David Tomás Sánchez Martínez
Lourdes García Rodríguez
(coordinadores)

Proceedings of the 7th International Seminar on ORC Power Systems

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Thermodynamic modification of CO_2-based combined cooling and power cycle with ejector

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ABSTRACT

CO_2-based combined cooling and power cycle (CCP) has received increasing attentions especially in scenarios with diversified energy desires for its excellent natural properties. In this paper, an ejector is introduced into a conventional CCP system to reduce exergy loss in refrigeration throttling. A refrigerated truck is selected as the typical application scenario. And the simulated results indicate that approximately 28% of the expansion work during throttling can be recovered by the ejector, concurrently saving 0.3kW and 0.7kW compression work with respect to 10.8kW and 6.5kW cooling load under the freezing and refrigeration conditions. To extend the application of CCP system in more general scenarios, the ejector based CCP system is further modified to suit single heat source conditions. Two modified ejector based CCP systems namely ME-CCP-I and ME-CCP-II are proposed to recover waste energy from the compressor discharge. The adaptations to various heat source and refrigerating temperature are comprehensively studied. And the results demonstrate the ME-CCP-II system will be more suitable for air-conditioning scenarios while the ME-CCP-I system is advanced in freezing scenarios. In addition, recovering waste energy from the compressor discharge contributes great to the total energy efficiency of the CCP system as well as the heat transfer in regeneration process. With the mass split ratio increased from 0.1 to 0.9, the total energy efficiency of the ME-CCP-I system can raise from 0.27 to 0.64 and the relative enhancement is ranging from 1.26% to 11.86% versus the baseline system. Meanwhile, the ME-CCP-I system can reduce 48.6% irreversible loss in the heat regeneration process versus the baseline one when the same amount of exhausted energy after expansion is recovered.

1 INTRODUCTION

With a global phase-down of hydrofluorocarbons (HFCs) after an October 2016 meeting of the parties to the Montreal Protocol(UNEP Ozone Secretariat., 2016), the substitute of high-GWP refrigerants has been the spotlight in energy and chemical industry. Carbon dioxide (CO_2), as one of the representative natural refrigerants and once widely used refrigerant in the early twentieth century (Lorentzen and Pettersen, 1993), has again arrested interests in the field due to its outstanding properties in both thermodynamics and ecology (Bianchi et al., 2021). Thereupon, CO_2 based power and refrigeration technologies have rapidly boomed, which is widely applied in solar thermal power system (White et al., 2021), electric vehicle heat pump (Song et al., 2022), supermarket refrigeration system (Dilshad et al., 2020) and etc.

In scenarios with multiple energy desires, the cogeneration systems are generally more attractive for its sufficient efficiency, include the combined heating and power (CHP) system, combined cooling and power (CCP) system, and combined cooling, heating and power (CCHP) system. In the last years, plenty of researchers have devoted to the theoretical constructions and modifications on CO_2-based CCP systems (Wang *et al*., 2012; Hou *et al*., 2018; Manjunath *et al*., 2018;Pan *et al*., 2020; Zhao *et al*., 2022). According to the coupled methods of the power subcycle and refrigeration subcycle, the CCP system can be simply classed into indirect type and direct type, wherein the latter directly shares a thermodynamic process of the two subcycles and thus more compact layouts with better exergoeconomic can achieve (Hou *et al*., 2018). In our previous studies (Zhang et al., 2022a, 2022b), the experimental prototype of direct type CO_2-based CCP system was developed and tested. The prototype integrated a transcritical CO_2 Rankine cycle and a reverse Carnot cycle (i.e. vapor compression refrigeration cycle, VRC) via the conjunct condenser and reservoir. The tested results indicate that switching the combined cycle from simultaneous cooling and power mode to power-alone mode or cooling-alone mode will either improve the power output or refrigeration output, concurrently ameliorating the effectiveness of heat transfer in the coupling condenser. Nevertheless, the VRC system with CO_2 commonly suffers lower efficiency due to the considerable exergy loss during throttling. In this context, ejector is approved an efficient solution to realize available work recovery in the throttling. Actually, approximately 8% to 27% improvement in COP could achieve in a transcritical ejector expansion refrigeration cycle (ERC), as reported by Banasiak *et al*.(2012), Elbel and Hrnjak (2008), Lawrence and Elbel (2019), Lee *et al*.(2014), Lucas and Koehler (2012), Minetto *et al*.(2013), Nakagawa *et al*.(2011).

As there is few attention paid to the direct type CCP system with ejector, the ejector is firstly introduced in the CCP system to enhanced COP of the conventional subcritical VRC subcycle in this paper. On the basis, another gap in the conventional CCP system is also considered, that is the available heat recovery from the refrigeration side. For typical CO_2 power cycles, the mismatch of heat capacity in the regenerator is the primary reason that results in the pinch point in heat transfer and performance deterioration (Luu et al., 2017). Shu *et al*.(2016) proposed a preheat-recuperative CO_2 Rankine cycle in engine waste heat recovery, wherein the engine coolant is employed as the preheat source. The crest span of specific heat capacity C_p of the CO_2 is accommodated well with the coolant and it simultaneously reforms the heat capacity allocation in the regenerator, which contributes to remarkable performance enhancement. In general scenarios with single heat source, waste heat from the compressor discharge in refrigeration subcycle is a possible preheat source. Hence, a further modification is conducted on the aforementioned ejector-assisted CCP system and two potential layouts derived from the scheme are investigated in the second case of the paper. Conclusively, the relevant results may offer some instructive references to future studies and applications of the CO_2-based CCP system.

2 METHODS AND MODELS 314

2.1 Description of CCP system

Fig.1 illustrated the schematic diagrams of the typical CCP system (Fig.1(a)) and its modified schemes with ejector(Fig.1(b-d)). The corresponding T-s diagrams are presented in Fig.2. In this paper, the basal CCP system (Figure.1(a), namely B-CCP) is firstly compared with the preliminarily modified CCP system with ejector (Figure.1(b), namely E-CCP). The power subcycle and refrigeration subcycle are integrated via the sharing condenser. And thus diverse outputs can be conveniently achieved by regulating the mass splitting in the condenser outlet. In the preliminary modification, the ejector is introduced to reduce the irreversible loss during throttling. Compared with the B-CCP system, the suction pressure of compressor in E-CCP system is quite enhanced so as to reduce the compression work. A refrigerated truck with two refrigeration requirements (i.e. refrigerating condition with evaporating temperature of 0°C and freezing condition with evaporating temperature of -20°C) is selected as the research objective, which is consistent with our previous work (Zhang et al., 2020). Both of the exhaust gas and the coolant from the engine are recovered to drive the CCP system. The refrigerated truck is equipped with a 246kW diesel engine and the refrigeration load is 10.8kW and 6.5kW in refrigerating condition and freezing condition, respectively. The specifications of the exhaust gas and coolant from the engine are listed as followings:

- Inlet temperature and mass flow rate of exhaust gas: $T_{eg,in}$ =470.7°C and m_g=1372.3kg/h.
- Inlet temperature, returned temperature, and mass flow rate of engine coolant: $T_{jw,in}$ =85°C, $T_{jw,re}$ =78.4°C, Q_{jw}=16.8m^3/h.

When turn to the common scenarios with single accessible heat source, the E-CCP system is further modified as ME-CCP-I system (Figure.1(c)) and ME-CCP-II system (Figure.1(d)). In this case, the outflow from pump is firstly preheated by compressor discharge. The particular difference is that in ME-CCP-II system, the back pressure of the expander is the suction pressure of compressor rather than the condensing pressure, which indicates the over-expansion in expander. In CO_2 transcritical power cycle, a proper over-expansion in expander is beneficial to thermal efficiency as suggested by Zhao et al.(2020). And therefore the scheme is also taken into consideration in this work to investigate its adaptation to the CCP system. Meanwhile, an additional precooler is needful to reduce the work consumption in compressor in the ME-CCP-II system.

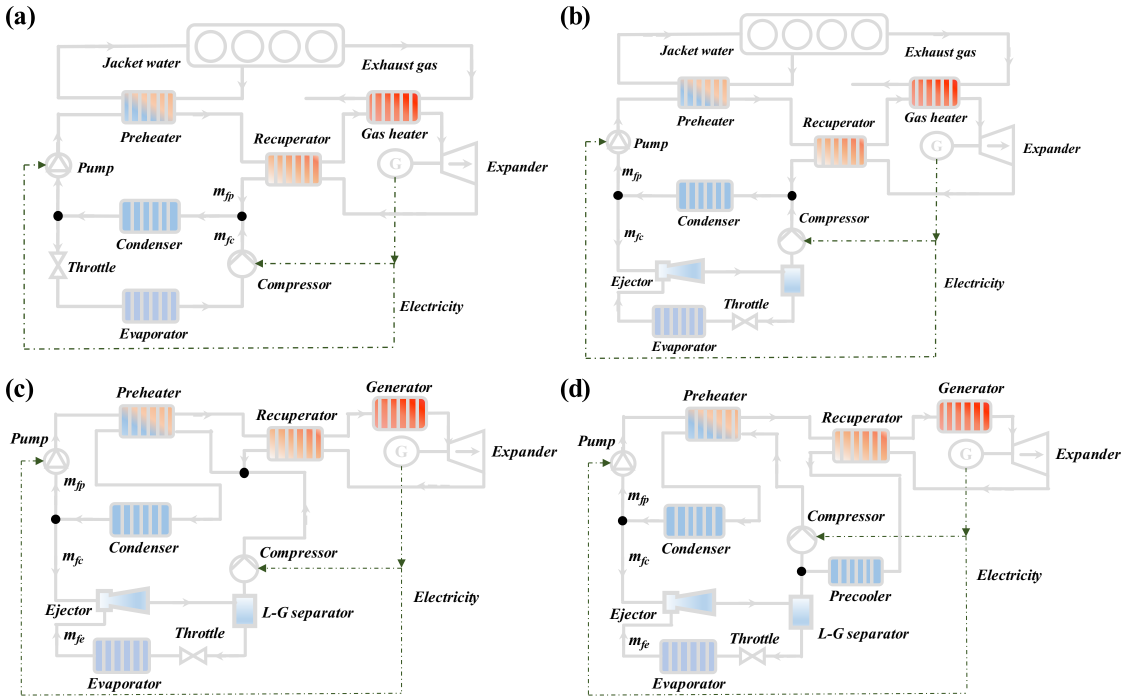

Figure.1: Schematic diagrams of the typical CCP system and the modified schemes with ejector: (a) Basal CCP system, namely B-CCP, (b) Ejector-assisted CCP system, namely E-CCP, (c) Modified ejector-assisted CCP system without over-expansion in power subcycle, namely ME-CCP-I, and (d) Modified ejector-assisted CCP system with over-expansion in power subcycle, namely ME-CCP-II.

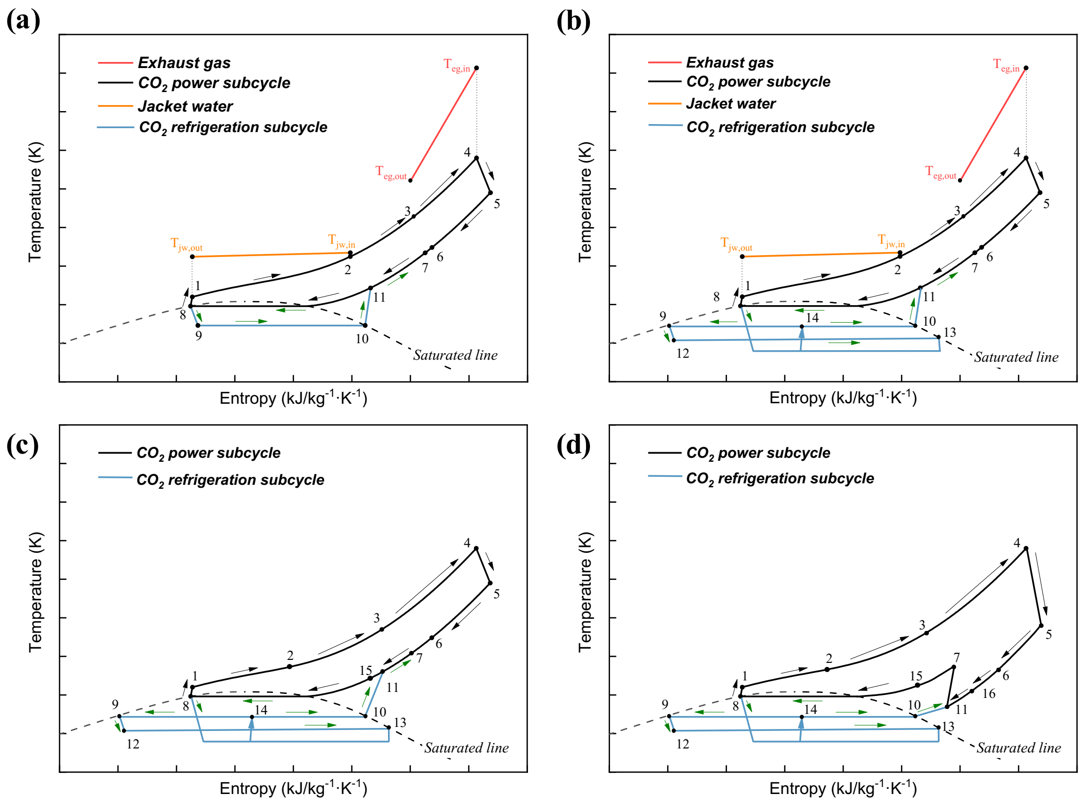

Figure.2: T-s diagrams of the typical CCP system and the modified schemes with ejector: (a) B-CCP system, (b) E-CCP system, (c) ME-CCP-I system, and (d) ME-CCP-II system.

2.2 Modeling of CCP system

To simulate the performance of the above systems, some necessary assumptions are made as followings:

- The systems perform in steady state condition, and pressure drop and heat loss in the components are negligible.
- Isentropic efficiencies are adopted to simulate the expander, compressor and pump, and the throttling process is regarded isenthalpic.
- The working fluids from evaporator outlet, condenser outlet and liquid-gas separator are saturated.
- Pinch temperature in the condenser, precooler, preheater, recuperator, and gas heater are settled at 5°C, 5°C, 5°C, 10°C and 30°C to guarantee efficient heat transfer.
- The condensing temperature should be less than 30°C to ensure condensing of the CO_2, which is set 25°C in this work.
- Inner flow in the ejector is steady and one-dimensional, and the internal irreversibility of ejector is described by nozzle efficiency ($\eta_n = 0.9$), mixing efficiency($\eta_m = 0.85$)and diffuser efficiency($\eta_d = 0.85$).

Based on the energy conversion and conservation, the quasi one-dimensional thermodynamic models of the CCP systems are built in MATLAB platform. The equations of each component and efficiencies of the machines are briefly summarized in **Table 1**. The properties of CO_2 are accessible in NIST database by invoking the REFPROP program. For the ejector, a constant-pressure mixing theory (Keenan *et al.*, 1950) is adopted to simulate the inner process in the two-phase ejector. An averaged relative discrepancy between the simulated and experimental results is 2.48% as conducted in our previous study (Zhang et al., 2020), that ensures the validation of the simulation.

In the first case with a refrigerated truck as objective, the pinch point temperature difference method is applied to acquire the mass flow in power subcycle (Shu et al., 2016), while the mass flow in refrigeration subcycle is determined by the refrigeration load. While in the second case, the heat source is turned to common scenarios, where the temperature of heat source is an independent variable and its impact on system performance is one of the discussed scope. To simplify the simulation, the inlet temperature of the expander is used to reflect heat source conditions and the mass flow rate in the power subcycle is allocated as 1kg/s, i.e. $m_{fp} = 1\text{kg/s}$. However, the mass flow in refrigeration subcycle is still depend on the refrigeration load. The mass split ratio in condenser outlet is defined as $SR = m_{fc}/m_{fp}$, and the mass flow rate of working fluid running through the evaporator can therewith be calculated by entrainment ratio of the ejector: $m_{fe} = m_{fc} \cdot \mu$. As there are diverse energy outputs in the CCP system, the net power output ($W_{net} = W_{exp} - W_{pump} - W_{com}$) with same refrigeration capacity and the total energy efficiency of the CCP system (defined $\eta_{en} = \frac{Q_{eva}+W_{net}}{Q_{absorb}}$) are used to evaluate the performance of different schemes. All of E-CCP system, ME-CCPI, and ME-CCPII systems are allocated with the identical SR to ensure the same refrigeration load. And its impact on systematic performance is also taken into account. 316

On the other hand, to investigate the potential amelioration on heat regeneration process in these systems, the exergy analysis in heat transfer is also conducted. The exergy of working fluid at the specific state point is calculated by Equation (1), wherein the T_0, h_0, s_0 is the referenced environmental temperature, enthalpy and entropy at thermodynamic equilibrium state. The EUD (energy-utilization diagram) methodology (Jin et al., 2004) is adopted to graphically present the endogenous exergy destruction in the corresponding section based on *A* (energy level) and *ΔH* (energy transformation quantity) diagram. Herein, the energy level *A* is defined by Equation (2), and the ΔE_x and ΔS represent the exergy change and the entropy change, respectively. For the specific energy transformation process, the energy and exergy of the donor and receptor can be signified by paired curves. And the area between the curves become equal to the exergy destruction during the process.

$$ex_i = h_i - h_0 - T_0(s_i - s_0) \tag{1}$$

$$A = \frac{\Delta E_x}{\Delta H} = 1 - \frac{T_0 \Delta S}{\Delta H} \tag{2}$$

Table 1: Energy balance equations of the key components

Component	**Equation**
Expander	$W_{exp} = (h_4 - h_{5,s})\eta_{exp}$, η_{exp}=0.8 (Hou et al., 2018)
Gas heater/Generator	$Q_{gh} = m_{fp}(h_4 - h_3) = m_g(h_{g,in} - h_{g,out})$
Recuperator	$Q_{re} = m_{fp}(h_3 - h_2) = m_{fp}(h_5 - h_6)$
Preheater	$Q_{pre} = m_{fp}(h_2 - h_1) = m_{jw}(h_{jw,in} - h_{jw,out})$ or $Q_{pre} = m_{fp}(h_2 - h_1) = (m_{fp} + m_{fc})(h_7 - h_{15})$
Precooler	$Q_{prec} = m_{fp}(h_5 - h_{16})$
Condenser	$Q_{cond} = (m_{fp} + m_{fc})(h_{11} - h_8)$ or $Q_{cond} = (m_{fp} + m_{fc})(h_{15} - h_8)$
Evaporator	$Q_{eva} = m_{fc}(h_{10} - h_9)$ or $Q_{eva} = m_{fe}(h_{13} - h_{12})$
Pump	$W_{pump} = m_{fp}(h_{1,s} - h_8)/\eta_{pump}$, η_{pump} = 0.75 (Zhao *et al.*2020)
Compressor	$W_{com} = m_{fc}(h_{11,s} - h_{10})/\eta_{com}$ or $W_{com} = m_{fc}(h_{7,s} - h_{11})/\eta_{com}$; η_{com}=1.003-0.121·P_r (Liao et al., 2000)
Throttle	$m_{fc}h_8 = m_{fe}h_9$ or $m_{fe}h_9 = m_{fe}h_{12}$
LG-separator	$(m_{fc} + m_{fe})h_{14} = m_{fc}h_{10} + m_{fe}h_9$
Ejector	$m_{fc}h_8 + m_{fe}h_{13} = (m_{fc} + m_{fe})h_{14}$, entrainment ratio $\mu = {}^{m_{fe}}/_{m_{fc}}$

3 RESULTS AND DISCUSSIONS

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3.1 Case 1: Effectiveness verification of ejector in CCP system (B-CCP vs. E-CCP)

The simulated results of the B-CCP and E-CCP systems are listed in Table 2. To achieve the maximum net power, the inlet temperature and pressure of expander are optimized with PSO algorithm. Obviously, the refrigeration COP (which is defined as: Q_{eva}/W_{com})is definitely enhanced with the help of the ejector, which is improved by 29.2% and 15.9% under freezing and refrigeration condition, respectively. The performance of the ejector is represented by the ejector efficiency η_{eje}, which is calculated by Equation (3), defined as the ratio of the recovered expansion work by the ejector to the maximum potential expansion work(Elbel and Hrnjak, 2008). Herein, approximately 28% of the expansion work during throttling can be recovered by the ejector, subsequently the compressor ratio is reduced by 14.7% and 7.0%. Although the absolute enhancement in net power output is just 0.7kW and 0.3kW under the two conditions, it accounts for 20.6% and 14.3% of the compression work. Hence, it indicates sufficient potential in energy saving with a larger refrigeration load. Furthermore, according to the results, the ejector will perform better in scenarios with lower evaporating temperature because of the lager pressure difference in throttle.

$$\eta_{eje} = \mu\frac{(h(P_{LG}, s_{sf}) - h_{sf})}{h_{pf} - h(P_{LG}, s_{pf})} \tag{3}$$

Table 2: Performance index of B-CCP and E-CCP system

	Freezing condition			**Refrigeration condition**		
	B-CCP	E-CCP	RV(%)	B-CCP	E-CCP	RV(%)
Net power output (kW)	31.6	32.4	+2.5	32.9	33.2	+0.9
Refrigeration COP (-)	1.92	2.48	+29.2	5.21	6.04	+15.9
Compressor work(kW)	3.4	2.7	-20.6	2.1	1.8	-14.3
Compressor ratio(-)	3.27	2.79	-14.7	1.85	1.72	-7.0
Ejector efficiency η_{eje}(%)	/	28.6	/	/	28.8	/

3.2 Case 2: Adaptation analysis of ME-CCP-I and ME-CCP-II

In this section, the original scheme without compressor discharge heat recovery (i.e. E-CCP system without preheater) is included as the baseline system. The performance index of the three CCP schemes with respect to different expander inlet temperature and evaporating temperature is firstly discussed. For consideration with the refrigeration span, an evaporating temperature from -20°C to 10°C is appropriate for freezing, refrigeration and air conditioning. While for the expander inlet temperature, T_g from 200°C to 300°C is possible to cover most of the heat source like solar energy, bioenergy, and industrial waste heat, etc. The expander inlet pressure of each system is optimized to achieve the maximal W_{net} and it is restricted within 20MPa in account of the state of the art . As for the refrigeration subcycle, the mass split ratio is set as 0.2 in the baseline condition.

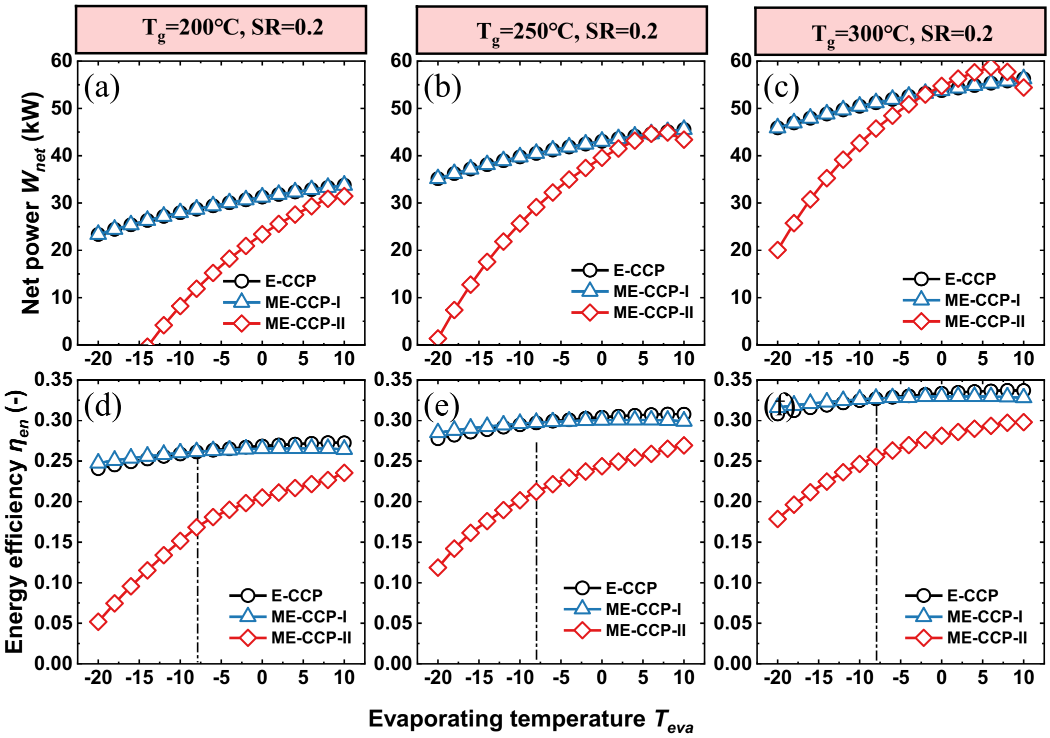

Figure 3: Performance comparison with respect to various heat source and evaporating temperature.

As is can be seen in Figure 3, the E-CCP system and ME-CCP-I system demonstrate identical changes in net power with respect to evaporating temperature and it has no concern with the heat source temperature. Overall, the net power output in these two system performs much better than the ME-CCP-II system when the expander inlet temperature is equal to 200°C and 250°C. As for the ME-CCP-II system, an extra overexpansion ratio imposes great energy consumption in the compressor and thus it will be more suitable for air conditioning scenarios. Simultaneously, a higher grade heat source makes greater sense to the ME-CCP-II system as well. As presented in Figure 3(c), the ME-CCP-II system excels the ME-CCP-I with evaporating temperature ranging from 0°C to 8°C at T_g=300°C condition. For instance, when the evaporating temperature is equal to 6°C, the ME-CCP-II can offer 58.7kW mechanical power output with 30.0kW refrigeration load, which is improved by 6.15% versus the ME-CCP-I and E-CCP systems (which is 55.3kW in the condition). However, the ME-CCP-II system has a poor performance on total energy efficiency in all of the simulated conditions, which is resulted from the reduced heat regeneration in recuperator.

For the ME-CCP-I system, only if the evaporating temperature is lower than -8°C, the ME-CCP-I system will present better energy efficiency than the baseline system. Hence, it will be more suitable for freezing scenarios. This is predictable because once the temperature of compressor discharge is higher than the pumped outflow will the preheater work. Meanwhile, the enhancement is also related to the mass split ratio, that is the refrigeration load. Figure 4 presents the variations of total energy efficiency of the ME-CCP-I and baseline E-CCP system with respect to varying mass split ratio when 318

T_g=300°C and T_{eva}=-20°C. With the mass split ratio increased from o.1 to 0.9, the total energy efficiency of the ME-CCP-I system can raise from 0.27 to 0.64 and the relative enhancement is ranging from 1.26% to 11.86% versus the E-CCP system.

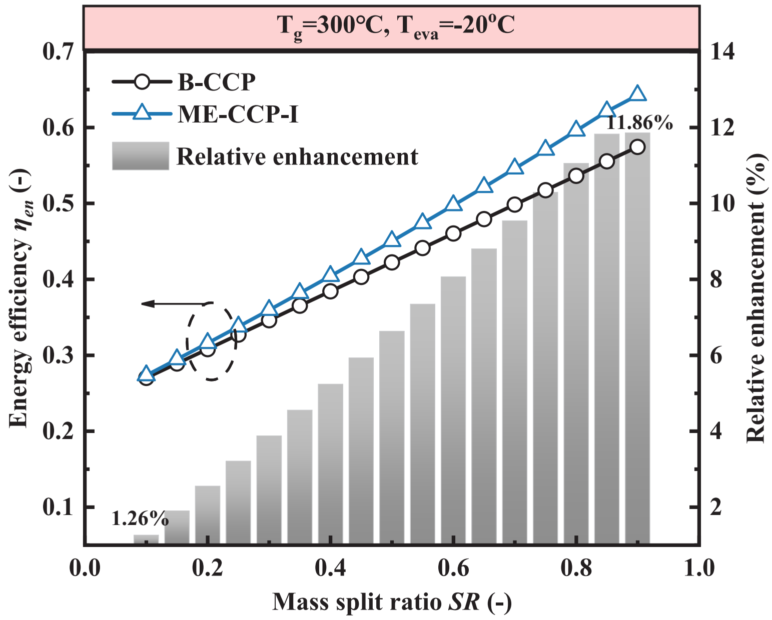

Figure 4: Variation of energy efficiency with respect to mass split ratio: T_g=300°C and T_{eva}=-20°C.

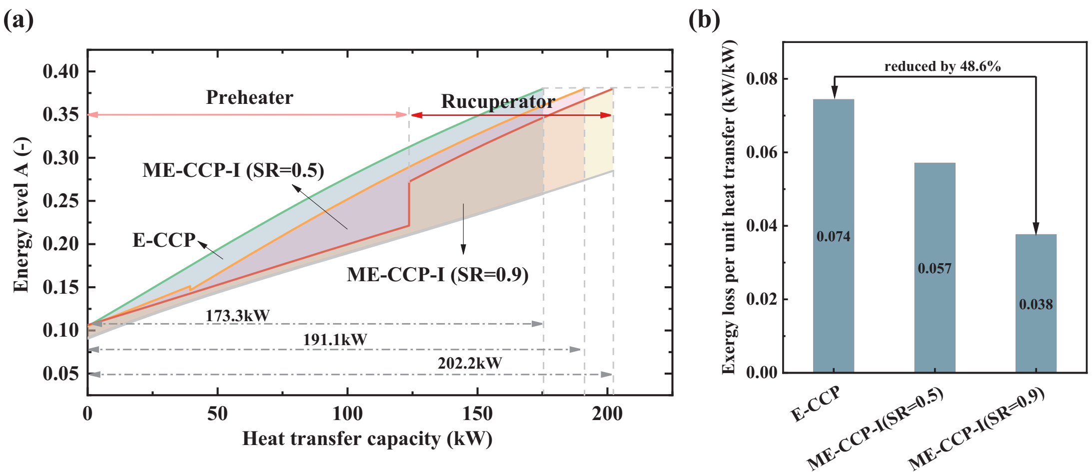

Figure 5: Heat transfer and exergy destruction in preheater and recuperator : T_g=300°C and T_{eva}=-20°C.

Figure 5 presents the energy utilization diagram in the preheater and recuperator of the ME-CCP-I system and E-CCP system with different mass split ratio. Therein, the grey solid curve represents the low-temperature fluid between the pump outlet and generator inlet, while the green, orange and red curves indicate the high-temperature fluid that performs as the regenerating heat source, with *SR* equal to 0, 0.5, and 0.9 respectively. The shaded area between the two heat transfer fluids signifies the exergy destruction during the heat transfer process and the horizontal axis indicates the actual heat transfer capacity in the exchanger. Obviously, preheat in the preheater can definitely ameliorate temperature match during the heat regeneration process, and improve the ultimate temperature of the high-pressure fluid at the recuperator outlet. As there is no preheater in the baseline E-CCP system, it can be equivalent to the condition with SR=0 in the ME-CCP-I system. Therefore, with the SR increased from 0 to 0.9, the available heat regeneration is enhanced from 173.3kW to 202.2kW, and the exergy loss during the regenerating process is simultaneously reduced from 13.0kW to 7.61kW. To concurrently reveal the variations of exergy loss and heat transfer capacity in the case, a dimensionless exergy loss over per unit heat transfer capacity is presented in Figure 5(b), which is calculated by Equation (4). For the E-CCP system, a high $ex_{d,Q}$ with 0.074 is obtained without heat recovery from the compressor discharge. While for the ME-CCP-I system, the values are just 0.057 and 0.038 when the SR is equal to 0.5 and 0.9, respectively. Hence, ME-CCP-I system can reduce 48.6% irreversible loss in the heat regeneration process versus the E-CCP system when the same amount of exhausted energy after expansion is recovered.

$$ex_{d,Q} = \frac{Ex_{d,tot}}{Q_{tot}} \quad (4)$$

4 CONCLUSIONS

In this paper, a thermodynamic modification on CO_2-based combined cooling and power cycle with ejector is conducted. A refrigerated truck is firstly selected as the typical application scenario to verify the effectiveness of the ejector and then further modification on the baseline ejector-based CCP system is performed to extend its application to more general scenarios with single heat source. The impacts of heat source, refrigerating temperature, and mass split ratio on adaptation of the modified systems are comprehensively investigated. Conclusively, the relevant results may offer some instructive references to future studies and applications of the CO_2-based CCP system, and the main conclusions are listed as followings:

- Approximately 28% of the expansion work during throttling can be recovered by the ejector, concurrently saving 0.3kW and 0.7kW compression work with respect to 10.8kW and 6.5kW cooling load under the freezing and refrigeration conditions.
- The ME-CCP-II system will be more suitable for air-conditioning scenarios while the ME-CCP-I system is advanced in freezing scenarios.
- Recovering waste energy from the compressor discharge contributes great to the total energy efficiency of the CCP system as well as the heat transfer in regeneration process. With the mass split ratio increased from 0.1 to 0.9, the total energy efficiency of the ME-CCP-I system can raise from 0.27 to 0.64 and the relative enhancement is ranging from 1.26% to 11.86% versus the baseline system. Moreover, the ME-CCP-I system can reduce 48.6% irreversible loss in the heat regeneration process versus the baseline system when the same amount of exhausted energy after expansion is recovered.

NOMENCLATURE

CCP	Combine cooling and power cycle
COP	Coefficient of performance
RV	Relative variation
h	Specific enthalpy
ex	Specific exergy
s	Specific entropy

Subscript

com	Compressor
cond	Condenser
d	Destruction
eje	Ejector
eva	Evaporating
exp	Expander
fc	Mass flow rate in compressor
fe	Mass flow rate in refrigeration side
fp	Mass flow rate in power side
gh	Gas heater or generator
i	Specific state point in the cycle
in	Inlet

jw	Jacket water
out	Outlet
pre	Preheater
pump	Pump
tot	Total

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The ORC conference, organized biennially, stems as the only conference that is specific to ORC technology, therefore gathering a diverse community whose affiliation spans across all the interested stakeholders, not only in this particular technology but also and in a broader context, in the energy transition. Original equipment manufacturers, professional associations, end-users, investors, policy makers, academics, scientists feel at home at ORC 2023.

The almost 100 proceedings in this book cover a wide variety of topics, from fundamentals to system integration through component design, accounting for thermodynamic performance as well as component design. In addition to this, and as a new track in 2023, works on heat pump technology were also accepted in order to raise awareness of the strong ties between both technologies, specifically in energy storage applications.

This book provides an excellent overview of the current maturity of power systems based on Organic Rankine Cycle technology for applications as diverse as geothermal and waste heat recovery in industry or downstream of other prime movers (e.g., marine applications). It is also an excellent source of information to understand the current challenges faced by the technology, stemming from a very competitive market and increasingly stringent environmental regulations.

The organizers of ORC 2023 hope that the reader finds this work as exciting as the attendees to the conference and, maybe, make the decision to join the 8th edition to the conference in 2025.

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