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(coordinadores)

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A novel combined cooling and power cycle integrated ejector refrigeration and composition adjustment for stationary engine waste heat recovery

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ABSTRACT

The combined cooling and power cycle has wide applications ascribed to the feasibility of simultaneously providing cooling and electricity. The layout comprised of vapor compression refrigeration cycle and Organic Rankine Cycle by sharing condenser is one of the most common structures. However, for this kind of structure, the high temperatures of working fluid in compressor outlet and expander outlet aggravate the condensation load and lead to the decline of system performance. This study proposes a novel combined cooling and power cycle integrated of ejector refrigeration and composition adjustment running with CO_2 mixture. Liquid separation condenser is introduced to realize composition adjustment. Working fluid with higher CO_2 mass fraction flows to power sub-cycle and evaporation process of refrigeration sub-cycle. On the other hand, working fluid with smaller CO_2 mass fraction is pumped to the inlet of ejector and heated by expansion exhaust. A stationary engine is chosen as the research objective. The co-generation system recovers waste heat of engine exhaust and coolant to provide cooling to refrigerating chamber. The results show the priority of the novel proposed co-generation system. Under the requirement of 0 °C chilled fluid, the basic combined cooling and power cycle could export 8 kW cooling capacity and 6.63 kW electricity, while the novel proposed system can export 4.5% more net power output with 8 kW cooling capacity output or export 18.6% higher cooling capacity with 6.63 kW electricity.

1 INTRODUCTION

Combined cooling and power cycle (CCP) has wide applications in energy conversion and utilizing processes, such as industry, distributed energy systems, and transportation, ascribed to the feasibility of simultaneously providing cooling and electricity. Thus, enhancing the performance of CCP is an effective approach to realize the target of peaking carbon dioxide emission and carbon neutrality.

The basic CCP is comprised of an Organic Rankine Cycle (ORC) and a Vapor Compression Cycle (VCC), as depicted in figure 1(a). Liang *et al.* (2021) established a lab-scale prototype of separated CCP, R245fa and R134a were selected as working fluids for ORC and VCC, experimental results showed that the system could produce 1.8 kW cooling capacity at -4 ºC with 95 ºC water as heat source and the overall heat-to-cooling efficiency was 0.18.

To improve the compactness and flexibility of CCP, coupled type is analyzed, as presented in figure 1(b). Zhang *et al.* (2022) built a CO_2-based CCP prototype to recover ICE waste heat, the prototype could operate in three modes and the test results revealed that the prototype could export 5.8 kW power at power-alone mode, 3.2 kW power and 8.0 kW refrigeration at co-generation mode as well as 9.8 kW refrigeration at cooling-alone mode. To further improve the performance of CCP, explorations focusing on structure optimization and working fluid upgrading are carried out. Ejector is introduced to decrease the irreversible loss of throttling process in VCC. Mondal and De (2017) established an ejector-based organic flash CCP for low-grade waste heat recovery and a compressor was installed between

evaporator and ejector to match the elevated condenser pressure. On the other hand, zeotropic mixture possesses the phenomenon of temperature glide in two-phase heat exchanging process due to the characteristics of vapor-liquid equilibrium, leading to better thermal match and less exergy destruction in heat exchangers. Yao *et al.* (2022) investigated the performance of coupled CCP running with a series of CO_2-based mixtures for ICE waste heat recovery, results showed that CO_2/R32 could maximally achieve 19% power increase or 80% cooling increase compared to CCP running with pure CO_2.
However, a fixed composition of mixture for coupled CCP ($x_P = x_R = x_T$), as figure 1(b) exhibits, is usually unsuitable since the operating temperature intervals of CCP are large and the features of sub-cycles are disparate. Thus, composition-adjustable CCP is proposed to realize a further enhancement of CCP performance, as figure 1(c) presents. Barkhordarian *et al*. (2017) proposed an ammonia-water CCP system integrated with ejector and rectifier, due to the adoption of rectifier, rich ammonia fluid entered the refrigeration sub-cycle and weak ammonia fluid flowed to power sub-cycle. Zhu *et al.* (2021) designed a combined cooling, heating and power cycle using R141b/R134a, refrigeration sub-cycle operated with rich volatile fluid and power sub-cycle operated with weak volatile fluid were achieved.

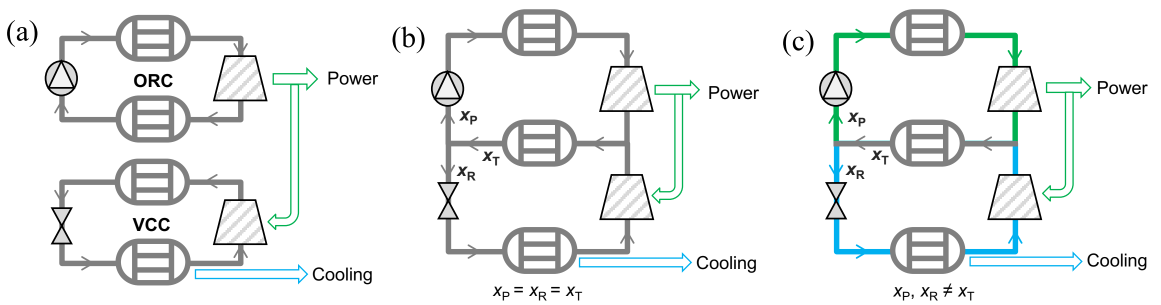

Figure 1: CCP layouts: (a) separated type; (b) coupled type; (c) composition-adjustable type.

Based on the previous discussion, ejector refrigeration and composition adjustment could both enhance 72
the performance of CCP. From the perspective of ‘mass’, the above two methods are respectively corresponding to the converging and separation of streams. From the perspective of ‘energy’, ejector refrigeration requires heat to drive while composition adjustment prefers smaller condensation load to maintain the thermal match in condenser (Sun *et al*. 2021). However, the deep coupling of the above two methods for CCP has not been discussed. In this study, a novel CCP integrated of ejector refrigeration and composition adjustment is proposed and a detailed case study is carried out for stationary ICE power station waste heat recovery.

2 SYSTEM DESCRIPTION

In this section, detailed structures of the basic coupled CCP and the novel proposed improved CCP for ICE dual heat sources recovery are described. Then, the suitable zeotropic mixture for CCP is discussed. At last, the case study of CCP for stationary ICE power station waste heat recovery is implemented.

2.1 Structure of the basic coupled CCP

Figure 2(a) depicts the schematic of the basic coupled CCP proposed by Yao *et al.* (2022) and the corresponding *T-s* diagram is illustrated in figure 2(b). As shown in figure 2, liquid working fluid is obtained after condenser (11-1) and divided into two streams. One flows to power sub-cycle, subsequently, this stream is pumped to high pressure (1-2) and respectively heated by preheater (2-3), regenerator (3-6) and gas heater (6-4). Then, high temperature and pressure steam expands in turbine (4-5). The other stream does isenthalpic throttling first (1-8) and cooled the chilled fluid in evaporator (8-9). Next, it is compressed to condensation pressure (9-10) and mixed with another stream (16,10-11). Then, the converged stream is cooled in condenser (11-1). A whole circulation process completes. The advantages of the basic coupled CCP are:

- Regenerator improves the inlet temperature of turbine and reduces the condensation load.

- Zeotropic mixture possesses the superiority of CO_2 and R32, it could realize effective heat utilization of engine exhaust and coolant (CO_2) and reduce condenser intel temperature (R32).

However, there are still serval problems that need to solve:

- Heat energy of engine exhaust and coolant could not be full recovery simultaneously.
- Condenser intel temperature is still high.
- Compression work of refrigeration sub-cycle is large.

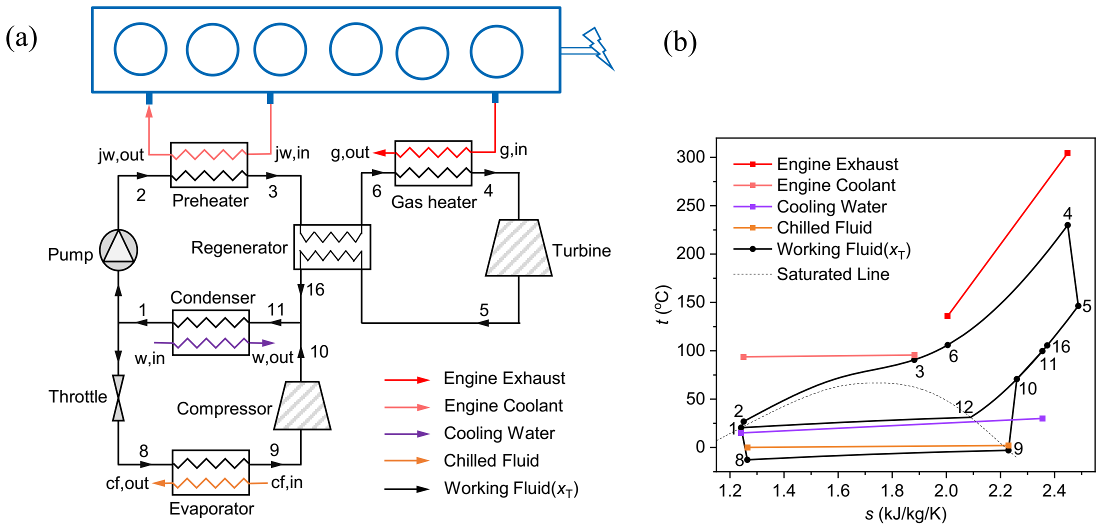

Figure 2: (a) Schematic and (b) *t-s* diagram of the basic coupled CCP.

2.2 Structure of the improved CCP

To solve the deficiency of basic coupled CCP, a novel improved CCP is proposed with the introduction of ejector refrigeration and composition adjustment, as shown in figure 3(a). The corresponding *T-s* diagram is illustrated in figure 3(b) and the four subgraphs are respectively corresponding to four compositions operating in the whole circulation process. As shown in figure 3, working fluid with composition of x_T is condensed to two-phase state in the first half of liquid separated condenser (LSC) (16-17) and separates into two streams (17-8,18). Then, working fluid with weak volatile constituent (x_1) is directly pumped to ejector (8-9), while working fluid with rich volatile constituent (x_2) is further condensed in the second half of LSC (18-1) and divides into two streams. One stream (x_2) is pumped to high pressure (1-2) and heated to high temperature by preheater (2-3) and gas heater (3-4). Then, the high temperature and pressure steam expands in turbine (4-5). The other stream (x_2) depressurizes in throttle (1-11) and exports cooling capacity in evaporator (11-12). To realize the flexible operation of ejector, an auxiliary compressor is set between evaporator and ejector (12-13) (Mondal and De 2017). After being pressured in pump2 (8-9), working fluid (x_1) divides into two streams. One stream is used to recover the heat at the outlet of ejector (R1, 9-91) and expanding exhaust (R2, 91-102), the other is used to fully utilize the unused heat of coolant (R3, 9-101). Then, these two streams converge into one flow (101,102-10) and ejects compressor exhaust (10,13-14). On the other hand, the exhaust of turbine (x_2) is cooled by another working fluid (x_1) (R2, 5-6) and joins in another stream (x_3) (6,14-15). At last, the converged stream is further cooled in R1 (15-16) and a whole circulation process completes. 73

The improvement of the improved CCP are:

- **Full utilization of dual waste heat sources**: regenerator is removed since the regenerative heat will improve the inlet temperature of working fluid in gas heater (3) and result in the insufficient utilization of engine exhaust. In addition, the residual heat energy of engine coolant is fully utilized to heat the primary flow of ejector in R3.
- **Better thermal match in condensation process**: the partial condensation load is utilized to preheat the primary flow of ejector (R1, R2), leading to a better thermal match in condensation process, as presented in figure 3(b).

- **Various compositions for different thermodynamic stages**: LSC is introduced to achieve composition adjustment, which could realize working fluid with rich volatile constituent operated in power sub-cycle and the evaporation process of refrigeration sub-cycle (higher evaporation pressure) as well as working fluid with weak volatile constituent operated in ejector of refrigeration sub-cycle.

It should be noted that the present proposed structure is a specific construction for ICE waste heat recovery. For a more universal application scene, similarly, an improved CCP could be designed by the method that realizes various compositions operated in different thermodynamic stages and using condensation load and unutilized heat of source to heat ejector.

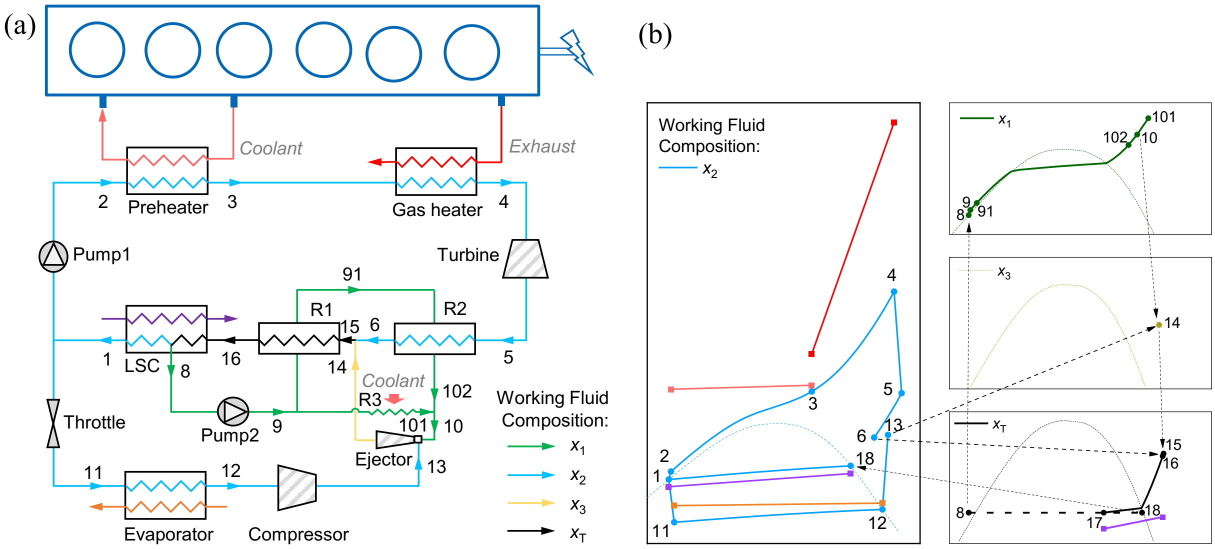

Figure 3: (a) Schematic and (b) *t-s* diagram of the improved CCP.

2.3 Case study

For an isolated factory, such as a foodstuff factory, as shown in figure 4(a), it usually needs ICE power station to produce electricity. Part of the electricity is used to drive the equipment and the other is used to drive the refrigeration unit to export cooling to refrigerating chamber. The huge energy contained in engine exhaust and engine coolant is directly discharged to ambience and wasted. To enhance the comprehensive performance of ICE, the coupled CCP and the improved CCP discussed above are applied, as figure 4(b) presents. On the one hand, the ICE waste heat is recovered to produce cooling and electricity. On the other hand, the supplied cooling by CCP could reduce the electricity consumed in refrigeration unit. To evaluate the performance of the new proposed CCP, a diesel ICE is selected as objective (Sun *et al.* 2021). The boundary conditions and operating parameters of CCP are enumerated in Table 1.

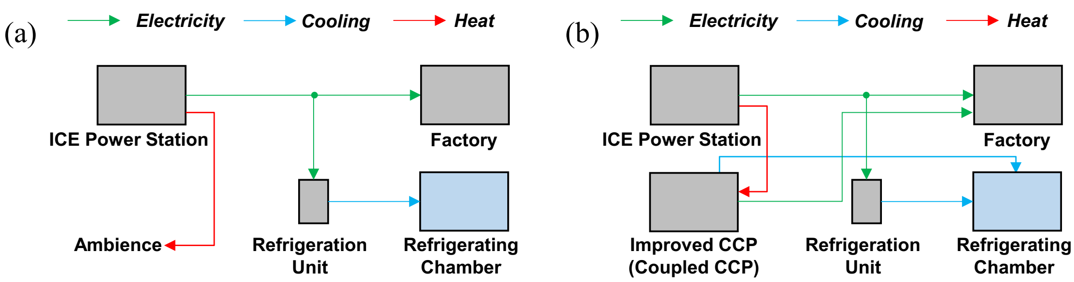

Figure 4: (a) The original isolated system; (b) The improved system combined with CCP.

Table 1: Boundary conditions and operating parameters of CCP

Term	Value
Heat sink	
Exhaust mass flow rate	0.164 kg/s
Exhaust temperature	304.4 °C
Coolant mass flow rate	4.25 kg/s
Coolant temperature	95.6-92.6 °C
Heat sink	
Cooling water inlet temperature	15 °C
Cooling water temperature rise	15 °C
Refrigeration conditions	
Cooling capacity	8 kW
Chilled fluid temperature	2-0 °C
Constraint conditions	
Pinch point of preheater, R3, LSC, evaporator	5 °C
Pinch point of R1, R2	15 °C
Pinch point of gas heater	30 °C
Pump isentropic efficiency	0.7
Turbine isentropic efficiency	0.8
Compressor isentropic efficiency	0.8
Mechanical efficiency	0.98
Generator efficiency	0.98

2.4 Fluid Selection

For ICE dual heat sources, CO_2 could realize good thermal match in heating process due to the character of specific heat in supercritical region (Yao *et al.* 2022). Therefore, CO_2 is usually selected as the optimal working fluid for ICE waste heat recovery. However, CO_2 is hard to be condensed under ambient temperature since the critical temperature is only 31 °C. In addition, the efficiency of heat to power for pure CO_2 is small. Therefore, CO_2-based mixture is proposed as the substitution of pure CO_2. Yao *et al.* (2022) investigated the performance of coupled CCP running with a series of CO_2-based mixtures for ICE waste heat recovery, results showed that CO_2/R32 could realize the best performance compared with pure CO_2, CO_2/R161, CO_2/R152a, CO_2/R134a, CO_2/R1234yf and CO_2/propane. Thus, in this study, CO_2/R32 is selected as working fluid. The properties of CO_2 and R32 are enumerated in Table 2.

Besides the remarkable thermodynamic performance, CO_2/R32 possesses excellent physical properties:

- **Environmental-friendly**: more proportion of CO_2, lower GWP of the mixture.
- **Safe**: CO_2 could efficiently cut down the flammability of refrigerants (Shi *et al*. 2021).
- **Thermostable**: the pyrolysis temperature of pure R32 and CO_2/R32 (mol 0.3/0.7) is more than 270 °C (Xin et al. 2022) while the wall temperature is around 260 °C according to heat transfer.

Table 2: Physical, environmental, and safety data of CO_2 and R32.

Substance	Molecular mass	t_b	t_c	P_c	ODP	DWP	Safety date
	g/mol	°C	°C	MPa			ASHRAE 34 Safety group
CO_2	44.01	-78.4	31	7.38	0	1	A1
R32	52.02	-51.7	78.1	5.78	0	716	A2L

3 METHODOLOGY

For simplicity of calculation, the following assumptions are made:

- The improved CCP is operated under steady state.
- Pressure drop and heat transfer destruction in pipes and heat exchangers are neglected.
- Energy loss in mixing and separation processes is neglected.
- Working fluid in the outlets of evaporator and condenser (LSC) is saturated.

- Pump, turbine and compressor work with fixed isentropic efficiencies.

In this study, ejector is simulated using one-dimensional model, the specific modeling refers to Zhao *et al.* (2023) and the isentropic efficiencies of the nozzle, mixing section and diffuser section are 0.9, 0.85 and 0.85, respectively.

Energy balance equations of all components are obtained based on the first law of thermodynamics and all heat exchangers are calculated by pinch point method, the details could refer to Yao *et al.* (2022).

As shown in figure 3, working fluid (x_T) is cooled to two-phase state by cooling water first in LSC (16-17), then the liquid phase (x_1) is artificially separated (17-8) and pumped to ejector (8-9), while the vapor phase (x_2) continues to be cooled by cooling water (18-1). The condensation pressure (P_{cond}) is acquired by iteration to satisfy that the minimum temperature difference between working fluid and cooling water is not less than the given pinch point of LSC (Sun *et al.* 2021).

For R1, the temperature of point 91 (t_{91}) is assumed the same as point 9 (t_9) first. After the calculation of ejector, the value of temperature at point 15 (t_{15}) is obtained. If t_{15} is larger than the sum of pinch point of R1 and t_{91}, regeneration in R1 happens and a small temperature increment is added to t_{91}. And this repetitive computation breaks up until t_{15} is less than the sum of pinch point of R1 and t_{91}. For R3, the maximum mass flow rate of coolant is limited by the result of preheater in power sub-cycle.

After satisfying the requirement of refrigeration conditions, the net power output (W_{net}) is selected as the evaluation index, given in Equation (1). In this study, maximum W_{net} is acquired through the method of exhaustion, and the independent controlling variables are obtained according to the optimization program. The program is implemented in MATLAB R2021a and the properties of working fluid are acquired by REFPROP 10.0a.

$$W_{\mathrm{net}} = W_{\mathrm{turb}} \cdot \eta_{\mathrm{gen}} - W_{\mathrm{pump1}}/\eta_{\mathrm{m}} - W_{\mathrm{pump2}}/\eta_{\mathrm{m}} - W_{\mathrm{comp}}/\eta_{\mathrm{m}} \tag{1}$$

4 RESULTS

4.1 Performance of the basic coupled CCP

There are three independent controlling variables of the basic coupled CCP namely CO_2 mass fraction (x_T), turbine inlet temperature ($t_{turb,in}$) and turbine inlet pressure ($P_{turb,in}$). Figure 5 demonstrates the performance of basic coupled CCP with the variation of $t_{turb,in}$ and x_T under the optimization of $P_{turb,in}$. 76

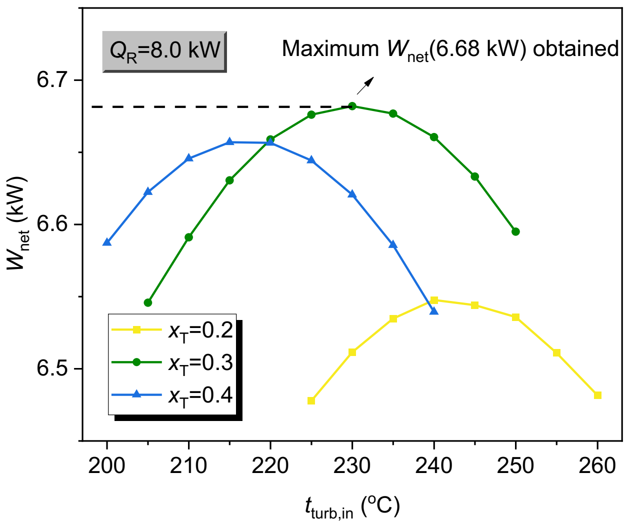

Figure 5: System performance of basic coupled CCP with variation of x_T and $t_{turb,in}$.

For a fixed x_T, the value of net power output (W_{net}) increases with $t_{turb,in}$ first and decreases then. Since the series connection of preheater and gas heater, the ideal mass flow rates to fully recover engine coolant and exhaust are interferential. A small $t_{turb,in}$ could realize full utilization of engine coolant while the utilization rate of engine exhaust is not guaranteed. For a large $t_{turb,in}$, the inverse situation will appear. Therefore, there always exists an optimal value of $t_{turb,in}$ to realize the maximum W_{net}.

For different x_T, the performance of W_{net} depends on the performance of thermal match in condenser since a better thermal match will lead a lower condensation pressure (Sun *et al.* 2021). As depicted in figure 2(b), when x_T = 0.3, thermal match in condenser performs well and temperature difference between working fluid and cooling water in most heat transfer processes approaches to pinch point. For x_T = 0.2 and x_T = 0.4, temperature glide changes and thermal match between working fluid and cooling water deteriorates, as a result, system performance declines.

4.2 Performance of the improved CCP

Since the optimal CO_2 mass fraction of mixture (x_T) equals 0.3 for the basic coupled CCP, the same value is set for the improved CCP. Thus, there are four independent controlling variables left namely primary flow pressure of ejector (P_{10}), normalized pressure index of ejector second flow (n), turbine inlet temperature ($t_{turb,in}$) and turbine inlet pressure ($P_{turb,in}$). To improve ejector performance, an auxiliary compressor is arranged between compressor and ejector. The index n represents the pressure boost amplitude and $n \geq 1$, as presented in equation (2), n =1 ($P_{13} = P_{evap}$) represents the compressor does not work.

$$P_{13} = P_{cond}^{1-\frac{1}{n}} \cdot P_{evap}^{\frac{1}{n}};\ n = \frac{\ln(P_{13}/P_{cond})}{\ln(P_{evap}/P_{cond})} \tag{2}$$

Compared to basic coupled CCP, the governing variables of refrigeration sub-cycle in improved CCP are two more. Therefore, the cooling capacity (Q_R) of improved CCP possesses a wider regulation range. Figure 6 illustrates seven optimal results of improved CCP and all of them perform better than basic coupled CCP. Conditions A to G refer to different proportions of Q_R and W_{net}. The values of governing variables are listed in Table 3. As shown in figure 6, Q_R increases with the decrease of W_{net} from condition A to G. The maximum promotion of W_{net} is 4.5% obtained in condition A and the maximum promotion of Q_R is 18.6% obtained in condition G. Beyond the superior output capacity, the improved CCP has a better flexibility in capacity adjustment than basic coupled CCP. 77

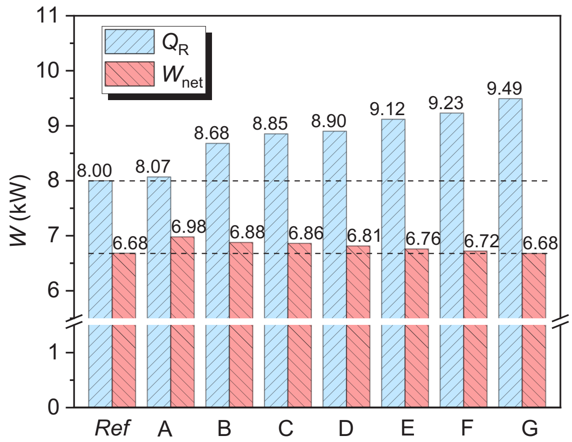

Figure 6: Optimal performance of basic coupled CCP (*Ref*) and improved CCP (A-G).

Table 3: Values of the governing variables for improved CCP

Condition	A	B	C	D	E	F	G
P_{10}/P_{cri}	0.85	0.85	0.80	0.80	0.75	0.80	0.75
n	4.4	4.2	5.0	5.0	5.0	4.4	5.2
$t_{turb,in}$(°C)	170	170	170	172	178	178	178
$P_{turb,in}$ (MPa)	9.1	9.1	9.1	9.1	9.1	9.1	9.1

P_{cri} means the critical pressure of zeotropic mixture with CO_2 mass fraction of x_1.

4.3 Effect of ejector refrigeration and composition adjustment

In this section, the effects of ejector refrigeration and composition adjustment on the improved CCP are investigated. For condition C as an example, the detailed operating parameters of the improved CCP are analyzed.

As figure 7 depicts, for the basic coupled CCP, heat utilization rate of exhaust (U_g) equals 0.914 since the installation of regenerator increases working fluid temperature at the inlet of gas heater and results in the insufficient utilization of exhaust. For the improved CCP, due to the removal of this regenerator, U_g equals 0.997. Although the optimal value of $t_{turb,in}$ for improved CCP decreases, which means the energy conversion efficiency declines, the heat imparted to CCP increases and the mass flow rate of working fluid in power sub-cycle ($m_{f,P}$) correspondingly rises. As shown in figure 7, the improvement rate is around 40% (from 0.16 kg/s to 0.22 kg/s).

Since the regenerator is removed, turbine exhaust vapor increases. To recover this part of heat energy and the residual heat of engine coolant, ejector refrigeration is introduced. As drawn in figure 7, for the improved CCP, heat utilization rate of coolant (U_{jw}) equals 1, and working fluid temperature at the inlet of condenser is 53 ºC, which is 47 ºC smaller than basic coupled CCP. As figure 2&3 exhibit, thermal match of condensation process in the improved CCP performs better than the basic coupled CCP.

Due to the high critical pressure of CO_2, the ejector would perform worse if CO_2 mass fraction becomes larger. Therefore, composition adjustment is introduced to realize a less CO_2 mass fraction in the primary flow of ejector. Besides, composition adjustment could realize higher CO_2 mass fraction in power sub-cycle, which could realize excellent heat recovery of dual heat sources for engine. In addition, composition adjustment could realize higher evaporation pressure in refrigeration sub-cycle, as depicted in figure 7, the evaporation pressure improves 4.5% (from 1.036 MPa to 1.082 MPa) and COP_m of refrigeration sub-cycle calculated by equation (3) improves 4.6% (from 4.80 to 5.02). It should be noticed that the introduction of LSC will increase the condensation pressure. However, the improvement of condensation process due to ejector refrigeration repairs the rise of condensation pressure. 78

$$COP_m = Q_R/(W_{pump2}/\eta_m + W_{comp}/\eta_m) \tag{3}$$

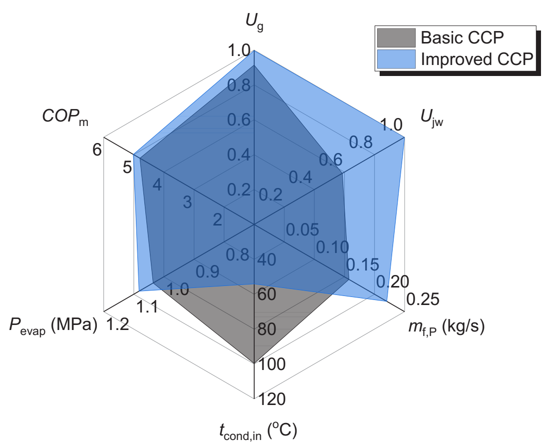

Figure 7: Parameter comparison between the basic coupled CCP and the improved CCP.

5 CONCLUSIONS

In this study, a novel improved CCP integrated of ejector refrigeration and composition adjustment is proposed, thermodynamic performance is analyzed and compared with the basic coupled CCP. The main conclusions are drawn as follows:

- The improved CCP has more flexible adjustability in capacity adjustment than the basic coupled CCP. Compared to the basic coupled CCP, the improved CCP can export 4.5% more net power output with the same cooling capacity output or export 18.6% higher cooling capacity with the same net power output.
- Ejector refrigeration could realize 100% recovery of dual heat sources and a smaller inlet temperature of condenser which means a better thermal match in condensation process.
- The introductions of composition adjustment and ejector supplement each other. Composition adjustment could realize lower CO_2 mass fraction of primary flow and higher inlet pressure of secondary flow for ejector, while ejector could repair the rise of condensation pressure brought by LSC.

NOMENCLATURE

Acronyms

COP	coefficient of performance
ICE	internal combustion engine
LSC	liquid separation condenser
R1/R2/R3	regenerator1/2/3

Symbol

m	mass flow rate	(kg/s)
t	temperature	(ºC)
P	pressure	(MPa)
Q	heat/cooling capacity	(kW)
U	utilization rate	
W	power	(kW)
x	mass fraction	(MPa)

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Subscript

g	engine exhaust
gen	generator
in	inlet
jw	engine coolant
m	mechanical
R	refrigeration sub-cycle
P	power sub-cycle

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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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