Optimization and Analyzing of Subcritical Organic Rankine Cycle Using R1234ze(E) for Low and Medium Temperature Heat Source

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Abstract. A model of Subcritical Organic Rankine Cycle (SORC) driven by trans-1,3,3,3-tetrafluoroprop-1-ene, R1234ze(E) as a new-typed environmental friendly refrigerant working fluid, was developed and analyzed in the view of the 180-220°C wet fluegas combustion and burning from natural gas, as a low to medium temperature heat source without the outlet temperature limit to generate optimum power by a turbine. The net power output, total efficiency, fluegas outlet temperature, and exergy efficiency of current SORC system were studied based on variation of vapor generator outlet temperature and turbine outlet pressure as optimized parameters in terms of per mass flow rates of working fluid, heat source, and heat sink. Results show that the maximum net power output is 5.2675 MW and with increasing of mass enthalpy change and temperature in expansion process causes to increase of net power output, steadily. And the maximum net power output was 53.1% increased. Furthermore, by decreasing the turbine outlet pressure, the total efficiency of the current SORC system is increased to 0.08107 because of the increase of net power output. Likewise, the exergy efficiency of the SORC system increasing significantly, and received to the 0.328, because of increasing the enthalpy change and decreasing the entropy change of fluegas in the evaporator system.

Keywords: Subcritical Organic Rankine Cycle (SORC); R1234ze(E); fluegas; without the outlet temperature limit; optimized parameters.

1. Introduction
A number of technologies have potential to convert medium to low temperature heat source (<350°C) to power which employ different organic working fluids [1-4]. Organic Rankine cycle (ORC) has been proven to be one of the most promising and efficient technologies for converting medium to low grade heat source e.g. solar thermal energy, geothermal energy, biomass energy and industrial waste heat such as fluegas in to power [5-20], because ORC system is based on Rankine cycle and has a lot of advantages, such as uses low boiling point organic working fluids, high efficiency in each components, simplicity, flexibility, stability, safety, wide applicable heat source temperature range, and wide installed capacity range [21,22].
In an ORC system, the organic working fluid has crucial effect on power production and system performance as the main aims of this technology [23-25]. Common working fluids for ORC systems include the hydrochlorofluorocarbons (HCFCs), hydrofluorocarbons (HFCs), hydrocarbons (HCs) and siloxanes. HCFCs, HFCs and light HCs are typically applied for low-temperature heat sources (<200°C), such as geothermal energy, solar energy with simple collectors and several waste heat sources. The heavy HCs and siloxanes are typically applied for medium-temperature heat sources (200-350°C), such as biomass energy, solar energy with concentrated solar collectors, waste heat from engine exhaust and industrial fluegas [3,15,16,20,26-30]. According to two controversial issues such as ozone depletion potential (ODP) and global warming potential (GWP), the ability to destroy the ozone layer and increases the greenhouse effect, respectively; Therefore, HCFCs, HFCs, and HCs are being phased out consequently. Furthermore, several international environmental protection protocols and agreements such as Montreal Protocol (1989), Kyoto Protocol (2005), and Paris Agreement (2016), based on the above environmental harmful effects have been restricted to phased out of these refrigerant working fluids groups. In this case, finding and establishing a new type of working fluid that would be very safe and environmental friendly is needed necessarily [1,31-33].

R1234ze(E) with chemical name of trans-1,3,3,3-tetrafluoroprop-1-ene and with molecular formula of CF3CH=CH2 from hydrofluoroolefines (HFOs) is a new type of environmental friendly working fluid with low boiling point, ultra-low GWP (GWP<1), zero ODP, and A2L safety classification which is useful for commercial applications. Also because of containing a carbon-carbon double bond it becomes low atmospheric lifetime. Many scholars try to investigate the thermodynamic properties of R1234ze(E) to analysis thermodynamic performance and net power output production of ORC [1,21,34,35].

A number of scholars attempt to use R1234ze(E) as an environmental friendly working fluid in ORC systems. For example, Li et al [1] focused on subcritical and transcritical ORCs that driven by R1234ze(E) and using the 100–200°C heat source without the outlet temperature limit, their results showed that between R1234ze(E), R245fa, and R600a, the maximized system net power output is belong to R1234ze(E). For various heat source temperatures Li et al. [34] applied the dual pressure heat absorption ORC thermodynamic cycle that driven by R1234ze(E) in heat source temperature 150-200°C and the result reveals that with optimize the exergy efficiency for heat source temperatures of 150–200°C which indicates that the heat source fluid and working fluid can achieve an excellent temperature match. Le et al. [31] utilized a transcritical ORC driven by R1234ze(E) and using 150°C hot water as a heat source. Their results confirmed that, between R134a, R152a, R32, R1270, R290, R1234yf, R1234ze, and R744 as organic working fluids, the maximum net power generated and maximum system efficiency belong to R1234ze(E). Yang et al. [36] investigate on selection the most suitable working fluid between R600, R600a, R601a, R245fa, R1234yf and R1234ze as working fluids in subcritical ORC that use exhaust waste heat recovery of diesel engine with 200–370°C of heat source temperature. Their results indicated that R1234ze(E) selected as a significant thermodynamic performance in this subcritical ORC compare with other working fluids. Zhang et al. [37] operated a subcritical air-cooled ORC for utilization of low-temperature geothermal brine at 150°C and using R245fa and two low GWP working fluids R1234ze(Z) and R1234ze(E) to maximize system exergy efficiency. Their results revealed that, the highest system exergy efficiency and highest total system efficiency between these three organic refrigerant working fluids for 100 kg/s geothermal source is belong to R1234ze(E). Ge et al. [38] studied on an ORC system in a low temperature of Geothermal Water as a heat source and drive with R1234ze(E) as a working fluid. Their results showed that, in different evaporation temperature and working fluid mass velocity have direct effect on net power output and total efficiency of ORC system.

A large group of researchers try to use some specific heat sources such as solar thermal energy [39-41], geothermal energy [42-44], heat from biomass [45-47]. But just a few of them try to carry out industrial waste gases especially fluegas as a heat source [48,49]. In this case has some lack of in the ORC technology knowledge. These industrial waste gases have a high temperature (>150 °C) and also have harmful environmental compounds such as: CO2, N2, O2, and H2O which cause a number of environmental problems, such as global warming, climate change, acid rain, and air pollution [50]. Finally, the last advantage of this study is using the AspenPlus simulation software (V10), applying REFFPROP (V10) to calculate the thermophysical properties of R1234ze(E) as a working fluid, and utilizing the Simulink of Matlab (V2017a) as a simulation analysis software, can decrease operational
costs in industry at real level, and also decrease the number of errors at implementation of each section of the current study and increase the safety of this study by using these powerful Chemical Engineering simulation softwares.

This paper focuses on a subcritical ORC using R1234ze(E) driven by the (180-220°C) wet fluegas combustion and burn from natural gas as a low to medium temperature open type heat source without the outlet temperature limit. The optimal cycle type (subcritical), analyze and optimized the variables cycle parameters (vapor generator outlet temperature and turbine outlet pressure) and in the same line, investigate the effect of these variable cycle parameters on ORC system performance and ORC exergy performance base on specific mass flow rates of R1234ze(E) as a working fluid by focus on generating optimum net power output.

2. Methodology

2.1. Thermophysical properties of R1234ze(E)
Thermophysical properties of R1234ze(E) are listed in Table 1 [1,21,34,35,51]. The maximum value of entropy reaches in maximum value in inflection point (S_ip) of two-phase region on saturation vapor curve. Above this inflection point, the slope of the saturation vapor curve is negative and (dS/dT)_sat < 0, therefore, R1234ze(E) indicates a wet working fluid property. On the other hand, below this inflection point, the slope of the saturation vapor curve is positive and (dS/dT)_sat > 0, thus, R1234ze(E) indicates a dry working fluid property [20].

| Properties                  |                   |
|-----------------------------|-------------------|
| Chemical name               | Trans-1,3,3,3-Tetrafluoroprop-1-ene |
| Molecular formula           | CF_3CH_3=CHF      |
| Critical temperature        | 109.4°C           |
| Critical pressure           | 3.636 MPa         |
| Normal boiling point        | -18.95°C          |
| Appearance                  | Colorless         |
| Molecular weight            | 114 g/mol         |
| ODP                         | 0                 |
| GWP                         | <1                |
| Safety classification       | A2L               |

2.2. ORC system and its thermodynamic process

The current subcritical organic Rankine cycle (SORC), that using R1234ze(E) is consists of; a pump, an evaporator, a vapor generator, a turbine, and a condenser. The schematic diagram of the present SORC system and the thermodynamic process of SORC are shown in Figures 1 & 2, respectively. Also, the thermodynamic state points mention in the following figures.

![Figure 1. Schematic of the studied SORC system.](image-url)
In this SORC system, the saturated liquid of working fluid pressurized under subcritical state by a feed pump to subcooled liquid (1-2 process). The subcritical working fluid absorbs heat from fluegas as a heat source to converting the phase of saturated liquid to saturated vapor by evaporator (2-5 process). Then the working fluids converted to superheated fluid by absorbing heat from fluegas as a heat source in vapor generator (5-6 process). The vapor expands in the turbine and generates optimum power (6-7 process). The exhausted working fluid is cooled to saturated liquid in a condenser (7-1 process) by using cooling water as a heat sink to complete a cycle.

**Figure 2.** Thermodynamic process of SORC system with using fluegas as a heat source, cooling water as a heat sink, and driven by R1234ze(E) as a working fluid.

2.3. **Model boundary conditions and optimized parameters**

Boundary conditions of SORC system are listed in Table 2. The heat source of current SORC system is wet fluegas combustion and burn from natural gas as a low and medium temperature open type heat source without the outlet temperature limit. The inlet temperature of fluegas is set on 180°C and 220°C as low and medium temperature in 0.1013 MPa and the outlet temperature of heat source has no restriction and can be reduce to ambient temperature or $T_2 + \Delta T_{\text{evp,min}}$.

In present SORC the outlet temperature of vapor generator, $T_6$ and the outlet pressure of turbine, $P_7$ were optimized based on generating optimum net power output (>3MW) in terms of specific mass flow rates (300 kg.s⁻¹) of R1234ze(E) as a working fluid. The lower limit of outlet temperature of vapor generator was selected to avoid expansion process does not pass through the two phase region and the higher limit of the outlet temperature of vapor generator is $T_{HS,\text{in}} - \Delta T_{\text{evp,min}}$, based on flash tolerance of R1234ze(E) in evaporator. Also limit of the outlet pressure of turbine was optimized according to outlet pressure of vapor generator that has direct effect from outlet temperature of vapor generator to reach the maximum optimum net power output. In this case, the lower limit of outlet
pressure of turbine is selected as 0.8 MPa to generate optimum net power output, to avoid temperature cross in condensation process, and on the other hand, to avoid sub-atmospheric pressure in the cycle and to mitigate air ingress, the outlet pressure of turbine should be equal or higher than 0.1 MPa, \( P \geq 0.1 \). But the higher limit of the outlet pressure of turbine is selected as 1.3 MPa to generate optimum net power output (>3MW) in current SORC system.

Another boundary condition is the outlet temperature of turbine that effective from outlet pressure of turbine, should be equal or higher than dew point temperature of R1234ze(E) as a working fluid of SORC at condensation pressure, to avoid liquid droplet formation in turbine, \( T \geq T_{\text{dew}}(P_{\text{cond}}) \).

Figure 2 depicts the evaporator minimal temperature difference, \( \Delta T_{\text{evap,min}} \) (pinch point of evaporator) locates on the inlet of evaporator (working fluid evaporation bubble point) with 10°C. Also the outlet temperature of fluegas as a heat source of current SORC system modified to reach to the limit of the evaporator minimal temperature difference.

### Table 2. Boundary conditions and constraints for the SORC system.

| Parameter                                      | Symbol | Value       |
|------------------------------------------------|--------|-------------|
| R1234ze(E) mass flow rate/kg.s\(^{-1}\)         | \( m_{\text{WF}} \) | 300         |
| fluegas mass flow rate/kg.s\(^{-1}\)           | \( m_{\text{HS}} \)     | 500         |
| fluegas inlet temperature/C                    | \( T_{\text{HS,in}} \)   | 180-220     |
| fluegas pressure/kpa                           | \( P_{\text{HS}} \)      | 101.325     |
| Evaporator minimal temperature difference/C    | \( \Delta T_{\text{evap,min}} \) | 10          |
| Condenser minimal temperature difference/C     | \( \Delta T_{\text{cond,min}} \) | 10          |
| Cooling water inlet temperature/C              | \( T_{\text{CS,in}} \)   | 20          |
| Cooling water pressure/kpa                     | \( P_{\text{CS}} \)      | 101.325     |
| Environment temperature/C                      | \( T_{0} \)              | 20          |
| Environment pressure/kpa                       | \( P_{0} \)              | 101.325     |
| Feed pump pressure head/m                      | \( H \)                  | 39.6317     |
| Feed pump efficiency/%                         | \( \eta_{p} \)           | 86          |
| Turbine efficiency/%                           | \( \eta_{t} \)           | 72          |

#### 2.4. Assumptions

The common following assumptions are made to simplify the current SORC analysis:

- The SORC system is operated under a steady state.
- Heat dissipation and pressure drop in pipes and heat exchangers are negligible.
- Effects of the fluid kinetic and gravitational potential energy are negligible.
- And the heat exchangers are in a counter flow arrangement.

#### 2.5. System equations

The heat absorption capacity of the SORC system is:

\[
Q_{\text{SORC}} = m_{\text{HS}}(h_{\text{HS,in}} - h_{\text{HS,out}})
\]  \( (1) \)

Where \( m_{\text{HS}} \) is mass flow rate of fluegas as a heat source, also \( h_{\text{HS,in}} \) and \( h_{\text{HS,out}} \) are fluegas as a heat source enthalpies at the system inlet and outlet, respectively.

The net power output of SORC system is calculated as:

\[
W_{\text{net}} = W_{t} - W_{p}
\]  \( (2) \)

Where \( W_{t} \) is power generated by turbine, \( W_{p} \) is power consumed by feed pump.

The total efficiency of SORC system is calculated as:

\[
\eta_{\text{SORC}} = \frac{W_{\text{net}}}{Q_{\text{SORC}}}
\]  \( (3) \)

Based on the reference of environment state, is selected as \( T_{0}=293.15 \text{ K} \) and \( P_{0}=101.325 \text{ kPa} \). The exergy released by the heat source in the evaporator and vapor generator system is calculated as:
\[ \Delta E_{HS} = \dot{m}_{HS}(h_{HS,\text{in}} - h_{HS,\text{out}} - T_0(S_{HS,\text{in}} - S_{HS,\text{out}}) \]  

(4)

Where \( S_{HS,\text{in}} \) and \( S_{HS,\text{out}} \) are heat source entropies at inlet and outlet, respectively. The exergy absorbed by the working fluid in the evaporator and vapor generator system is:

\[ \Delta E_{\text{evap}} = \dot{m}_{\text{wf}}(h_6 - h_2 - T_0(S_6 - S_2)) \]  

(5)

Where \( \dot{m}_{\text{wf}} \) is mass flow rate of working fluid, also \( h_6 \) and \( h_2 \) are working fluid enthalpies at the outlet of vapor generator and inlet of evaporator, respectively. Furthermore, \( S_6 \) and \( S_2 \) are working fluid entropies at the outlet of vapor generator and inlet of evaporator, respectively. Therefore, the exergy loss (the unused exergy) in the evaporator and vapor generator system is:

\[ l_{\text{evap}} = \Delta E_{HS} - \Delta E_{\text{evap}} = T_0(\dot{m}_{\text{wf}}(S_6 - S_2) - \dot{m}_{\text{HS}}(S_{HS,\text{in}} - S_{HS,\text{out}})) \]  

(6)

The exergy loss in the turbine is:

\[ l_t = T_0\dot{m}_{\text{wf}}(S_7 - S_{7s}) \]  

(7)

Where \( S_7 \) and \( S_{7s} \) are working fluid entropies at outlet and isentropic state of turbine, respectively. The exergy loss in the feed pump is:

\[ l_{\text{pump}} = T_0\dot{m}_{\text{wf}}(S_2 - S_{2s}) \]  

(8)

Where \( S_2 \) and \( S_{2s} \) are working fluid entropies at outlet and isentropic state of feed pump, respectively. Look like the heat transfer process in the evaporator and vapor generator system, the exergy released by the working fluid in the condenser is:

\[ \Delta E_{\text{cond}} = \dot{m}_{\text{wf}}(h_7 - h_1 - T_0(S_7 - S_1)) \]  

(9)

Where \( h_1 \) and \( h_7 \) are working fluid enthalpies at the outlet and inlet of condenser, respectively. Furthermore, \( S_1 \) and \( S_7 \) are working fluid entropies at the outlet and inlet of condenser, respectively. And the exergy absorbed by the heat sink in the condenser is calculated as:

\[ \Delta E_{CS} = \dot{m}_{CS}(h_{CS,\text{out}} - h_{CS,\text{in}} - T_0(S_{CS,\text{out}} - S_{CS,\text{in}})) \]  

(10)

Where \( \dot{m}_{CS} \) is mass flow rate of cold stream as a heat sink, also \( h_{CS,\text{out}} \) and \( h_{CS,\text{in}} \) are heat sink enthalpies at the outlet and inlet, respectively. In addition, \( S_{CS,\text{out}} \) and \( S_{CS,\text{in}} \) are heat sink entropies at the outlet and inlet, respectively. Therefore, the exergy loss of condenser is:

\[ l_{\text{cond}} = \Delta E_{\text{evap}} - \Delta E_{CS} = T_0(\dot{m}_{CS}(S_{CS,\text{out}} - S_{CS,\text{in}}) - \dot{m}_{\text{wf}}(S_7 - S_1)) \]  

(11)

In current SORC system, the coefficient of exergy loss in each process is:

\[ \xi_i = \frac{l_i}{\Delta E_{HS}} \]  

(12)

Finally, the overall exergy efficiency is calculated as:

\[ \eta_{ex} = \frac{W_{net}}{\Delta E_{HS}} = 1 - \sum \xi_i \]  

(13)

3. Results and Discussion

Figure 3 shows the effect of vapor generator outlet temperature and turbine outlet pressure on the net power output of SORC driven by R1234ze(E) at \( \dot{m}_{\text{wf}}=300 \text{ kg.s}^{-1} \) and \( \dot{m}_{\text{HS}}=500 \text{ kg.s}^{-1} \). As shown in Figure 3, with increasing the outlet temperature of vapor generator \( (T_6) \) in the specific range of outlet pressure of turbine \( (P=0.8-1.3 \text{ MPa}) \), the net power output of SORC system increases significantly. The superheat degree of vapor generator increases from 5 to 50 \(^\circ\text{C}\), because of flash tolerance in evaporator. Furthermore, the limit range of the turbine outlet pressure is from 0.8 MPa for preventing sub-atmospheric pressure in the cycle and to mitigate air ingress in condenser, to 1.3 MPa for
generating optimum net power output in this SORC system (>3 MW). In a constant turbine outlet pressure, with increasing of mass enthalpy change and temperature in expansion process lead to increase of net power output of current SORC system, steadily because of conversion of enthalpy to power in turbine. On the other hand, with reducing of turbine outlet pressure and temperature cause to increase the mass enthalpy change in expansion process and as a result, increases of net power output considerably. These results are in agreement with results of Oyewunmi et al. [49].

![Figure 3](image3.png)

**Figure 3.** Effects of the vapor generator outlet temperature and turbine outlet pressure on the net power output of SORC system driven by R1234ze(E).

Figure 4 depicts the simulation model of SORC efficiency calculation based on equations 1-3 by using Matlab Simulink software to increase accuracy of analysis.

![Figure 4](image4.png)

**Figure 4.** Simulation model analysis of SORC efficiency.
As displayed in Figures 5 (a and b), the influence of the vapor generator outlet temperature ($T_6$) and turbine outlet pressure ($P_7$) on the total efficiency of current SORC system ($\eta_{\text{SORC}}$) which driven by R1234ze(E) as a refrigerant working fluid in two different inlet fluegas temperature, $T_{\text{HS}}=180^\circ\text{C}$ and $T_{\text{HS}}=220^\circ\text{C}$ at atmospheric fluegas pressure, $P_{\text{HS}}=0.101325$ MPa is considered, respectively.

![Figure 5](image_url)

**Figure 5.** Effects of the vapor generator outlet temperature and turbine outlet pressure on the total efficiency of the SORC system driven by R1234ze(E): (a) $T_{\text{HS}}$=180°C and (b) $T_{\text{HS}}$=220°C.

With reducing of the turbine outlet pressure, total efficiency of current SORC system is increased because of increasing of net power output especially in output of turbine generation. However, with increasing the vapor generator outlet temperature, the outlet fluegas mass enthalpy is slightly increased and as a result can causes to a bit glide in fluegas mass enthalpy change then leads to a little increases
the heat absorption capacity of current SORC system and in final the total efficiency slightly decreases because of reverse effect of heat absorption capacity in evaporator and vapor generator system with total efficiency of current SORC system. Figure 5 (a) and (b) have been differences in mass enthalpy of inlet fluegas likewise mass enthalpy of outlet fluegas from evaporator but non-significant because of changing inlet temperature of the fluegas from 180°C to 220°C. The overall measurement results of total efficiency of this SORC system are similar to those produced by Li et al. [1], Hamdi et al. [52], Ge et al. [38], and Thurairaja et al. [53].

**Figure 6.** Effects of the vapor generator outlet temperature and turbine outlet pressure on the fluegas outlet temperature: (a) $T_{\text{HS,in}}=180^\circ\text{C}$ and (b) $T_{\text{HS,in}}=220^\circ\text{C}$. 
Figures 6 (a) and (b) reveal the Effects of the vapor generator outlet temperature (from 5°C to 50°C superheated increase of R1234ze(E) with pay attention of flash tolerance in evaporation system process) and turbine outlet pressure on the fluegas outlet temperature based on two different inlet temperature of fluegas, $T_{HS,in}=180\, ^{\circ}C$ and $T_{HS,in}=220\, ^{\circ}C$, respectively on fluegas outlet temperature.

As shown in Figures 6 (a) and (b) with increasing of vapor generator outlet temperature in specific turbine inlet pressure, the fluegas outlet temperature reducing significantly, and as a result the enthalpy change in evaporation process system (that including evaporator and vapor generator) increases and in final the power generation by turbine increases consequently. Meanwhile, in a specific outlet temperature of vapor generator, with increasing the outlet pressure of turbine, the fluegas outlet temperature is increasing significantly and leads to decrease the heat absorption capacity of current SORC system and as a result, causes to reducing the power generation by turbine. This result is similar to another research that conducted, and analyzed by Li et al. [1].

![Diagram](image)

**Figure 7.** Simulation model analysis of SORC exergy efficiency.

![Graph](image)
Figure 8. Effects of the vapor generator outlet temperature and turbine outlet pressure on the exergy efficiency of SORC system: (a) $T_{\text{HS,in}}=180^\circ$C and (b) $T_{\text{HS,in}}=220^\circ$C.

Figure 7 depicts the simulation model of SORC exergy efficiency calculation based on equation 13 by using Matlab Simulink software to increase accuracy of analysis. As shown in Figures 8 (a) and (b), effect of the vapor generator outlet temperature and turbine outlet pressure on the exergy efficiency of SORC system that driven by R1234ze(E) is considered by using equations 2, 4, and 13 in (a) $T_{\text{HS,in}}=180^\circ$C and (b) $T_{\text{HS,in}}=220^\circ$C, respectively. Also the exergy efficiency of SORC system is modeled and analyzed by matlab simulink software that shows in Figures 7. Figure 8 (a) and (b) depict, in specific outlet pressure of turbine, with increasing the vapor generator outlet temperature, the exergy efficiency is increasing significantly, because the net power output increases significantly and on the other hand, the enthalpy change of fluegas in evaporator system is increasing but the entropy change of fluegas in evaporator system is decreasing. However, in $T_{\text{HS,in}}=220^\circ$C compare with $T_{\text{HS,in}}=180^\circ$C, the entropy change reduces and leads to reducing exergy efficiency of current SORC system. The result of the SORC exergy as shown in the figure 8 (a) and (b) is in agreement with the result of Zhu et al. [54] and Hamdi et al. [52].

4. Conclusion

Current subcritical organic rankine cycle (SORC) driven by R1234ze(E) as an environmental friendly refrigerant working fluid and consist of a pump (compression process to generate subcritical pressure state), an evaporator and a vapor generator (evaporation process) that using wet fluegas combustion and burn from natural gas as a medium-temperature open type heat source (180-220$^\circ$C) without the outlet temperature limit, a turbine (expansion process) and a condenser (condensation process). In this SORC system, the outlet temperature of vapor generator, $T_o$ and the outlet pressure of turbine, $P_7$ were optimized and analyzed based on generating optimum net power output, then the fluegas outlet temperatures were considered, also overall system efficiency and exergy efficiency in the view of R1234ze(E) as a working fluid, fluegas as a heat source, and water as a heat sink mass flow rate: 300, 500, and 500 kg.s$^{-1}$ respectively, were studied. The results are detailed below.

- For the current SORC system that using R1234ze(E), the variation of the superheat degree increasing of vapor generator (5-50$^\circ$C) and increasing turbine outlet pressure (0.8-1.3 MPa) has a significant increasing effect on the system net power output. The outlet temperature of vapor generator increases from 5 to 50$^\circ$C, because of flash tolerance in evaporator. Also, the limit range of the turbine outlet pressure is from 0.8 MPa for preventing sub-atmospheric pressure in the cycle and to mitigate air ingress in condenser, to 1.3 MPa in the view of the subcritical state and leads to generate optimum net power output in this SORC system by
expansion process in turbine (>3 MW). With increasing of mass enthalpy change and temperature in expansion process, because of conversion of enthalpy to power in turbine, causes to increase of net power output of current SORC system, steadily. And the maximum net power output with 5.2675 MW was 53.1% increases compare with the minimum net power output of current SORC system.

- Total efficiency of SORC system has an effective from net power output of system and has slightly revers relationship with outlet fluegas mass enthalpy based on variation of outlet vapor generator temperature and outlet pressure of turbine. Therefore, by decreasing of the turbine outlet pressure, total efficiency of current SORC system is increased because of increasing of net power output. The highest efficiency with 0.08107 at \( T_{HS}=220 \) °C, was 32.1% increased compare with the lowest efficiency of SORC system.

- Increasing of vapor generator outlet temperature in specific turbine inlet pressure leads the fluegas outlet temperature reducing significantly then causes to increase enthalpy change in evaporation process system and in final the power generation by turbine as well. But, in a specific outlet temperature of vapor generator, with increasing the outlet pressure of turbine, the fluegas outlet temperature is increasing significantly and as a result the heat absorption capacity of SORC system and the net power output were decrease significantly.

- By two main reason of increasing the enthalpy change of fluegas in evaporator system and decreasing the entropy change of fluegas in evaporator system, and on the other hand, increasing of net power output, the exergy efficiency of SORC system increasing significantly, and received to the 0.328 at \( T_{HS}=180 \) °C that was 50.8% increases compare with lowest total exergy efficiency of SORC system at \( T_{HS}=220 \) °C.

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Data Availability
The data used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest
The authors declare that there is no conflict of interest.

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Nomenclature








Greek Symbols

η  efficiency
ξ  coefficient of exergy loss

Abbreviations

SORC  subcritical organic rankine cycle
ODP  ozone depletion potential
GWP  global warming potential
R1234ze(E)  trans-1,3,3,3-tetrafluoroprop-1-ene

1-8  sate points shown in figure 2