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Research Paper

# Thermodynamic analysis and comparison between $CO_2$ transcritical power cycles and R245fa organic Rankine cycles for low grade heat to power energy conversion

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In this paper, a theoretical study is conducted to investigate and compare the performance of $CO_2$ transcritical power cycles (T-$CO_2$) and R245fa organic Rankine cycles (ORCs) using low-grade thermal energy to produce useful shaft or electrical power. Each power cycle consists of typical Rankine cycle components, such as a working fluid pump, gas generator or evaporator, turbine with electricity generator, air cooled condenser and recuperator (internal heat exchanger). The thermodynamic models of both cycles have been developed and are applied to calculate and compare the cycle thermal and exergy efficiencies at different operating conditions and control strategies. The simulation results show that the system performances for both cycles vary with different operating conditions. When the heat source (waste heat) temperature increases from 120 °C to 260 °C and heat sink (cooling air) temperature is reduced from 20 °C to 0 °C, both thermal efficiencies of R245fa ORC and T-$CO_2$ with recuperator can significantly increase. On the other hand, R245fa ORC and T-$CO_2$ exergy efficiencies increase with lower heat sink temperatures and generally decrease with higher heat source temperatures. In addition, with the same operating conditions and heat transfer assumptions, the thermal and exergy efficiencies of R245fa ORCs are both slightly higher than those of T-$CO_2$. However, the efficiencies of both cycles can be enhanced by installing a recuperator in each system at specified operating conditions. Ultimately, optimal operating states can be predicted, with particular focus on the working fluid expander inlet pressure for both cycles.



## Nomenclature

| | |
|---|---|
| $E$ | exergy energy (W) |
| $GWP$ | global warming potential |
| $h$ | enthalpy (J/kg) |
| $I$ | exergy destruction (W) |
| LEL | lower explosive limit |
| $\dot{m}$ | mass flow rate (kg/s) |
| $ODP$ | Ozone Depletion Potential |
| $OEL$ | occupational exposure limit |
| $P$ | pressure (pa) |
| $Q$ | heat transfer rate (W) |
| $q$ | specific heat transfer rate (J/kg) |
| $S,s$ | entropy (J/kg K) |
| $T$ | temperature (K) |
| $\overline{T}$ | average temperature (K) |
| W | work input/output (W) |
| $\eta$ | efficiency (–) |
| $\varepsilon$ | heat transfer effectiveness (–) |

### *Subscripts*

| | |
|---|---|
| *a* | air |
| *b* | boiling point |
| *c* | critical point |
| *cd* | condenser |
| *exp* | expander |
| *f* | working fluid |
| *gg* | gas generator |
| *in* | input |
| *is* | isentropic |
| *max* | maximum |
| *min* | minimum |
| *oil* | thermal oil |
| *pmp* | pump |
| *rec* | recuperator |
| *source* | heat source |
| *tot* | total |
| 0 | dead state |

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## 1. Introduction

Currently, global power generation is predominantly from the combustion of fossil fuels, which instigates serious problems in terms of atmospheric pollution, excess $CO_2$ emissions, and energy resource depletion. Therefore there is an urgent requirement to generate more power from low-grade heat sources such as solar thermal [1], biomass [2,3], geothermal [4] and industrial waste heat [5].

For a low-grade heat energy conversion system, the conventional steam Rankine cycle cannot achieve a high thermal efficiency and compactable system size and thus is not an appropriate nor economic option [6–8]. This is because a low-grade heat source cannot produce steam at high enough temperatures and pressures required by the steam turbine. In contrast, an Organic Rankine Cycle (ORC) is a more feasible option for the application of low-grade heat sources in terms of operating parameters, system size, thermal and exergy efficiencies. The ORC functions similarly to a Clausius-Rankine steam power plant, but uses an organic working fluid such as R245fa instead, which is able to condense at a lower pressure and evaporate at a higher pressure. The most challenging aspect for a low-grade energy conversation system design is to select an appropriate working fluid and high efficient organic Rankine cycle. At a fixed working fluid temperature range, both subcritical and supercritical ORCs with up to 31 pure working fluids were analysed thermodynamically by Saleh et al. [9]. The thermal efficiencies ranged between 0.36% and 13%, indicating the importance of thermodynamic cycles and working fluid selections. It was also found that supercritical working fluids can receive a greater heat transfer from sensible heat sources such as waste heat compared to those of subcritical ones when considering matchable cold and hot side temperature glides in the heat exchanger. Consequently, at the same operating conditions, the working fluid temperature at the turbine inlet will be relatively higher for the supercritical heat addition process and thus enhance its cycle thermal efficiency. A recuperator can be integrated into an ORC system but it may or may not enhance the system thermal efficiency depending on the working fluid, applicable fluid state at the expander outlet and application. An ORC working fluid can be classified as dry, wet or an isentropic fluid depending on the slope of its saturated vapour curve [10]. For a dry working fluid such as R245fa, the fluid at the ORC turbine outlet is superheated and its temperature is high enough to heat up the liquid fluid from the ORC pump, thereby boosting system performance when the recuperator is installed. Such circumstances may change when the ORC is applied in different heat recovery systems. In an application of solar ORC using R245fa as a working fluid, an experimental rig was developed in which a solar collector acted as the ORC evaporator [11]. The introduction of recuperator into the system increased the temperature at the collector inlet and thus lead to a reduced collector performance and system thermal efficiency. In addition, an ORC thermal efficiency can also be affected by the performance of system components, particularly the expander. Kang [12] experimented on an R245fa ORC system with nominal 30 kW power generation using a radial turbine as expander. The maximum average turbine and overall thermal efficiencies were found to be 78.7% and 5.22% respectively, which both increased with higher evaporator temperatures when the expander pressure ratio was fixed. In addition, the overall thermal efficiency could be further improved with enhanced turbine efficiency.

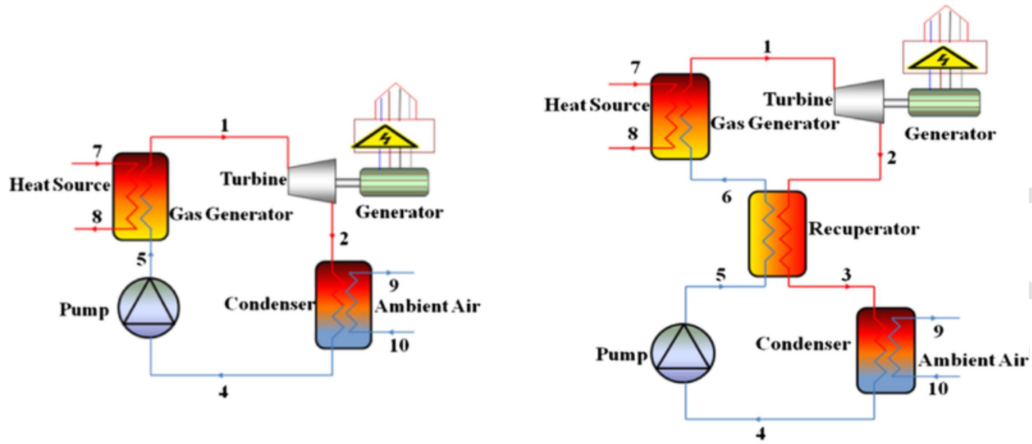
It is worth noting that R245fa is now widely applied in ORCs considering its zero Ozone Depletion Potential (ODP) and appropriate thermophysical properties. Nonetheless it is still classified as a HFC and does have a relatively high Global Warming Potential (GWP), which will undoubtedly affect its future application in ORCs. On the other hand, as a natural working fluid, $CO_2$ has been widely applied into refrigeration (Ge et al. [13]) and heat pump (Jiang et al. [14]) systems due to its zero ODP, negligible GWP and superb thermophysical properties, with an exception of its high critical pressure and low critical temperature. The high operating pressures of a $CO_2$ energy system requires special designs for system components and controls, while the low critical temperature will turn a $CO_2$ low-grade power generation system into a transcritical Rankine (T-$CO_2$) or supercritical Brayton cycle (S-$CO_2$). To tackle high operating pressures, instead of using a pure $CO_2$ working fluid, a zeotropic mixture of $CO_2$ and another fluid such as R1234yf or R1234ze could be used in the low-grade power generation although further efficiency improvement is needed [15]. Kim et al. [16] conducted a comparison between T-$CO_2$ and S-$CO_2$ cycles in terms of energy and exergy analyses. They found that the T-$CO_2$ is better equipped for low-grade heat sources due to the thermal match in heat transfer process of high pressure side. Furthermore, Velez et al. [17] conducted a theoretical analysis of low-grade power generation with T-$CO_2$. Simulation results showed that exergy and energy efficiencies could increase up to 25% and 300% respectively when the turbine inlet temperature increased from 60 to 150 °C at different expander inlet pressures. Similar T-$CO_2$ low temperature power cycles for different applications have also been carried out by various researchers [18,19].

It is recognized that $CO_2$ is a promising working fluid to be applied into a low-grade power generation system with T-$CO_2$ cycles. However, the thermodynamic analysis and performance comparison between the T-$CO_2$ and conventional R245fa ORC systems at their applicable operating conditions need to be further investigated, which has not been thoroughly implemented so far. In this paper, the thermal and exergy efficiencies at different heat source and sink temperatures are calculated and analysed for the T-$CO_2$ and R245fa ORC systems with and without an integrated recuperator. The predictions and analyses will contribute towards justifying the feasibility of applying T-$CO_2$ into low-grade power generation and further development in this area.

## 2. Systems description

The system schematic diagrams to be analysed in this paper are shown in Fig. 1. The only difference between Fig. 1(a) and (b) is the presence or not of a recuperator in the system, which is used as an internal heat exchanger to improve the performance of the system. The system components shown in Fig. 1(a) include a liquid pump, gas generator/evaporator, turbine/expander with electricity generator and condenser. Operationally, the liquid working fluid from the condenser outlet is drawn into the pump (point 4) and thus pressurised to point 5. It then flows into the gas generator, where it absorbs heat from the heat source to be vaporised and superheated. The vapour working fluid with high pressure and high temperature then expands in the expander (point 1) to generate electrical power through the electricity generator. After expansion (point 2), the low-pressure vapour enters the condenser where it is condensed into its liquid state (point 4). Finally, the working fluid flows back to the pump (point 4) and the cycle repeats. As shown in Fig. 1(b), a recuperator can also be installed just after the expander so as to desuperheat the fluid from the expander outlet and in the meantime preheat the liquid after the pump. The installation of a recuperator in the system is expected to reduce the heating and cooling demands from the heat source and sink respectively when the system power generation is specified.

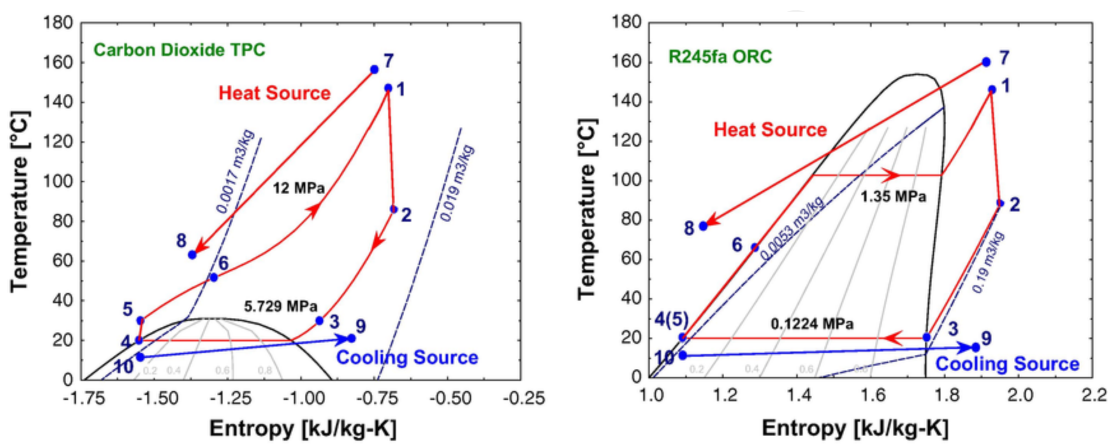
Corresponding to Fig. 1(b), sample T-S diagrams for the T-$CO_2$ and R245fa ORC systems are depicted in Fig. 2(a) and (b) respec-




(a) Schematic diagram of T-$CO_2$/R245fa ORC without recuperator

(b) Schematic diagram of T-$CO_2$/R245fa ORC recuperator

**Fig. 1.** (a) Schematic diagram of T-$CO_2$/R245fa ORC without recuperator. (b) Schematic diagram of T-$CO_2$/R245fa ORC recuperator.




(a) T-S diagram of T-$CO_2$

(b) T-S diagram of R245fa ORC

**Fig. 2.** (a) T-S diagram of T-$CO_2$. (b) T-S diagram of R245fa ORC.

tively. It should be noted that the T-S diagram for the T-$CO_2$ is based on the specifications of 12 MPa and 5.729 MPa for the supercritical gas generator and condenser pressures respectively. While for the T-S diagram of the R245fa ORC, the evaporator and condenser pressures are specified as 1.35 MPa and 0.1224 MPa each. Based on their operational pressure ranges, turboexpanders (turbines) can be used in T-$CO_2$ cycles while both turboexpanders (turbines) and scroll expanders are applicable for R245fa ORCs. In addition, for both systems, the heat source and sink temperatures are assigned as 160 °C and 10 °C individually. All state points in the diagrams are calculated by EES® (Engineering Equation Solver) [20].

As depicted in Fig. 2, the working fluid temperature profile in heat addition process of T-$CO_2$ matches well with the sensible heat source flow compared to the corresponding temperature profiles of R245fa ORC which has an obvious pinch point. It can be expected that at the same pinch point temperature difference and expander inlet temperature of heat addition processes, higher heat source temperature will be required for a R245fa ORC system.

## 3. Thermodynamic models

### *3.1. Working fluid properties*

Different working fluids can be used in low-grade power generation systems. However, the appropriate working fluids should reveal good thermophysical properties, none safety issues and less environmental impacts. Accordingly, in this paper, the $CO_2$ and R245fa are selected and analysed in the transcritical and subcritical ORC power cycles respectively. Some relevant thermophysical, safety and environmental data of $CO_2$ and R245fa are listed in Table 1 [21] based on ASHRAE 34 and REFPROP 9.0 such that the data accuracies are acceptable.

### *3.2. Assumptions for the thermodynamic analysis*

The following assumptions have been made for the analysis of each system and corresponding cycle:

(1) The system operates under steady state.

**Table 1**
Thermophysical safety and environmental data for $CO_2$ (R744) and R245fa.

| Substance | Thermophysical data | | | | Safety data | | | Environmental data | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Molecular mass | $T_b$ (°C) | $T_c$ (°C) | $P_c$ (Mpa) | OEL (PPMv) | LEL (%) | ASHRAE safety group | Atmospheric (yr) | ODP | GWP |
| $CO_2$ (R744) | 44.01 | −78.4 | 31.1 | 7.38 | 5000 | None | A1 | >50 | 0 | 1 |
| R245fa | 134.05 | 15.1 | 154 | 3.65 | 300 | None | B1 | 7.6 | 0 | 1030 |

(2) The heat and friction losses, the kinetic and potential energy, as well as pressure drops of the working fluid through the system are neglected.
(3) The hot thermal oil is used as the heat source and the temperature difference between heat source and expander inlet is 20 K; the thermal oil flow rate is 1.2 kg/s.
(4) The ambient air is used as heat sink and the working fluid state at condenser outlet is saturated liquid which has a temperature of 10 K higher than the incoming air flow; the air mass flow rate is 5 kg/s.
(5) The isentropic efficiencies of the pump and turbine are both set to 85% based on previous research outcomes [16,22] while the effectiveness of recuperator is 0.8. It should be noted that the turbine isentropic efficiency is assumed based on T-$CO_2$. For an R245fa ORC when a scroll expander is applied, the isentropic efficiency could be lower [23]. To fairly compare the system performances of T-$CO_2$ and R245fa ORC, the same isentropic efficiency is assumed in this paper.
(6) The dead state pressure and temperature are 1 bar (atmospheric pressure) and ambient air respectively for exergy analysis.
(7) The design power generation from expander is 5 kW.

*3.3. Energy calculations*

The purpose of energy analysis is to evaluate system performance based on the first law of thermodynamics in term of thermal efficiency which can be calculated as Eq. (1).

$$\eta_{th} = \frac{W_{net}}{Q_{gg}} \tag{1}$$

where the $W_{net}$ and $Q_{gg}$ are system net power output and heat input to gas generator or evaporator respectively. To obtain these two performance parameters, energy balance calculation for each system component is necessary.

(i) Gas generator or evaporator

The heat capacity:

$$\begin{aligned} Q_{gg} &= \dot{m}_f(h_1 - h_5) \\ &= \dot{m}_{oil}(T_7 - T_8), \quad \text{for system without recuperator} \end{aligned} \tag{2}$$

$$\begin{aligned} Q_{gg} &= \dot{m}_f(h_1 - h_6) \\ &= \dot{m}_{oil}(T_7 - T_8), \quad \text{for system with recuperator} \end{aligned} \tag{3}$$

(ii) Expander

The power output:

$$W_{\exp} = \dot{m}_f(h_1 - h_2) \tag{4}$$

The isentropic efficiency

$$\eta_{\exp} = \frac{h_1 - h_2}{h_1 - h_{2,is}} \tag{5}$$

(iii) Recuperator

The effectiveness:

$$\varepsilon_{rec} = \frac{h_2 - h_3}{q_{rec,\max}} = \frac{h_2 - h_3}{h_2 - h_{3,\min}} \tag{6}$$

where $h_{3,\min}$ is calculated based on the temperature at point 5 and pressure at point 3.

The heat capacity:

$$Q_{rec} = \dot{m}_f(h_2 - h_3) = \dot{m}_f(h_6 - h_5) \tag{7}$$

(iv) Condenser

The heat capacity:

$$\begin{aligned} Q_{cd} &= \dot{m}_f(h_2 - h_4) \\ &= \dot{m}_a(T_9 - T_{10}), \quad \text{for system without recuperator} \end{aligned} \tag{8}$$

$$\begin{aligned} Q_{cd} &= \dot{m}_f(h_3 - h_4) \\ &= \dot{m}_a(T_9 - T_{10}), \quad \text{for system with recuperator} \end{aligned} \tag{9}$$

(v) Pump

The power input:

$$W_{pmp} = \dot{m}_f(h_5 - h_4) \tag{10}$$

The isentropic efficiency

$$\eta_{pmp} = \frac{h_{5,is} - h_4}{h_5 - h_4} \tag{11}$$

(vi) The net power output

$$W_{net} = W_{\exp} - W_{pmp} \tag{12}$$

### 3.4. Exergy calculations

The thermal efficiency alone however is not enough to evaluate and characterize the quality of system components such as heat exchangers. To achieve these, exergetic analysis based on the second law of thermodynamics is required [24–26]. The exergetic analysis is necessary to understand the extent of irreversibility in each component process, identify where the most irreversibility is and therefore the potential of improvements. The component exergy destructions of both T-$CO_2$ and R245fa ORC systems are calculated as below:

(i) Gas generator or evaporator

$$I_{gg} = \dot{m}_f T_0 \left( s_1 - s_5 - \frac{q_{gg}}{\bar{T}_{source}} \right) = \dot{m}_f T_0 \left( s_1 - s_5 - \frac{h_1 - h_5}{\bar{T}_{source}} \right), \quad \text{for system without recuperator} \tag{13}$$

$$I_{gg} = \dot{m}_f T_0 \left( s_1 - s_6 - \frac{q_{gg}}{\bar{T}_{source}} \right) = \dot{m}_f T_0 \left( s_1 - s_6 - \frac{h_1 - h_6}{\bar{T}_{source}} \right), \quad \text{for system with recuperator} \tag{14}$$

(ii) Expander:

$$I_{\exp} = \dot{m}_f T_0 (s_2 - s_1) \tag{15}$$

(iii) Recupertaor

$$I_{rec} = \dot{m}_f T_0 (s_6 + s_3 - s_5 - s_2) \tag{16}$$

(iv) Condenser

$$I_{cd} = \dot{m}_f T_0 \left( s_4 - s_2 + \frac{q_{cd}}{\bar{T}_{\sin k}} \right) = \dot{m}_f T_0 \left( s_4 - s_2 + \frac{h_2 - h_4}{\bar{T}_{\sin k}} \right), \quad \text{for system without recuperator} \tag{17}$$

$$I_{cd} = \dot{m}_f T_0 \left( s_4 - s_3 + \frac{q_{cd}}{\bar{T}_{\sin k}} \right) = \dot{m}_f T_0 \left( s_4 - s_3 + \frac{h_3 - h_4}{\bar{T}_{\sin k}} \right), \quad \text{for system with recuperator} \tag{18}$$

(v) Pump

$$I_{pmp} = \dot{m}_f T_0 (s_5 - s_4) \tag{19}$$

The total exergy destruction:

$$\sum I_{tot} = I_{gg} + I_{\exp} + I_{cd} + I_{pmp}, \quad \text{for system without recuperator} \tag{20}$$

$$\sum I_{tot} = I_{gg} + I_{\exp} + I_{rec} + I_{cd} + I_{pmp}, \quad \text{for system with recuperator} \tag{21}$$

The system exergy input:

$$E_{in} = \sum I_{tot} + W_{net}$$

The system exergy efficiency:

$$\eta_{exg} = \frac{W_{net}}{E_{in}} \tag{22}$$

## 4. Performance evaluation, comparison and analysis

In order to conduct performance comparison between the T-$CO_2$ and R245fa ORC, the developed thermodynamic models are simulated at specific operating conditions and control strategies. These include different heat source and sink temperatures varying in a range of 120–260 °C and 0–20 °C, respectively. These specifications are reasonable since the heat source temperatures are applicable for most of low grade heat sources [27] and the ambient air is used as heat sink for these power cycles to be analysed. For the working fluid pressures in heat addition process of gas generators or evaporators, they vary in a range of 80–300 bar and 8–22 bar for T-$CO_2$ and R245fa ORC systems respectively. These pressure ranges can represent the applicable operating conditions for both power cycles. In addition, to facilitate the system applications, a recuperator is an option to be applied into each power cycle for the model simulation and comparison.

### *4.1. Thermal efficiency analysis*

At constant $CO_2$ expander inlet pressure (120 bar), the variations of system thermal efficiencies with heat source and sink temperatures for the T-$CO_2$ with and without recuperator are shown in Fig. 3. The simulation results show that at a constant heat source temperature, the thermal efficiency increases with lower heat sink temperature for both systems with and without recuperator. This can be explained that the lower heat sink temperature causes increased expander pressure ratio and thus power output or less required work fluid mass flow rate when the power output is fixed. The smaller working fluid mass flow rate indicates that the less heat source heat input is required and therefore higher thermal efficiency can be achieved. Simultaneously, at a constant heat sink temperature, the thermal efficiency rises with higher heat source temperature. However, the effect of heat source temperature on the thermal efficiency is more sensitive for the system with recuperator. When comparing the system performance with and without recuperator, at the same operating condition, the thermal efficiency is always higher for the system with recuperator.

At a constant evaporator pressure (14 bar), the variations of thermal efficiencies with heat source and sink temperatures for the R245fa ORC systems with and without recuperator are also simulated and depicted in Fig. 4. Similar to the T-$CO_2$ cycles, at a constant heat source temperature the thermal efficiency increases with lower heat sink temperature for the cycles with and without recuperator considering of its effect on expander pressure ratio. On the other hand, at a fixed heat sink temperature, the thermal efficiency increases with higher heat source temperature for the system with recuperator but decreases with increased heat source temperature for the cycle without recuperator. This demonstrates that an installation of recuperator in an R245fa ORC can benefit the system performance in term of thermal efficiency with higher heat source temperature.

To compare the performance of both T-$CO_2$ and R245fa ORC systems, at the same operating conditions of heat source and sink, the thermal efficiency of R245fa ORC is generally higher than that of T-$CO_2$. This is also based on the assumptions made in Section 3.2 for these two power cycles.

At constant heat sink temperature (10 °C), the variations of thermal efficiencies with heat source temperatures and $CO_2$ pressures at expander inlet are predicted and shown in Fig. 5 for the T-$CO_2$ systems with and without recuperator. It is seen that at a constant $CO_2$ pressure, the thermal efficiency increase with higher heat source temperature no matter if a recuperator is installed. On the other hand, at a constant heat source temperature, when the $CO_2$ pressure increases the thermal efficiency of both cycles (with and without recuperator) increases first, reaches to its peak value and then drops. This demonstrates that there is an optimum operating $CO_2$ pressure at expander inlet for the T-$CO_2$ cycles at fixed heat source and sink temperatures. For the effect of recuperator installation, the thermal efficiency of the cycle with recuperator is not always higher than that without recuper-

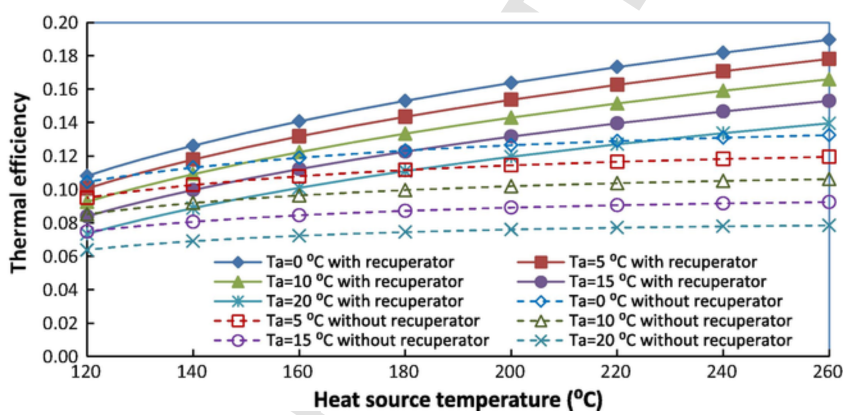



**Fig. 3.** Variations of thermal efficiencies with heat source and sink temperatures for T-$CO_2$.

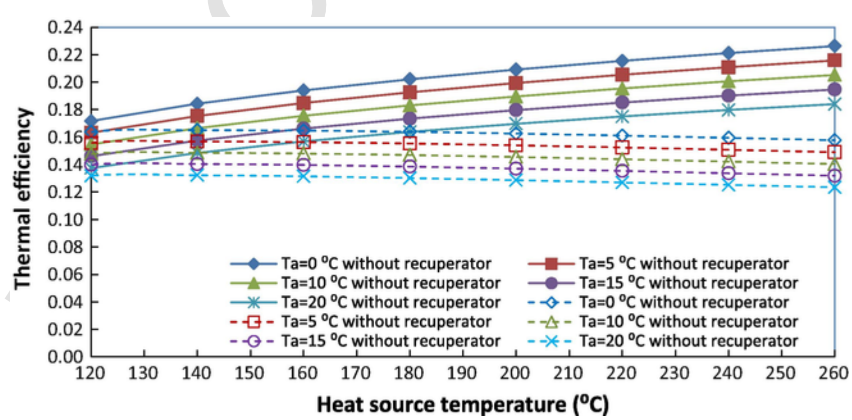



**Fig. 4.** Variations of thermal efficiencies with heat source and sink temperatures for R245fa ORC.

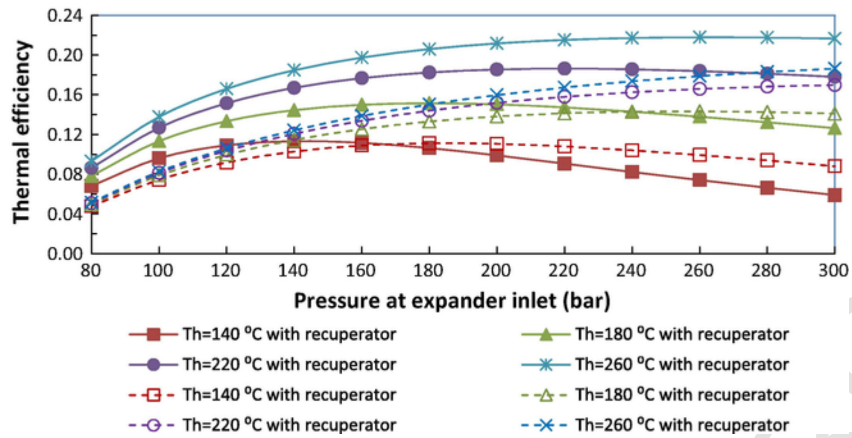



**Fig. 5.** Variations of thermal efficiencies with heat source temperatures and working fluid pressures at expander inlet for T-$CO_2$.

ator at different $CO_2$ pressures which depends also on the heat source temperature. When the heat source temperature is less than about 180 °C, the thermal efficiency for the cycle with recuperator is even lower than that without recuperator when the $CO_2$ pressure increases further.

For the effect of R245fa pressure at expander inlet as shown in Fig. 6, again at a constant R245fa pressure, the thermal efficiency increases with higher heat source temperature for the system with recuperator but mostly decreases with increased heat source temperature if a recuperator is not installed. Even so, the effect of heat source temperature on the thermal efficiency for the cycle without recuperator is not as significant as that with recuperator. Furthermore, different from the T-$CO_2$ systems, at a specified operating state, the thermal efficiency for the R245fa ORC system with recuperator is always higher than that without recuperator.

### *4.2. Exergy efficiency analysis*

At a constant $CO_2$ expander inlet pressure (120 bar), the variations of exergy efficiencies with heat source and sink temperatures for the T-$CO_2$ system with and without recuperator are also calculated and depicted in Fig. 7. Similar to the thermal efficiencies, at a constant heat source temperature, the exergy efficiency increases with lower heat sink temperature for both cycles with and without recuperator. On the other hand, at a constant heat sink temperature, the effect of heat source temperature on the exergy efficiency is insignificant for the system with recuperator. Alternatively, when a recuperator is not installed, the system exergy efficiency decreases with higher hear source temperature. Therefore, considering of its effect on the thermal efficiency, the higher heat source temperature is more preferable for the T-$CO_2$ system with recuperator than the one without recuperator.

For the R245fa ORC as shown in Fig. 8, at a constant heat source temperature the exergy efficiency increases with lower heat sink temperature no matter if a recuperator is installed in the system. However, at a fixed heat sink temperature, the exergy efficiency decreases with higher heat source temperature irrespective of the recuperator installation. Therefore, it is understood that there is a compromise for the utilisation of high heat source temperature in a R245fa ORC system with recuperator. Alternatively, both thermal and exergy efficiencies can't be improved with higher heat source temperature for the system without recuperator.

At a constant heat sink temperature (10 °C), the variations of exergy efficiencies with heat source temperatures and $CO_2$ expander inlet pressures for the cycles with and without recuperator are predicted and shown in Fig. 9. For both cycles with and without recuperator, at a constant heat source temperature, the exergy efficiency increases firstly with higher $CO_2$ expander inlet pressure and then decreases. This indicates that there is an optimal $CO_2$ high side pressure where the exergy efficiency can be maximised for both cycles when heat source and sink conditions are fixed. However, at the same conditions, the optimal pressure for the cycle with recuperator is much less

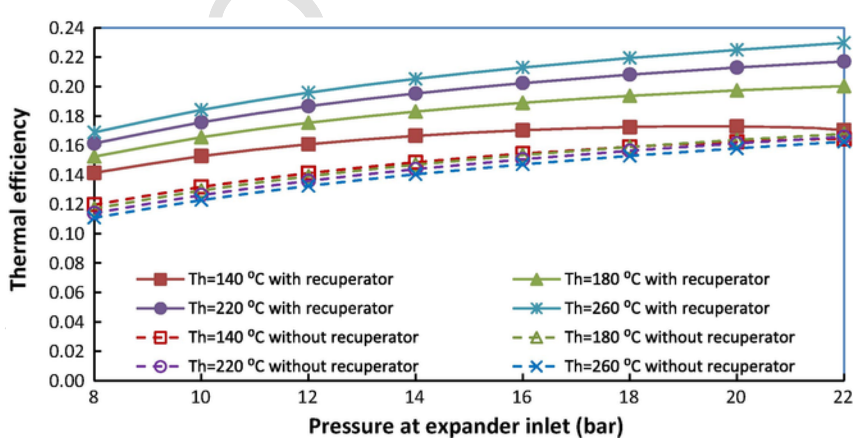



**Fig. 6.** Variations of thermal efficiencies with heat source temperatures and working fluid pressures at expander inlet for R245fa ORC.

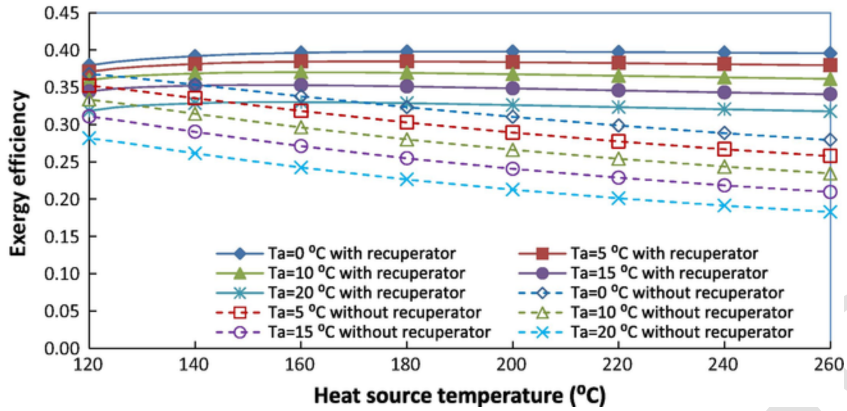



**Fig. 7.** Variations of exergy efficiencies with heat source and sink temperatures for T-$CO_2$.

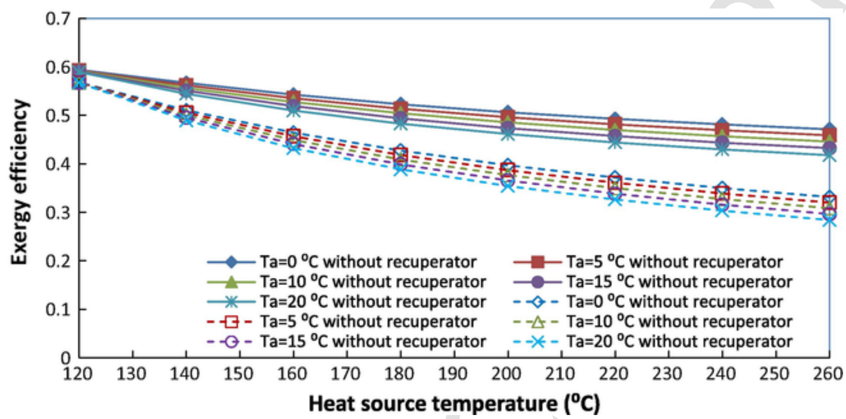



**Fig. 8.** Variations of exergy efficiencies with heat source and sink temperatures for R245fa ORC.

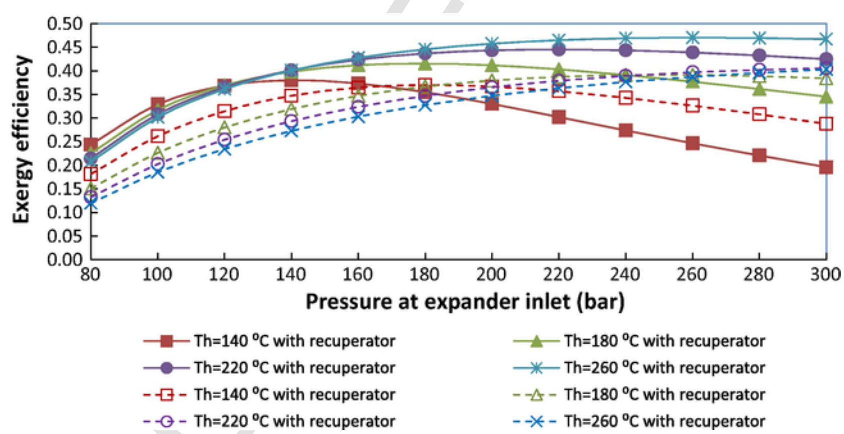



**Fig. 9.** Variations of exergy efficiencies with heat source temperatures and working fluid pressures at expander inlet for T-$CO_2$.

than that without recuperator. On the other hand, at a constant $CO_2$ high side pressure, the exergy efficiency decreases with higher heat source temperature if the $CO_2$ pressure is less than a specific value for each cycle (about 120 bar for the cycle with recuperator, much higher for the system without recuperator), otherwise will increase with higher heat source temperature. In addition, the exergy efficiencies for both cycles are higher for the system with recuperator if the $CO_2$ pressure is not too high.

The effect of expander inlet pressure on the exergy efficiency of R245fa ORC is a bit different, as shown in Fig. 10. At a constant heat sink temperature (10 °C), when the heat source temperature is fixed, the exergy efficiency for both cycles with and without recuperator increases mostly with higher R245fa pressure at expander inlet. Within the operating high side pressure range, the optimal pressure is only detected when the heat source temperature is not too high (140 °C) for the cycle with recuperator. In addition, at a fixed R245fa expander inlet pressure, the exergy efficiency decreases with higher

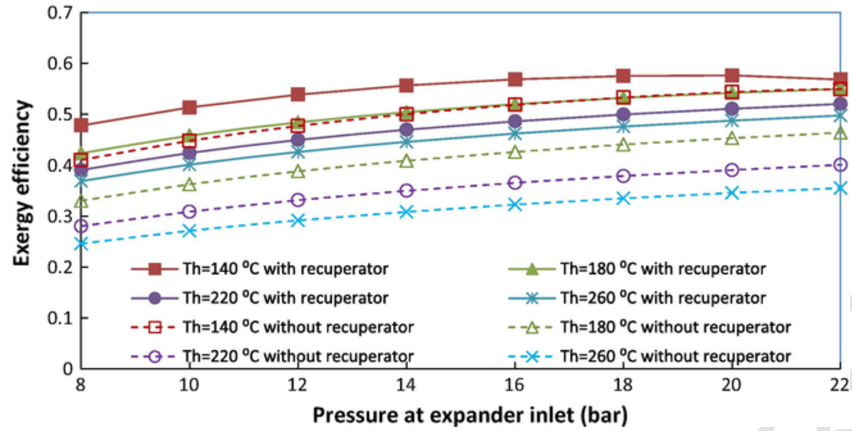



**Fig. 10.** Variations of exergy efficiencies with heat source temperatures and working fluid pressures at expander inlet for R245fa ORC.

heat source temperature. In general, at the same operating condition, the exergy efficiency of the cycle with recuperator is higher than that without recuperator.

The exergy destruction rate is defined as the exergy destruction of each component over the total system exergy destruction, which is helpful to evaluate and identify the system components with significant exergy destruction rates. At constant heat sink temperature (10 °C), constant expander inlet pressure (120 bar for T-$CO_2$ and 14 bar for R245fa ORC) and varied heat source temperature, these exergy destruction rates are therefore calculated and depicted in Figs. 11 and 12 respectively for T-$CO_2$ and R245fa ORC. For the T-$CO_2$, the exergy destructions of most components decrease with higher heat source temperature except for the recuperator for the cycle with recuperator and the condenser for the system without recuperator. In addition, at a specified operating state, for both T-$CO_2$ systems, the gas generator has the most exergy destruction rate while the liquid

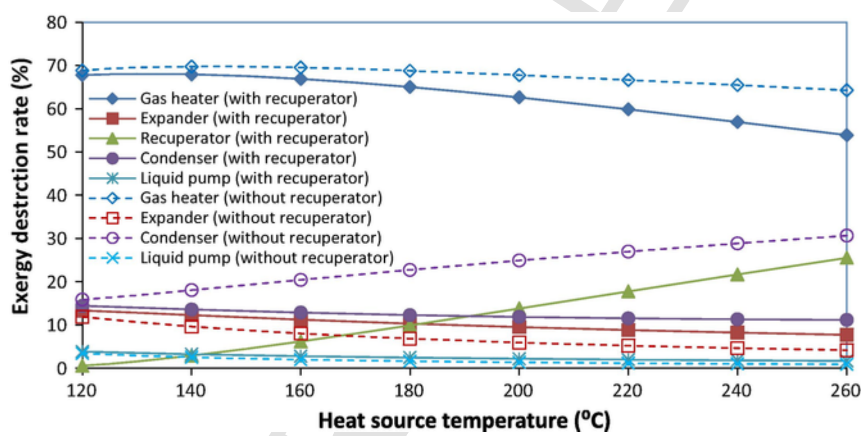



**Fig. 11.** Variations of component exergy destruction rates with heat source temperatures for T-$CO_2$.

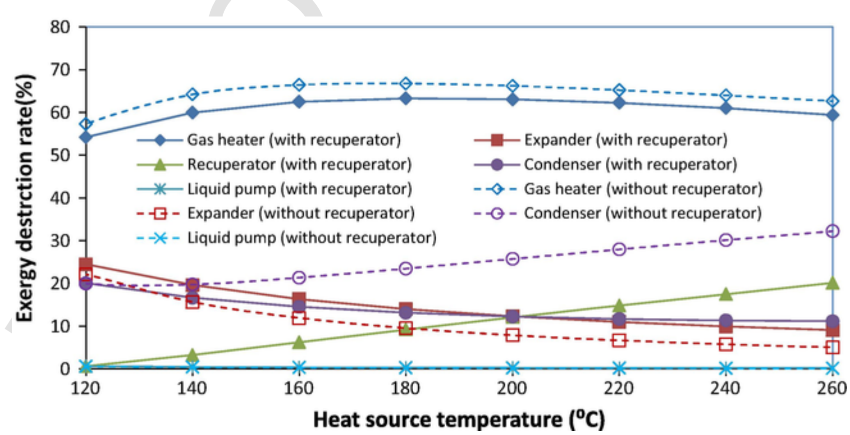



**Fig. 12.** Variations of component exergy destruction rates with heat source temperatures for R245fa ORC.

pump has the least exergy destruction rate. As shown in Fig. 12, although the magnitudes of component exergy destruction rates for the R245fa ORC are different from those in T-$CO_2$ cycles, the trends are quite similar. The component of evaporator has the maximum exergy destruction rate while the liquid pump has the least exergy destruction rate. Subsequently, more efforts are necessary to optimise the design of heat source heat exchangers for the power generation systems and cycles to maximise the component and system performance.

## 5. Conclusions

$CO_2$ and R245fa have both been acknowledged as applicable working fluids in low temperature (120–260 °C) power generation systems in terms of their thermophysical properties and safety data. However, $CO_2$ working fluid proves to be more promising due to its negligible global warming potential regardless of its high operating pressure. As to their application in low temperature power generation, $CO_2$ will inevitably work in supercritical power cycles such as T-$CO_2$ considering its low critical temperature. Meanwhile, R245fa will most likely be effective in an organic Rankine cycle (ORC) in view of its relatively high critical temperature. There are essential components required for both T-$CO_2$ and R245fa ORC systems and the installation of a recuperator is an option for each cycle. This study has comprehensively evaluated, compared and analysed the performances of these two power cycles with the following outcomes:

- For the T-$CO_2$ system, installing a recuperator is preferable and there is an optimal $CO_2$ expander inlet pressure for constant heat source and sink parameters where either thermal or exergy efficiency is maximised. Ideally, the system with recuperator can operate at a higher expander inlet pressure (>120 bar), higher heat source temperature and lower heat sink temperature.
- For the R245fa ORC system, installing a recuperator is also desirable. There is an optimal R245fa expander inlet pressure for constant heat source and sink parameters where either thermal or exergy efficiency is maximised but this is only available for lower heat source temperatures. The system with recuperator can preferably operate at a higher expander inlet pressure and lower heat sink temperature. There is a compromised selection for the heat source temperature considering its contrary effect on thermal and exergy efficiencies.
- From exergetic analysis for both T-$CO_2$ and R245fa ORC systems, the heat source heat exchangers have the most exergy destruction, closely followed by condensers, expanders and recuperators; this requires greater attention when optimising component designs and controls.
- Based on the assumptions in this paper, the thermal and exergy efficiencies of the T-$CO_2$ system are generally lower than those of the R245fa ORC. Further detailed heat transfer analysis and experimental investigation on the heat source and sink heat exchangers are necessary in the future to explore the potential of both systems. In addition, future work may include the utilisation of a mixture working fluid of $CO_2$ and one HFC in a low-grade power generation system to enhance the system performance and minimise the environmental impact.

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