1                    | 3.65                    | 0   | 1050|
| R1234ze        | 104.04           | 109.36                   | 3.635                   | 0   | <1  |
| R134a          | 102.03           | 101                      | 4.059                   | 0   | 1430|

| Temperature difference (°C) | Current model | Dudley et al. [22] | Al-Sulaiman [7] |
|-----------------------------|---------------|--------------------|-----------------|
| 100.6                       | 9.78          | 10.6               | 8.719           |
| 149.1                       | 16.24         | 19.3               | 19.3            |
| 196.7                       | 31.15         | 30.6               | 34.2            |
| 245.8                       | 53.48         | 45.4               | 53.0            |
| 293.3                       | 66.69         | 62.9               | 75.5            |

| Cases | $G_b$ (W/m²) | Discharge (m³/s) | $T_s$ (K) | $T_e$ (K) | $T_i$ (K) | Dudley et al. [22] | Current model | Error (%) |
|-------|--------------|------------------|-----------|-----------|-----------|---------------------|---------------|----------|
| 1     | 933.7        | 47.7             | 294.35    | 375.35    | 397.15    | 397.56              | 0.10          |
| 2     | 968.2        | 47.8             | 295.55    | 424.15    | 446.45    | 446.72              | 0.06          |
| 3     | 982.3        | 49.1             | 297.45    | 470.65    | 492.65    | 492.02              | 0.12          |
| 4     | 909.5        | 54.7             | 299.45    | 523.85    | 542.55    | 541.99              | 0.10          |
| 5     | 937.9        | 55.5             | 299.35    | 570.95    | 589.55    | 590.11              | 0.09          |
| 6     | 880.6        | 55.6             | 301.95    | 572.15    | 590.35    | 590.26              | 0.01          |
| 7     | 903.2        | 56.3             | 300.65    | 529.05    | 647.15    | 647.35              | 0.03          |
| 8     | 920.9        | 56.8             | 304.25    | 652.65    | 671.15    | 671.51              | 0.05          |
Table 5: Verification of the SORC system

| Baseline conditions | Literature | Thermal efficiency (%) |
|---------------------|------------|------------------------|
| R134a, heat-source temperature = 139°C | Le et al. [29] | 12, 11.99, –1 |

Table 6: COP variation of the VCC system with the evaporator average temperature

| Evaporator average temperature (°C) | Current model | Krishnan et al. [30] |
|-------------------------------------|---------------|----------------------|
| 13                                  | 2.85          | 2.99                 |
| 13.5                                | 3.22          | 3.12                 |
| 14                                  | 3.33          | 3.45                 |
| 14.5                                | 3.82          | 3.95                 |
| 15                                  | 3.98          | 4.02                 |
| 15.5                                | 4.10          | 4.21                 |
| 16                                  | 4.42          | 4.38                 |

the VCC system with the evaporator average temperature and validated by the experimental study by Krishnan et al. [30]. Table 7 shows that the thermal efficiency of the SORC–VCC cogeneration system matches that from the study by Javanshir et al. [18]. Therefore, the current model is ready for analysis.

3 Results and discussion

In the present study, a thermal performance analysis of the PTC-driven SORC–VCC cogeneration system was conducted. The effects of solar irradiation (G_s), solar-beam incidence angle, HTF velocity in the absorber tube, turbine inlet pressure (TIP), condenser temperature (T_c) and evaporator temperature (T_e) on thermal performance were examined. While investigating the effects of the parameters, other parameters were kept constant as listed in Table 1. A computer program was made using EES software to solve the modelling equations.

3.1 Performance evaluation with solar irradiation

Solar irradiation, also termed as the direct normal irradiation, is a primary parameter to be examined. The variation in solar irradiation was taken as 0.5–0.95 kW/m² as per the climate of Mumbai, India. Variation in solar irradiation has a large impact on the system performance ranging from 0.5 to 0.95 kW/m². The thermal performance of the SORC–VCC cogeneration system increases with solar irradiation due to the effective utilization of collector rows as shown in Fig. 3. Among the selected working fluids, R227ea gives the highest thermal efficiency that varies from 47.55% to 51.13% at 0.5–0.95 kW/m², respectively, because R227ea has the lowest critical pressure compared to other selected working fluids. Near the critical conditions, the working fluid performs better. The exergy efficiency of the cogeneration system also improved with the solar irradiation. The reason behind this is that, as the irradiation increases, the inlet exergy to the cogeneration system increases according to Equation (20) and simultaneously the exergy destruction decreases. R227ea has the highest exergy efficiency ranging from 86.38% to 92.9% at 0.5–0.95 kW/m², respectively.

The overall system (PTC-SORC–VCC) exergy and thermal efficiency were also examined. It is observed from Fig. 4 that the overall system thermal efficiency increases with the solar irradiation due to the same reason as above. R227ea and R134a, among the other selected working fluids, gave respectively the highest and lowest thermal efficiency. Using R227ea, the solar irradiation increases from 0.5 to 0.95 kW/m², which results in a nearly 58.88% increase in the thermal efficiency. The rise in solar irradiation results in an increase in the overall system exergy efficiency. As the values of the solar irradiation vary from 0.5 to 0.95 kW/m², the exergy efficiency based on R227ea improves by almost 58.90%, as illustrated in Fig. 4.

The variation in solar irradiation also has a significant effect on the COPs, as denoted in Fig. 5. Fig. 5 shows that the COPs is a function of the solar irradiation. As the solar irradiation increases, this leads to an increase in the COPs. Increased solar irradiation has no effect on the VCC subsystem, which means no effect on the COPVCC. However, an increase in solar irradiation leads to an increase in COPSORC [8]. The COPs increases with solar irradiation based on Equation (19). R134a has the highest COPs and R227ea the lowest among the considered working fluids. This means that if this system only works for cooling purposes, R134a is more effective than R227ea. Using the working fluid R134a, as the solar irradiation varied from 0.5 to 0.95 kW/m², the COPs increased by nearly 7.69%. It is also shown that the exergy destruction decreases with the solar irradiation. It has the reverse trend to that of the exergy efficiency, as explained above. R134a and R227ea gave the highest and lowest exergy destruction as shown in Fig. 5. R134a gave exergy-destruction decreases from 3966 to 2060 kW when the solar irradiation varied from 0.5 to 0.95 kW/m², respectively. Furthermore, without the solar SORC–VCC, the system is more thermodynamically efficient than with solar.

3.2 Performance evaluation with solar incidence angle

Fig. 6 shows that the thermal and exergy efficiency of the cogeneration system decreases with the increase in the
solar incidence angle. An increase in the incidence angle reduces the exergetic, energetic and optical efficiencies of the PTCs [11] and leads to system-performance reduction. The maximum and minimum thermal efficiency and exergy efficiency were found for R227ea and R134a, respectively. The thermal and exergy efficiency decreased by 51.02–50.57% and 92.70–91.88% with increases in the solar incidence angle from 3° to 30°, respectively, based on R227ea. Therefore, it is important to minimize the solar incidence angle to obtain better performance of the system. Also, the COP $s$ decreased with the solar incidence angle as shown in Equation (19), the COP decreases with the solar incidence angle. The highest and lowest COP $s$ were found with R134a and R227ea, respectively. The COP $s$ decreased from 2.6 to 1.3 as the incidence angle increased from 3° to 30°, respectively, whereas the exergy destruction increased with the solar incidence angle, as shown in Fig. 7. It has been discussed above that the exergy efficiency decreased with the solar incidence angle, which is why the exergy destruction showed a reverse trend. The exergy destruction for all other selected working fluids was found to be between these of the fluids R227ea and R134a. As the solar incidence angle increased from 3° to 30°, the exergy destruction increased from 3966 to 2060 kW on the basis of R134a, as seen in Fig. 7.

### Table 7: Verification of the cogeneration SORC-VCC system

| Baseline conditions                     | Literature       | Thermal efficiency (%) | Error (%) |
|-----------------------------------------|------------------|------------------------|-----------|
| R143a, TIP = 3600 kPa, heat-source temperature = 150°C | Javanshir et al. [18] | 27.21                  | 27.98     | 2.49     |

Fig. 3: Variation in the efficiency of the SORC–VCC cogeneration system with solar irradiation

Fig. 4: Variation in the efficiency of the overall system (SORC–VCC–PTC) with solar irradiation
3.3 Effects of HTF velocity in the absorber tube on system performance

Fig. 8 displays that the thermal and exergy efficiency of the SORC–VCC cogeneration system has increased with the HTF velocity in the absorber tube. The increase in the efficiency with the velocity is due to the increase in the velocity of the fluid. The Reynolds number is increased as a result of the increase in the convective heat-transfer
coefficient; therefore, much heat is carried by the HTF. This leads to an increase in efficiency. Among the operating fluids considered, R227ea and R134a were reported to have the greatest and smallest thermal and energy efficiency. Thermal and exergy efficiency increased from 47.55% to 51.13% and from 86.38% to 92.9% when the velocity increased from 0.01 to 0.1 m/s on the basis of R227ea. The COPs also increased with the velocity of the HTF. As the velocity increased, the efficiency of the SORC was enhanced. However, there was no effect on the COPvcc. Therefore, according to Equation (19), the COPs increased with the HTF velocity as seen in Fig. 9. The highest value of the COPs was found with R134a followed by R245fa, R236fa, R1234ze and R227ea. The COPs also increased with the velocity of the HTF. As the velocity increased, the efficiency of the SORC was enhanced. However, there was no effect on the COPvcc. Therefore, according to Equation (19), the COPs increased with the HTF velocity as seen in Fig. 9. The highest value of the COPs was found with R134a followed by R245fa, R236fa, R1234ze and R227ea. The exergy destruction of the SORC–VCC cogeneration system was also investigated with the HTF velocity in the absorber tube. The exergy destruction decreased with the HTF velocity. The highest exergy destruction was identified with R134a followed by R1234ze, R245fa, R236fa and R227ea with values of 3966, 3673, 3515, 3241 and 2757 kW, respectively, at an HTF velocity of 0.01 m/s, as shown in Fig. 9.

### 3.4 Effects of the TIP on system performance

The system performance also depends on the TIP. As shown in Fig. 10, the efficiency of the cogeneration system increased slightly with the TIP. The TIP variation had no impact on the VCC subsystem performance. The VCC subsystem does not contribute to the efficiency of the SORC–VCC cogeneration system. Therefore, the efficiency of the cogeneration system depends only on the performance of the SORC subsystem. It is due only to the $\eta_{\text{SORC}}$ because the $\eta_{\text{SORC}}$ increases with the TIP [8]. It is observed that R227ea and R134a provided the maximum and minimum thermal efficiency of the cogeneration system. In the case of the R227ea, the TIP increase from 40 to 55 bar improved the cogeneration system thermal efficiency by almost 0.85%. From Fig. 10, it can be seen that as the TIP increased, the exergy efficiency of the cogeneration system also increased. The destruction of exergy decreases with the TIP led to exergy-efficiency improvement [8]. Using R227ea, the increase in the TIP from 40 to 55 bar improved the exergy efficiency from 91.82% to 92.60%.

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**Fig. 8:** Variation in the efficiency of the SORC–VCC cogeneration system with HTF velocity

**Fig. 9:** Variation in the COPs with the HTF velocity in the absorber tube
In addition to the cogeneration system performance, the performance of the overall system was also investigated and the overall system (SORC–VCC–PTC) thermal efficiency was found to increase with the TIP, as seen in Fig. 11. The exergy efficiency of the overall system also depended on the TIP. The exergy efficiency of the overall system improved with the TIP. The TIP did not affect the PTC and VCC subsystems. As a result, the exergy efficiency was only enhanced by the SORC subsystem. In the case of the working fluid R227ea, the TIP increases from 40 to 55 bar improved the exergy efficiency by 5.5%, as shown in the right axis of Fig. 11.

The COPs is also a function of the TIP. The COPs slightly increased with the TIP. The reason behind this is that the variation in the TIP had no effect on the COPvc, although $\eta_{\text{SORC}}$ increased with the TIP [8, 18], from Equation (19), the COPs increased. R134a and R227ea show the highest and lowest COPs, respectively. If the TIP increased from 40 to 55 bars, the COPs increased by nearly 1.43%, as shown in Fig. 12. Fig. 12 also shows that the exergy destruction decreased with the TIP. As already discussed, the exergy efficiency increased with the TIP and, consequently, the exergy destruction decreased with the TIP. The highest and lowest exergy were obtained with the working fluids R134a and R227ea, respectively, although the exergy destruction for fluids R236fa, R245fa and R1234ze was found between these two. Further, comparing Fig. 10 with Fig. 11, the performance of the PTCs combined system performance was reduced. It does not mean that the solar system is worse overall, but only from the efficiency point of view.

### 3.5 Performance evaluation with condenser temperature

The condenser temperature also has an impact on both subsystems. As the condenser temperature increased, this led to a slight decrease in the exergy and thermal efficiency of the cogeneration system (SORC–VCC), as shown in Fig. 13. Improvement in the condenser temperature reduced the enthalpy drop in the turbine leading to the lower output work. Therefore, the thermal efficiency of the SORC and COPvc system was reduced. Consequently, the thermal efficienc...
The performance of the cogeneration system was reduced. The exergy efficiency of the cogeneration system also decreased with the condenser temperature because, according to Equation (12), the exergy efficiency is directly related to the thermal efficiency, as revealed in Fig. 13. Furthermore, the thermal efficiency of the overall system also depends on the condenser temperature. As the temperature of the condenser increased from 303 K to 328 K, the thermal efficiency decreased by almost 14.07%, as shown in Fig. 14. This was due only to the performance of the SORC–VCC system, not to that of the PTC system. The effect of the condenser temperature on the exergy efficiency of the overall system (SORC–VCC–PTC) was discussed. The exergy efficiency also decreased with the condenser temperature. It is already known that thermal efficiency decreases with the temperature of the condenser. Hence the exergy efficiency also decreased with the condenser temperature by Equation (12). The efficiencies of the other fluids are between those of the R227ea and R134a fluids, as shown in Fig. 14.

Finally, in this section, the impact of the condenser temperature variation on the system COPs was also discussed, as shown in Fig. 15. In the case of R134a, as the condenser temperature varied from 303 K to 328 K, the COPs decreased by 49.8%. It is already known that the ηSORC decreases with the condenser temperature. However, the temperature, pressure and enthalpy increase with the condenser temperature at the outlet of the compressor. This results in a reduction in the COPvcc. As a result, the COPs decreases according to Equation (19). R134a and R227ea show the highest and lowest COPs, respectively. It can also be seen in Fig. 15 that if this system only works for the purpose of cooling, then R134a is selected as the appropriate working fluid among other selected working fluids. The exergy destruction increases with the condenser temperature. The exergy efficiency slightly increases with condenser temperature, therefore the exergy decreases with the condenser temperature. The highest exergy destruction was found with R134a, followed by R1234ze, R245fa, R236fa and R227ea of 2347, 2176, 2061, 1932 and 1644 kW, respectively, as displayed in right axis of Fig. 15.
3.6 Performance evaluation with evaporator temperature

Furthermore, the effect of the evaporator temperature was also examined. The cooling capacity is directly affected by the evaporator temperature. During examining the evaporator-temperature effect on the system performance, all other assumed parameters are kept constant, as listed in Table 1 such as the condenser temperature and pressure, evaporator pressure, etc. Fig. 16 reveals that the efficiency is not significantly affected by the evaporator temperature because there was no impact of the evaporator temperature on the SORC subsystem performance. The working fluids R227ea and R134a gave the highest and lowest efficiency, respectively. Further to Fig. 16, it is noticed that the evaporator temperature also did not significantly affect the exergy efficiency of the cogeneration system because the exergy efficiency has a direct relation with the thermal efficiency [31]. However, the COPs was affected by the evaporator temperature significantly and it increased with the evaporator temperature. The reason behind this is that the COPvcc increases with the evaporator temperature [17]. Also, the saturation pressure of the evaporator improves with the improvement of the evaporator temperature for all working fluids leading to a decline in the compressor work with constant condenser temperature. Conversely, the refrigeration effect improves with the evaporator temperature. Both impacts enhance the COPvcc. However, simultaneously, the ηSORC is not affected by the evaporator temperature. As per Equation (19), this resulted in improvement in the COPs, as shown in Fig. 17. Also, R134a and R227ea showed the highest and lowest values, respectively. Regarding the exergy destruction of the cogeneration cycle, the exergy destruction also was not affected much by the evaporator temperature like the efficiencies, as discussed above. At a constant evaporator temperature, the highest and lowest exergy destruction was for R134a and R227ea, respectively, whereas the exergy destruction for the R245fa was significantly affected due to different thermophysical properties of R245fa, as shown in the right axis of Fig. 17.
3.7 Exergy destruction in each component

In addition to the above parameters, the exergy destruction for each component and for each working fluid has been also evaluated, as revealed in Table 8. On the basis of R134a, a major part of the energy destruction rate was found in the PTCs followed by the turbine and heat exchanger with values of 8067, 1787 and 312.8 kW, respectively. It can be seen that PTCs alone accounted for 76.32% of the total exergy destruction of the overall system. The total exergy input into the cogeneration system was calculated at ~20 250 kW. It is therefore shown that 39.83% of the total inlet exergy was destroyed in the solar field only. This is due to the high exergy losses in the PTCs as well as the high outlet and inlet temperature difference of the HTF stream of the collector. Therefore, the PTC is the worst subsystem in terms of exergetic performance in the current model. Therefore, it is necessary to design the PTC system carefully. It can be also observed from Table 8 that the expansion valve was responsible for the lowest exergy-destruction rate among the other components and that it was ~3.695 kW only. Among the other considered working fluids, R227ea showed the lowest exergy-destruction rate. R227ea is superior because this has a more supercritical operating range due to its low critical values compared with other selected fluids, as listed in Table 2. The optimal parameters are summarized in Table 9.

![Fig. 16: Variation in the efficiency of the SORC–VCC cogeneration system with the evaporator temperature](https://academic.oup.com/ce/article/5/3/476/6375889)

![Fig. 17: Variation in the COP, with the evaporator temperature](https://academic.oup.com/ce/article/5/3/476/6375889)

| Components          | R227ea | R236fa | R245fa | R1234ze | R134a |
|---------------------|--------|--------|--------|---------|-------|
| Solar field         | 6894   | 7265   | 7253   | 8006    | 8067  |
| HX                  | 218.3  | 309.1  | 311.8  | 288.5   | 312.8 |
| Turbine             | 1092   | 1216   | 1361   | 1596    | 1787  |
| Recuperator         | 46.3   | 52.62  | 26.99  | 24.21   | 17.24 |
| Condenser           | 23.03  | 64.64  | 130.5  | 15.51   | 22.01 |
| Mixer               | 181.7  | 191.9  | 172.5  | 175.9   | 167.6 |
| Compressor          | 170.3  | 186.1  | 153.9  | 159.5   | 147.2 |
| EX valve            | 2.937  | 3.088  | 3.109  | 3.509   | 3.695 |
| Evaporator          | 15.36  | 21.41  | 7.783  | 24.9    | 27.54 |
| ORC pump            | 16.47  | 10.87  | 18.31  | 19.19   | 17.61 |

Table 8: Exergy-destruction rate (kW) for each component
4 Conclusions

In the present study, an exergy and energy analysis of the PTC-driven SORC–VCC cogeneration system was conducted. The following conclusions were made from the results:

- In Mumbai, India, the exergy efficiency, thermal efficiency and COP of the SORC–VCC cogeneration system increased with solar irradiation and HTF velocity in the absorber tube, but declined with the solar-beam incidence angle. As a result, in order to increase the performance of the cogeneration system, the PTCs must be carefully designed.

- The maximum values of exergy efficiency, thermal efficiency and COP of the SORC–VCC cogeneration system were 92.9%, 51.13% and 2.27, respectively, using R227ea and R134a at 0.95 kW/m² of solar irradiation. At 0.95 kW/m², the overall system exergy efficiency and thermal efficiency were 62.78% and 34.5%, respectively.

- The exergy efficiency, thermal efficiency and COP decreased with the condenser temperature and increased with the TIP. The COP also increased continuously with the evaporator temperature and reached a maximum value of 3.764 at 328 K for R134a.

- The exergy destruction decreased with the solar irradiation and HTF velocity in the absorber tube, although it increased with the solar-beam incidence angle.

- For the working fluid R134a, the maximum total exergy-destruction rate was found to be ~8067 kW in a solar field. It contributed 76.32% of the total exergy-destruction rate of the system. It was also discovered that the solar field alone destroyed 39.83% of the input exergy.

- The overall system performance was reduced by considering the performance of the PTCs from an efficiency point of view. If the performance of the PTCs was not considered, then the cogeneration system performed well with solar energy.

- It was not necessarily true that a working fluid that was better for power generation was also better for cooling. Among the other working fluids considered, R227ea and R134a fared better for power generation and cooling, respectively.

- In future research, an exergoeconomic and thermoeconomic analysis should be performed for the current model. Pragmatic applications of the current model will be also possible in those areas for simultaneous cooling and power generation where conventional heat sources are not present.

| Parameters | Values |
|------------|--------|
| Optimum thermal efficiency | 51.13% |
| Optimum exergy efficiency | 92.9% |
| Optimum COP of system | 2.27 |
| Best working fluid for power generation | R227ea |
| Best working fluid for cooling | R134a |

| Nomenclature |
|--------------|
| Q | Heat transfer (kW) |
| s | Specific entropy (kJ/kg-K) |
| Ed | Exergy-destruction rate (kW) |
| Gs | Solar irradiation (kW/m²) |
| Ap | Aperture area (m²) |
| To | Atmospheric temperatures |
| T | Temperature (K) |
| W | Work (kW) |
| Ed | Exergy efficiency |
| ηth | Thermal efficiency |
| ηex | Exergy efficiency |
| ηHTF | Mass flow rate of HTF (kg/s) |
| Tab | Temperature of the Sun (K) |

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Conflict of interest statement

None declared.

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