Supercritical CO_2-cycle configurations for internal combustion engine waste-heat recovery: A comparative techno-economic investigation

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Abstract:

Supercritical-CO_2 (S-CO_2) cycle systems have appeared as an attractive option for waste-heat recovery from internal combustion engines (ICEs) thanks to the advantages offered by CO_2 as a working fluid, which is nontoxic and non-flammable, and does not suffer decomposition at high temperatures. Since the high density of CO_2 in the supercritical region enables compact component design, various S-CO_2 cycle system configurations have been presented involving different layouts and combinations of heat exchangers with which to enhance heat recovery from both engine exhaust gases and jacket water streams. Despite the thermodynamic performance improvement offered by more complex configurations, the additional heat exchangers bring extra costs and therefore key thermo-economic decisions need to be considered carefully during the design and development of such systems. This paper seeks to conduct both thermodynamic and economic (cost) assessments of a variety of S-CO_2 cycle system configurations in ICE waste-heat recovery applications, with results indicating that in some cases a significant thermodynamic performance improvement can compensate the extra costs associated with a more complex system structure. The comparison results across a range of ICEs can also be a valuable guide for the early-stage S-CO_2 cycle system design in ICE waste-heat recovery and other similar applications.

Keywords:

ICE waste-heat recovery, supercritical CO_2, thermo-economic, performance assessment, configurations

1. Introduction

Internal combustion engines (ICEs) have been used as primary propulsion equipment in transportation sectors as well as in combined heat and power (CHP) units. However, in typical ICEs, nearly half of the total energy from fuel is rejected via engine exhaust gases and jacket water. Waste-heat recovery from these two streams is acknowledged as an attractive pathway to improve the engine performance [1,2], so as to alleviate the current crucial issues associated with energy shortage and environmental deterioration. Compared to organic Rankine cycle (ORC) systems, which has been proven as an effective technology for heat-to-power generation [3,4], especially from heat sources in the temperature range of 100 – 400 ℃ [5], supercritical CO_2 (S-CO_2) cycle systems have also emerged as a promising solution thanks to the favourable properties offered by CO_2 as the working fluid. CO_2 is non-toxic, non-flammable, and it is also free from decomposition under high-temperature conditions, which ensures safe operations in certain cases of heat recovery from engine exhaust gases with high temperature up to 500 ℃.

There is an increasing interest in the research of S-CO_2 cycle systems for ICE waste-heat recovery and relevant patents can also be found from Echogen (Ohio, USA) and General Electric (New York, USA) [6,7]. Di Bella and Francis [8] pointed out that S-CO_2 cycle could potentially replace the steam Rankine cycle to utilise waste heat from shipboard engines and further improve the overall system performance. Manjunath et al. [9] investigated the performance of a supercritical regenerative CO_2 cycle combined with a transcritical CO_2 cycle cooling cycle for shipboard applications, and the results revealed that the shipboard power and cooling capacity were enhanced by about 18% and 15% at the full load condition. Sharma et al. [10] presented a waste-heat-recovery system based on S-CO_2 regenerative recompression cycle, and the overall thermal efficiency was improved by 10% while the gross net power was found to be increased by up to 25% relative to the engine rated power. Hou et al. [11] proved that advantages of S-CO_2 cycle systems for engine waste-heat recovery included sufficient heat utilisation, high system compactness and low costs, through their case studies of using a combined cycle system that coupled S-CO_2 recompression and regenerative cycles.

The high density of CO_2 under supercritical conditions enables compact designs of the key components, e.g., heat exchangers, compressors and expanders, which is particularly favourable in space-constraint situations. On the other hand, various configurations/layouts that involve different combinations of heat exchangers can be considered as the additional space required would not be a significant concern in practical applications. However, the added costs associated with the extra heat exchangers and complex system structure need to be carefully considered, although thermodynamic performance improvement can be expected. Relevant studies on S-CO_2 cycle systems for ICE waste-heat recovery are extensive, which mainly focus on parametric analyses and system optimisation, while only few have implemented comparison on different configurations, in particular on the trade-off between better thermodynamic performances and higher system costs. This paper seeks to explore the comprehensive comparison of various configurations, i.e., recuperated/non-recuperated, single-/dual-preheating, without/with the additional recuperator, to explore the influence of extra heat exchangers and select the optimal system from thermodynamic and economic perspectives. A detailed case study for a specific ICE is carried out in terms of parametric analyses and thermo-economic optimisation, and further investigation and comparison are also implemented considering a wide range of various ICEs.

2. Methodology

2.1. Configurations

Four configurations of S-CO_2 cycle systems, i.e., non-recuperated single-preheating system, recuperated single-preheating system, recuperated dual-preheating system and recuperated dual-preheating system with an additional recuperator, are considered in this study to recover waste heat from both the jacket water and the engine exhaust gases, with schematic diagrams shown in Fig. 1.

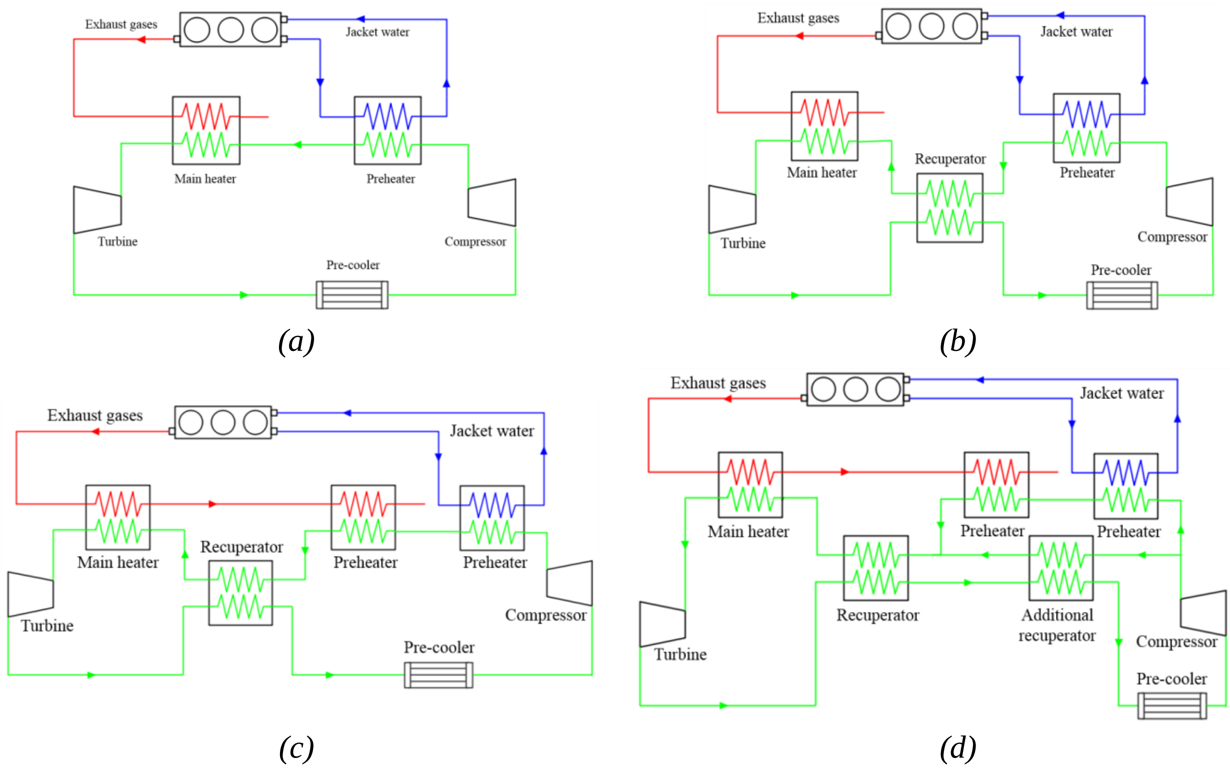

Figure 1. Schematic diagrams of S-CO_2 cycle systems for ICE waste-heat recovery: (a) non-recuperated single-preheating system, (b) recuperated single-preheating system, (c) recuperated dual-preheating system, and (d) recuperated dual-preheating system with an additional recuperator.

The low-temperature jacket water is aimed to preheat the supercritical CO_2 working fluid, while the high-temperature engine exhaust gases is mainly utilised in the main heater. The four configurations are presented in an order of achieving a more sufficient heat utilisation but also a more complex system structure with additional heat exchangers. A recuperator is added to the non-recuperated single-preheating system firstly to recover energy of the hot CO_2 stream flowing from the turbine. As a consequence, the higher temperature at the main heater inlet (recuperator outlet) will hinder the heat recovery from engine exhaust gases, since the ending temperature of the gases is raised. In the recuperated dual-preheating system (see Fig. 1(c)), the low-temperature portion of the engine exhaust gases can be further utilised in the second preheater. Although a sufficient utilisation of the heat sources can be achieved in that case, the recuperation will be suppressed due to the higher preheating temperature. In the most complex system, namely the recuperated dual-preheating system with an additional recuperator, heat rejected in the pre-cooler will be reduced by including the second/low-temperature recuperation branch.

2.2. Thermodynamic models

A *T-s* diagram representing the recuperated dual-preheating system configurations (see Fig. 1(c)) is shown in Fig. 2. The corresponding thermodynamic relations across all components are listed below.

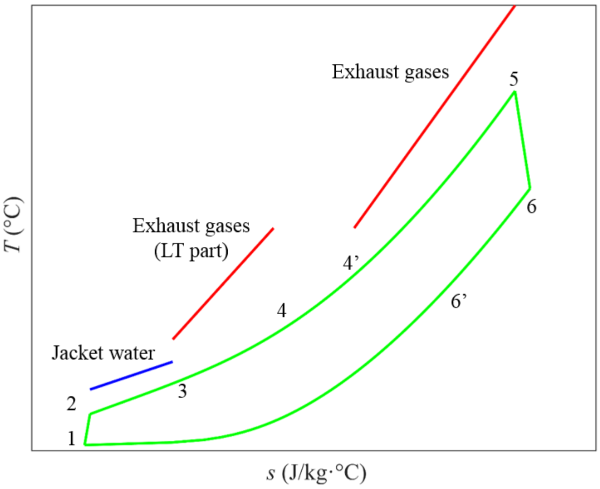

Figure 2. T-s diagram of recuperated dual-preheating S-CO_2 cycle system for ICE waste-heat recovery.

Process 1 to 2 in the compressor:

$$W_{\mathrm{comp}} = \frac{\dot{m}\cdot\left(h_{2s}-h_1\right)}{\eta_{\mathrm{comp}}}. \tag{1}$$

Process 2 to 3 in the low-temperature preheater (for jacket water):

$$Q_{\mathrm{preh},1} = \dot{m}\cdot\left(h_3-h_2\right)=\dot{m}_{\mathrm{jw}}\cdot c_{\mathrm{p_{jw}}}\cdot\left(T_{\mathrm{jw,in}}-T_{\mathrm{jw,out}}\right), \tag{2}$$

where $\dot{m}_{\mathrm{jw}}$ is the mass flow rate of jacket water that flows into the preheater, which can be less than the total. In other words, this mass flow rate can be controlled according to the preheating heat demand.

Process 3 to 4 in the high-temperature preheater (for engine exhaust gases):

$$Q_{\mathrm{preh},2} = \dot{m}\cdot\left(h_4-h_3\right)=\dot{m}_{\mathrm{gas}}\cdot c_{\mathrm{p_{gas}}}\cdot\left(T_{\mathrm{gas,m}}-T_{\mathrm{gas,out}}\right). \tag{3}$$

Process 4 to 4' in the recuperator:

$$Q_{\mathrm{recup}} = \dot{m}\cdot\left(h_{4'}-h_4\right)=\dot{m}\cdot\left(h_6-h_{6'}\right). \tag{4}$$

Process 4' to 5 in the main heater:

$$Q_{\mathrm{main}} = \dot{m}\cdot\left(h_5-h_{4'}\right)=\dot{m}_{\mathrm{gas}}\cdot c_{\mathrm{p_{gas}}}\cdot\left(T_{\mathrm{gas,in}}-T_{\mathrm{gas,m}}\right). \tag{5}$$

Process 5 to 6 in the turbine:

$$W_T = \dot{m} \cdot (h_5 - h_{6s}) \cdot \eta_T \; . \quad (6)$$

2.3. Cost models

The module costing technique is used to calculate the bare module cost of each component, with the chemical engineering plant cost index (*CEPCI*) to obtain the capital cost of the systems [12]. The specific investment cost (*SIC*) is calculated by Eqs. (7)-(11) with the coefficients summarised in Table 1.

$$C_{BM} = C_p^0 F_{BM} = C_p^0 (B_1 + B_2 F_M F_P) \; , \quad (7)$$

$$\log(C_p^0) = K_1 + K_2 \log(X_i) + K_3 \left[\log(X_i)\right]^2 , \quad (8)$$

$$\log(F_p^0) = C_1 + C_2 \log(P_i) + C_3 \left[\log(P_i)\right]^2 , \quad (9)$$

$$Cost = \sum_i C_{BM} \frac{CEPCI_{2017}}{CEPCI_{2001}} \; , \quad (10)$$

$$SIC = \frac{Cost}{W_{net}} \; , \quad (11)$$

where i denotes different components and X is the capacity of each component, e.g., power (for compressor and turbine) and heat exchange areas. Shell-and-tube heat exchangers are selected for the S-CO_2 cycle systems, and the Bell-Delaware method [13] is used to calculate heat transfer coefficients (HTCs) and pressure drops. Detailed models used to calculate the heat exchanger area can be found in the authors' previous work [14,15]. In addition, $CEPCI_{2001}$ and $CEPCI_{2017}$ are set to be 397 and 567.5 [16], which are dimensionless numbers employed to update capital cost required to erect a power-cycle system from a past date to a later time.

Table 1. Coefficients for each component in cost models [12].

Component	K_1, K_2, K_3	C_1, C_2, C_3	B_1, B_2	F_M	F_{BM}
Pump	K_1 = 3.3892 K_2 = 0.0536 K_3 = 0.1538	C_1 = -0.3935 C_2 = 0.3957 C_3 = -0.0023	B_1 = 1.89 B_2 = 1.35	1.0	/
Compressor	K_1 = 2.2897 K_2 = 1.3604 K_3 = -0.1027	/	/	/	2.8
Turbine	K_1 = 2.2476 K_2 = 1.4965 K_3 = -0.1618	/	/	/	3.5
Heat exchanger	K_1 = 4.3247 K_2 = -0.3030 K_3 = 0.1634	C_1 = -0.0016 C_2 = -0.0063 C_3 = 0.0123	B_1 = 1.63 B_2 = 1.66	1.35	/

3. Heat source conditions and assumptions

A range of different ICEs (E200, E250, E310, E375, E500 and E1165, with parameters under the rated conditions listed in Table 2) manufactured by Ener-G House are considered, among which E375 is selected for a case study to carry out parametric analyses and thermo-economic optimisation of the presented configurations (see Section 4.1), while further investigation and comparison in terms of both thermodynamic and economic performance will be implemented on all the ICEs (see Section 4.2).

Table 2. ICE parameters from technical datasheets.

Type	E200	E250	E310	E375	E500	E1165
Power output (kW)	205	255	310	376	502	1170
Jacket water inlet temperature (℃)	80	80	80	80	80	78
Jacket water outlet temperature (℃)	90	90	90	90	90	89
Jacket water mass flow rate (kg/s)	2.9	4.2	3.6	3.8	5.0	13.0
Exhaust gases temperature (℃)	453	460	490	482	482	457
Exhaust gases mass flow rate (kg/s)	0.32	0.39	0.50	0.60	0.79	1.69

Thermo-economic model for the S-CO_2 cycle system for ICE waste-heat recovery used in this study is developed in in-house MATLAB codes and working fluid properties are acquired from NIST REFPROP [17]. The interior-point algorithm *fmincon* [18] is chosen as the solver to maximise the system net power output and minimise the *SIC* (or cost), by optimising the operating conditions. The main conditions and assumptions are given below:

(1) In the parametric analysis (results shown in Figs. 3-5), compressor inlet conditions of the topping S-CO_2 cycle are set to 32 ℃ and 7.8 MPa [19].
(2) Compressor and turbine efficiencies are assumed to be 0.8.
(3) In the parametric analysis (results shown in Figs. 3-5), pinch point temperature difference for all heat exchangers is set to be 6 ℃.
(4) Heat losses of the system are neglected.
(5) Mass flow rate of jacket cooling water that flows to the preheater can be adjusted according to the preheating heat demand.
(6) In the recuperated dual-preheating system with an additional recuperator, outlet temperatures of the second (high-temperature) preheater and the additional recuperator are set to the same.

4. Results and discussion

4.1. Case study of E375

E375 is selected to conduct parametric analyses of all the presented configurations. For the engine exhaust gases, in order to avoid acid corrosion to the pipes and heat exchangers of the system, its outlet/ending temperature from the heat recovery unit is set to be above 120 ℃ [20]; thus the maximum available heat amounts to 237 kW. The available heat from the jacket water is around 161 kW. Recovering the thermal energy from these two streams to generate additional power offers a promising potential for improving the overall system performance without increasing the fuel consumption.

Thermodynamic performance of the S-CO_2 cycle systems under various operating conditions are shown in Fig. 3. The non-recuperated single preheating system delivers the lowest net power output as no energy of the hot S-CO_2 stream from the turbine is recovered, which confirms that recuperator is preferable and also necessary in S-CO_2 cycle systems. Moreover, with more heat exchangers involved in the system, system can achieve a better thermal performance as expected.

Turbine inlet condition of $T_{T,in}$ = 350 ℃ and $P_{T,in}$ = 20 MPa is chosen to compare the system thermodynamic performance in detail, since all the configurations can yield a larger net power output and differences among system performances are the most significant as shown in Fig. 3.

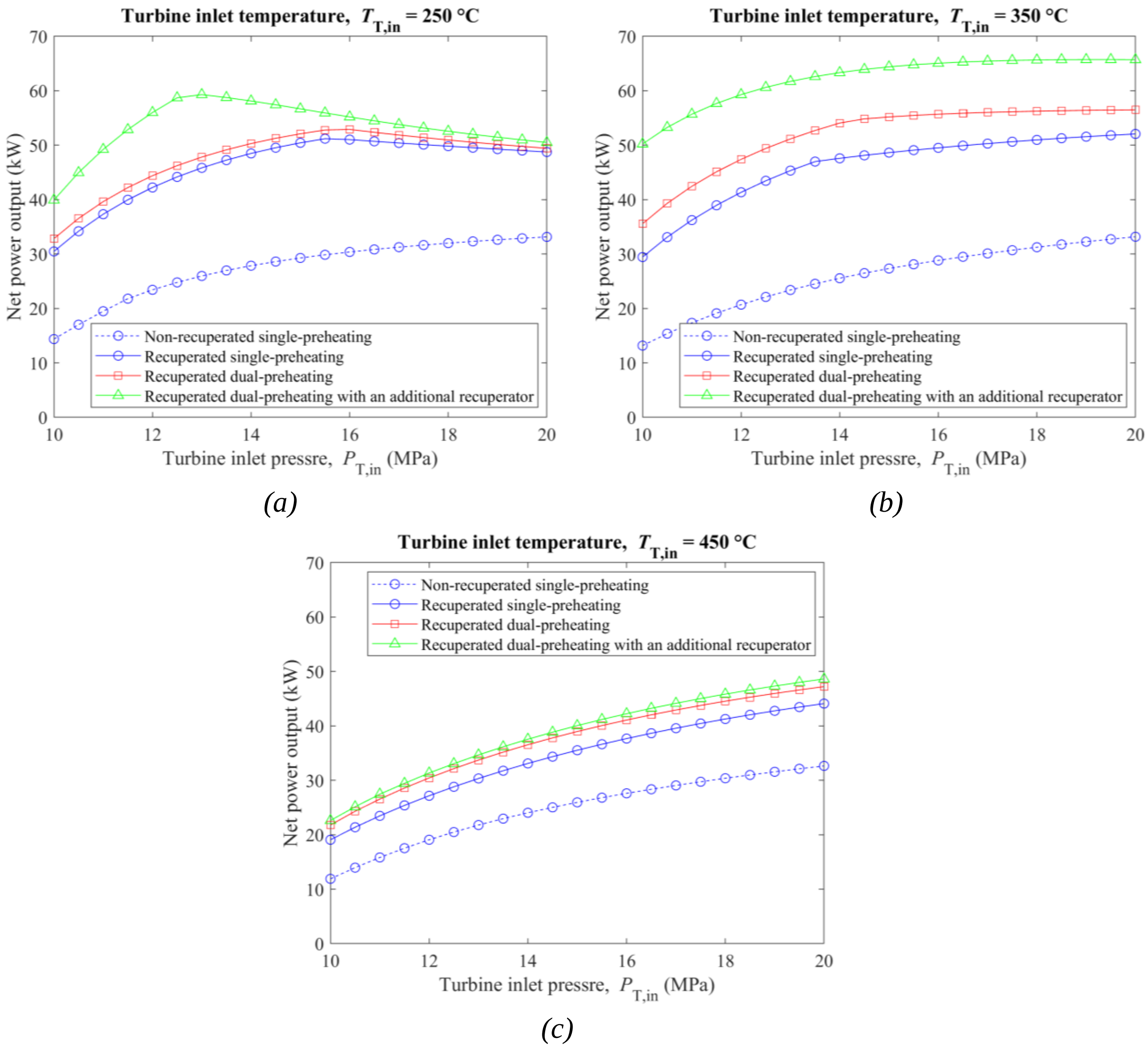

Figure 3. Thermodynamic performance of S-CO_2 cycle systems under various operating conditions.

T-Q plots of the four configurations under the selected operating condition ($T_{T,in}$ = 350 ℃ and $P_{T,in}$ = 20 MPa) are shown in Fig. 4. In the non-recuperated single-preheating system, thermal load in the preheater is only 40 kW, which accounts for a small fraction of the total available from the jacket water (161 kW). The narrow temperature range (80 – 90 ℃) of the jacket water makes it difficult to preheat the CO_2 working fluid to a high temperature and to be sufficiently recovered. Although the engine exhaust gases can be cooled down to 120 ℃ and hence the entire heat from this heat source can be utilised, the total heat absorbed by the system is only 270 kW, which therefore hinders the system thermodynamic performance. By adding the recuperator in the system as shown in Fig. 4(b), the total heat load absorbed by the system can be significantly increased to 430 kW, of which the contribution by the recuperator amounts to nearly 44%. However, the higher inlet temperature of the main heater (after recuperation) raises the ending temperature of the engine exhaust gases to ~200 ℃. The dual-preheating system is then considered to further recover the low-temperature portion of the engine exhaust gases as a second preheating medium after the jacket water. The engine exhaust gases can be then cooled down to 120 ℃ again, while the heat in the recuperator is slightly reduced. The S-CO_2 stream flowing to the pre-cooler has a high temperature of ~120 ℃, therefore there is still a potential to utilise its thermal energy and achieve a better thermodynamic performance. Figure 4(d) shows that with an additional recuperator involved, the pre-cooler inlet temperature can be decreased to 66 ℃ and the heat rejected in the pre-cooler is reduced. Although the recovered heat from the jacket water is reduced to 26 kW, the increased thermal load in the other heat exchangers can obviously compensate this slight decrement in the (first/low-temperature) preheater. The total heat absorbed by the most complex system reaches the highest of 540 kW. Figure 5 shows a direct comparison of the thermal load in different exchangers involved in all the configurations, which additionally indicates the influence of extra heat exchangers on the heat utilisation and distribution.

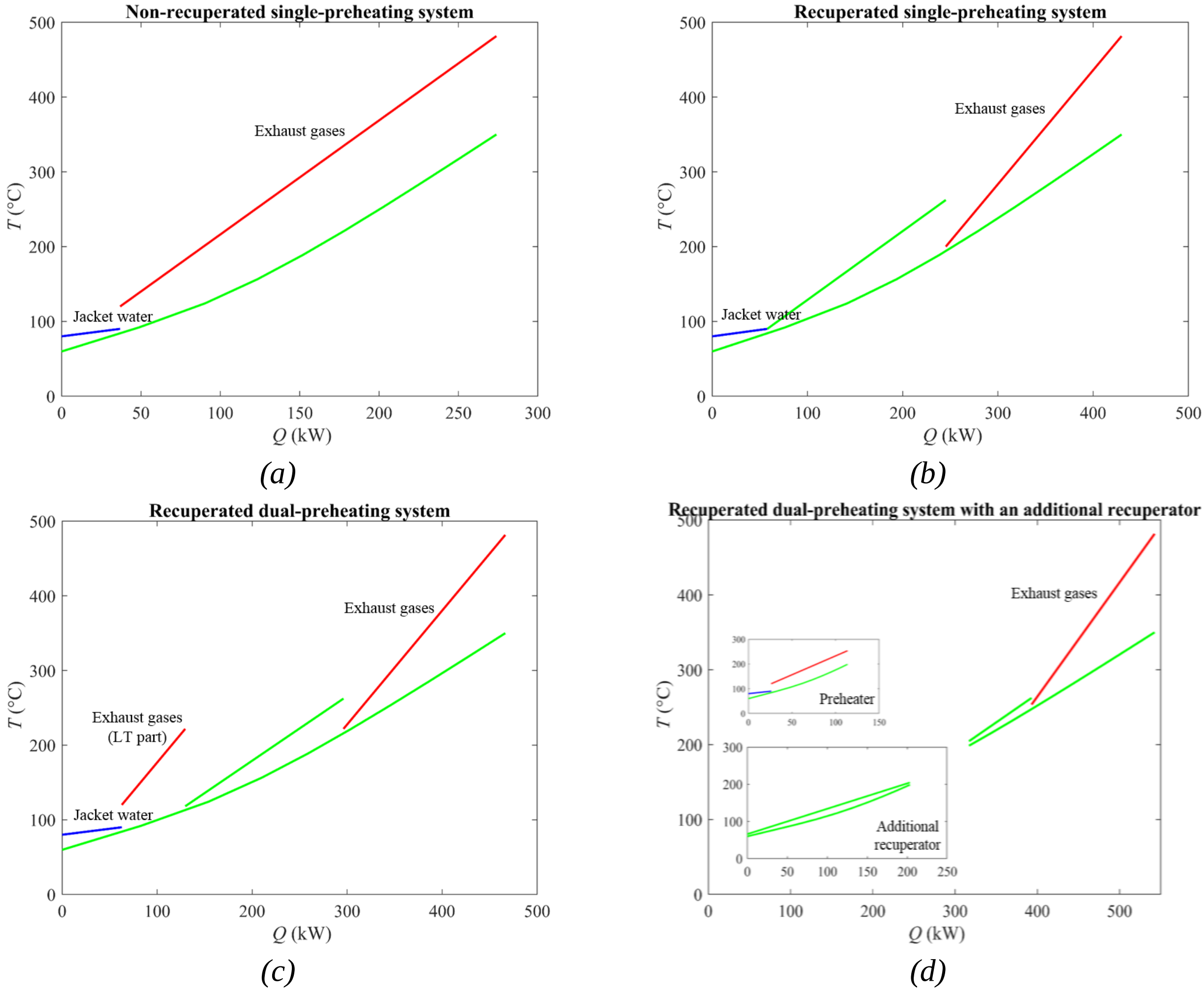

Figure 4. T-Q plots of S-CO_2 cycle systems with operating condition of $T_{T,in}$ = 350 ℃ and $P_{T,in}$ = 20 MPa.

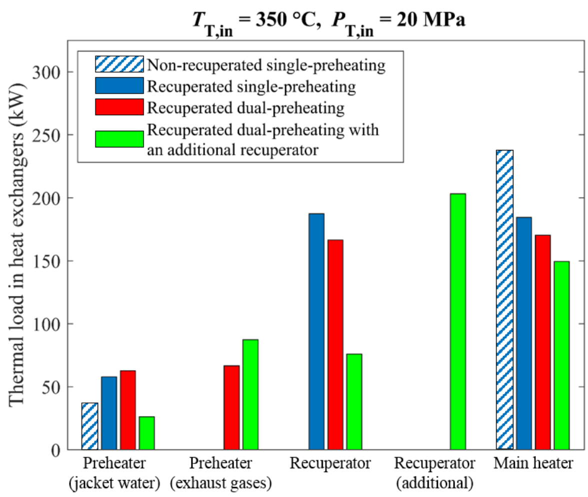

Figure 5. Thermal load in heat exchangers of different S-CO_2 cycle system configurations with operating condition of $T_{T,in}$ = 350 ℃ and $P_{T,in}$ = 20 MPa.

Figure 6 shows the maximum net power output, the corresponding *SIC* (to the maximum net power output) and the minimum *SIC* of the four S-CO_2 cycle system configurations considering the full spectrum of possible operating conditions. In other words, all the design parameters including compressor and turbine inlet temperature and pressure, and pinch point temperature difference are released here so as to obtain the optimal operating conditions and to either maximise the net power output or minimise the *SIC*, which is different from the parametric analysis above (the compressor inlet temperature and pressure, and the pinch point temperature difference are fixed, while the turbine inlet temperature and pressure vary). The maximum net power output of the non-recuperated single preheating system, recuperated single-preheating system, recuperated dual-preheating system and recuperated dual-preheating system with an additional recuperator are 35 kW, 57 kW, 62 kW and 72 kW, and the corresponding *SIC* are 9430 $/kW, 7120 $/kW, 6850 $/kW and 9410 $/kW, respectively. While from the perspective of the lowest system cost, the three recuperated configurations delivers a similar *SIC* at 6900 $/kW approximately, which is nearly 24% lower than that of the non-recuperated single-preheating system.

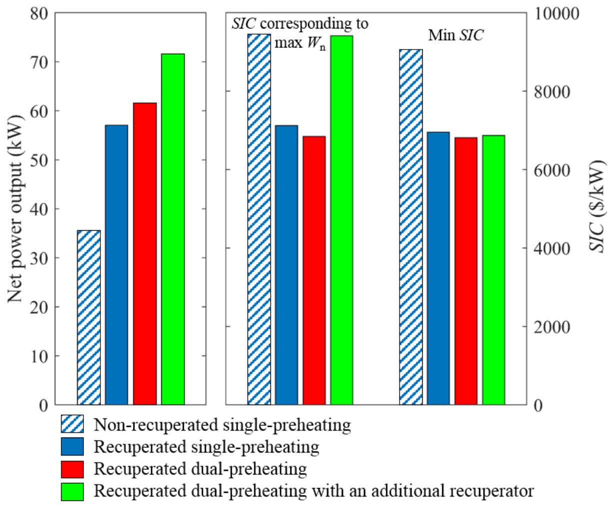

Figure 6. Net power output (left) and SIC (right) of different S-CO_2 cycle system configurations for ICE waste-heat recovery.

Multi-objective optimisation (i.e., thermo-economic optimisation) is also implemented for the presented configurations via NSGA-II (Non-Dominated Sorting Genetic Algorithm II) [18]. NSGA-II is one of the most popular multi-objective optimisation algorithms that works by initializing random individuals subjected to a set of constraints, calculating, evaluating and ranking the fitness function of each individual, and achieving a final non-dominated solution where there exists no other solution that outperforms it in both objective functions. Pareto front solutions are obtained and plotted in Fig. 7. Recuperator is recommended to be included when the S-CO_2 cycle system has a capacity higher than 25 kW, as it would enhance the thermodynamic performance with a given capital cost of the system. Although the Pareto fronts of the recuperated single-preheating system, dual-preheating system and dual-preheating system with an additional recuperator are similar, a higher net power output can be achieved with more heat exchangers involved while the overall capital cost will also be higher, which is consistent with the results as shown in Fig. 6.

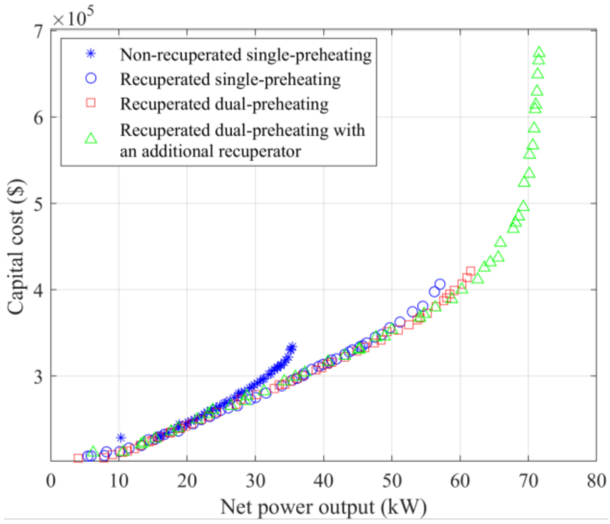

Figure 7. Thermo-economic optimisation results of different S-CO_2 cycle system configurations.

4.2. Investigation on various ICEs

A range of ICEs listed in Table 2 are considered to further compare the thermodynamic and economic performances of the presented S-CO_2 cycle system configurations. Figure 8(a) shows that adding more heat exchangers can significantly improve the maximum net power output as expected. More precisely, the recuperator can bring an improvement of 60% in the optimal thermodynamic performance across all the selected ICEs, the high-temperature preheater (to achieve sufficient heat recovery from engine exhaust gases) would increase the net power output by 7% averagely, and the additional recuperator results in an increment of 12%-20% in the net power output relative to the dual-preheating system. While from the perspective of economic performance, the recuperated dual-preheating system delivers the minimum *SIC* among all the presented configurations except for E310. However, with the increment of the ICE power as well as the waste heat available from the engine exhaust gases and the jacket water, the recuperated dual-preheating system with an additional recuperator also emerges as an attractive alternative. Considering the significant thermodynamic performance improvement by involving the additional preheater and recuperators, which can compensate the extra cost associated with the added components, the recuperated dual-preheating system with an additional recuperator is still appealing in practical applications, although with a complex structure.

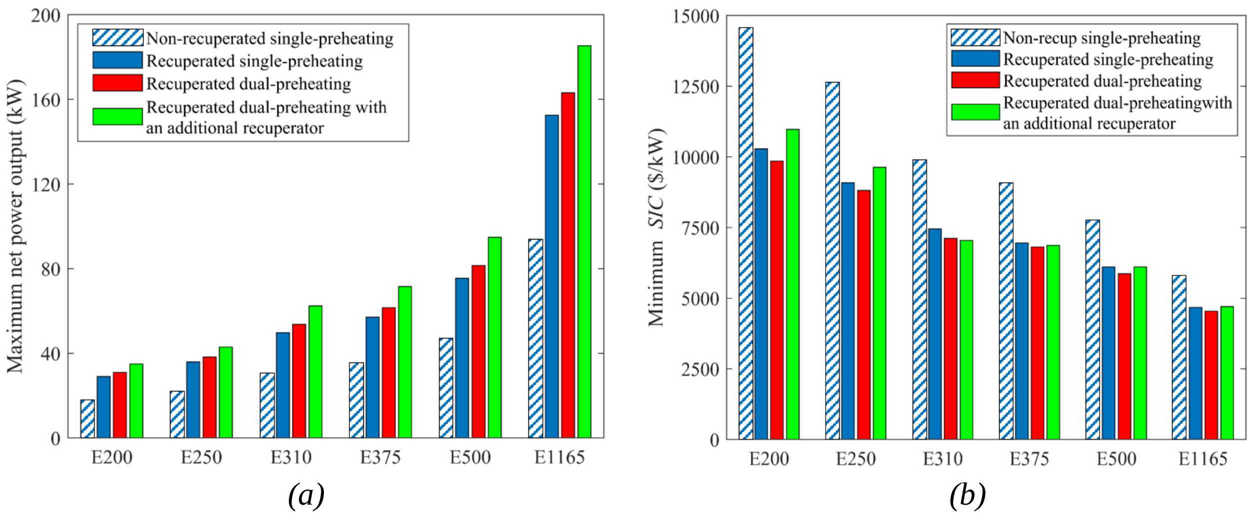

Figure 8. (a) Maximum net power output, and (b) minimum SIC of different S-CO_2 cycle system configurations for waste-heat recovery from a range of ICEs.

5. Conclusions

In this paper, we presented and discussed results from a study that aimed to consider a number of different S-CO_2 cycle system configurations for ICE waste-heat recovery, including a non-recuperated single-preheating system, a recuperated single-preheating system, a recuperated dual-preheating system and a recuperated dual-preheating system with an additional recuperator. The influence of adding heat exchangers on the thermodynamic and economic (capital cost) performance of these S-CO_2 cycle system configurations were investigated and compared. The results revealed that a recuperator is preferable in S-CO_2 cycle systems as it can aid the recovery of significant thermal energy from the hot CO_2 stream from the turbine. A second/high-temperature preheater used to further cool down the engine exhaust gases can achieve sufficient utilisation of this high-temperature heat source. An additional recuperator can further maximise the potential of heat utilisation by decreasing the pre-cooler inlet temperature and reducing the heat rejected by the S-CO_2 cycle system.

An ICE (E375) with a rated power of 376 kW was selected for a detailed comparison of the presented system configurations as a case study. The maximum net power output of the configurations proposed above was: 35 kW, 57 kW, 62 kW and 72 kW, respectively. Further, the minimum *SIC* was found to be around 6900 $/kW, achieved by the three recuperated systems, which was 24% lower than that of the non-recuperated single-preheating system. Further investigation over a range of ICEs indicates that the recuperator can bring an improvement of 60% in the optimal thermodynamic performance in all selected cases, the high-temperature preheater (to achieve sufficient heat recovery from engine exhaust gases) can increase the net power output by 7% on average, and the additional recuperator results in an increment of 12%-20% relative to the dual-preheating system. While from the perspective of economic performance, although the recuperated dual-preheating system delivers the minimum *SIC* among almost all the cases, with the increment of the ICE power as well as the waste heat available from the engine exhaust gases and the jacket water, involving the additional recuperator also appears attractive. The recuperated dual-preheating system with an additional recuperator is a relatively complex configuration, however, its significant performance improvement can compensate for the costs associated with the additional heat exchangers, making this appealing and promising in practical applications.

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Nomenclature

Abbreviations

B, C, F, K	coefficients for cost calculation	*P*	pressure (Pa)
C_{BM}	bare module cost ($/W)	*Q*	heat (W)
c_p	specific heat capacity (J/kg K)	*T*	temperature (K)
f	friction coefficient	*W*	power (W)
h	specific enthalpy (J/kg)	*X*	component capacity indexes
$\dot{m}$	mass flow rate (kg/s)		

Greek symbols

η	efficiency

Subscripts

1 - 6, 4’, 6’	state point	m	middle
comp	compressor	main	main heater
gas	engine exhaust gases	out	outlet
in	inlet	T	turbine
jw	jacket water	*i*	component

Acronyms

CEPCI	chemical engineering plant cost index	SIC	specific investment cost
CHP	combined heat and power	S-CO_2	supercritical CO_2
ICE	internal combustion engine	ORC	organic Rankine cycle

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