

OPTIMAL DESIGN OF SUPERCRITICAL CO₂ (S-CO₂) CYCLE SYSTEMS FOR INTERNAL COMBUSTION ENGINE (ICE) WASTE-HEAT RECOVERY CONSIDERING HEAT SOURCE FLUCTUATIONS

Jian Song

Imperial College London
London, UK

Jiangfeng Wang

Xi'an Jiaotong University
Xi'an, China

Yaxiong Wang

Xi'an Jiaotong University
Xi'an, China

Christos N. Markides*

Imperial College London
London, UK

c.markides@imperial.ac.uk

ABSTRACT

Supercritical CO₂ (S-CO₂) cycle systems have emerged as an attractive alternative for internal combustion engine (ICE) waste-heat recovery thanks to the advantages offered by CO₂ as a working fluid, including robust performance and system compactness. The engine exhaust gases are the main available heat source from ICEs with promising thermodynamic potential for further utilisation, and whose conditions, i.e., temperature and mass flow rate, vary based on the ICE operating strategy/load. These heat source variations have a critical influence on the performance of a bottoming S-CO₂ cycle system, which needs to be carefully considered in the design stage. This paper explores the optimal design of S-CO₂ cycle systems for ICE waste-heat recovery considering heat source fluctuations as well as the probability of their occurrence as arising from actual ICE operation. A variety of heat source conditions are selected for separate designs of an S-CO₂ cycle system and performance prediction under all possible scenarios is evaluated via detailed design and off-design models, so as to select the optimal design that is able to match the heat source fluctuations and exhibit the best performance from thermodynamic and economic perspectives. The advantage of this approach relative to the conventional ones that only consider one specific design condition is that it avoids either over- or under-sizing of the S-CO₂ cycle system, which also achieves comprehensive insight of the interplay between the bottoming heat recovery system and the ICE, and provides valuable guidance for further system optimisation.

INTRODUCTION

More than half of the energy through fuel combustion in internal combustion engines (ICEs) is dissipated in the form of waste heat mainly via exhaust gases and jacket water [1]. Therefore, waste-heat recovery is considered as an important pathway for ICE performance improvement, of which organic Rankine cycle (ORC) appears as an attractive solution [2-5]. More recently, supercritical CO₂ (S-CO₂) cycles have also emerged as a promising option for ICE waste-heat

recovery thanks to the advantages offered by CO₂ as a working fluid, including better thermal match with the heat source, compact system structure and avoidance of working fluid decomposition under high-temperature conditions in particular for the recovery of engine exhaust gases [6,7]. S-CO₂ cycle systems also offer potential to be applied in a wide range of applications such as solar [8,9], nuclear [9,10] and geothermal energy utilisation [11,12].

Extensive research on S-CO₂ cycle systems for ICE waste-heat recovery is available, showing the growing interest in adopting this promising heat-to-power generation technology in the specific area to further enhance ICE performance. Chen et al. [13] compared the performance of supercritical CO₂ Brayton cycle and transcritical CO₂ cycle for heat recovery from automobiles via a theoretical study and the results revealed that S-CO₂ cycle had a higher efficiency thanks to the higher operating pressure. Song et al. [14] updated a single preheating S-CO₂ cycle system by adding a regeneration branch and an increase of 7% in net power output was achieved. Further to that work, a combined S-CO₂ cycle and ORC system was presented for a wide range of ICEs, with results indicating that a power output improvement of up to 40-70% could be obtained by the addition of the bottoming cycle system [15]. Hou et al. [16] analysed a combined-cycle system that coupled S-CO₂ recompression and regenerative cycles for marine engine waste-heat recovery and the system was able to offer various advantages including sufficient heat utilisation, high system compactness and low cost. Sharma et al. [17] presented a regenerative S-CO₂ recompression cycle also for shipboard applications and an increase up to 25% in engine power output at rated condition was achieved.

Most of the previous research focuses on the design and optimisation of S-CO₂ cycle systems with a specific heat source condition, i.e., fixed temperature and mass flow rate, normally at the rated operating condition of the ICE. It is common that ICEs will be operated under frequent part-load conditions, which

results in different heat source conditions for the bottoming heat recovery system. In other words, the S-CO₂ cycle system will be forced to operate under off-design conditions associated with the ICE load and its performance as well as the improvement in the ICE power output will be significantly influenced. Therefore, the off-design performance of the S-CO₂ cycle system needs to be considered in the design stage to avoid performance deterioration due to the mismatch between the determined system configurations (e.g., sizes and geometry of the components) and the heat source conditions, which has been rarely taken into consideration in the previous studies. This paper seeks to explore the optimal design of an S-CO₂ cycle system for ICE waste-heat recovery considering the heat source fluctuations and the probability of occurrence of such part-load conditions. All the possible heat source conditions corresponding to actual ICE operating load are selected for separate designs of the S-CO₂ cycle system and performance evaluation under all possible scenarios via detailed design and off-design models are performed, so as to select the optimal design scheme from thermodynamic and economic perspectives.

METHODOLOGY

System description

Recuperated S-CO₂ cycle system is considered in this study for waste-heat recovery from engine exhaust gases, of which the schematic diagram and the corresponding T-s diagram are shown in Figures 1 and 2. The CO₂ working fluid is firstly compressed by the compressor and heated in the recuperator by the hot CO₂ stream that flows from the turbine. CO₂ continuously absorbs heat from the engine exhaust gases and then expands in the turbine to produce power. Afterwards, CO₂ goes into the pre-cooler to be cooled down to start the next cycle.

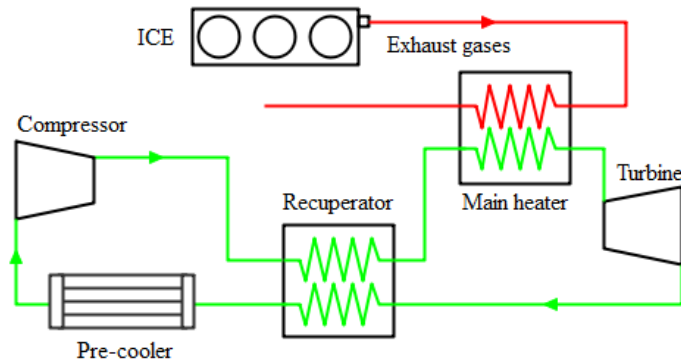


Figure 1: Schematic diagram of recuperated S-CO₂ cycle system for ICE waste-heat recovery.

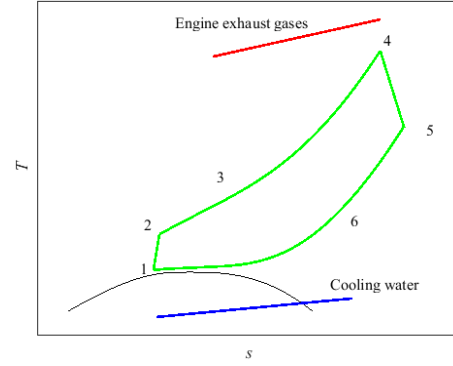


Figure 2: T-s diagram of recuperated S-CO₂ cycle system for ICE waste-heat recovery.

Thermodynamic models of the S-CO₂ cycle system can be found in the authors' previous work [14,15] and they are omitted here.

Turbine model

A one-dimensional (1-D) design and off-design model of radial-inflow turbines [18,19] based on the mean-line method is used to predict the turbine efficiency, which has been employed in ORC [20] and CO₂-based systems [6,21]. In the design model, all relevant parameters, including the nozzle and rotor velocity coefficients, velocity ratio, reaction degree, wheel diameter ratio, absolute flow angle at the rotor inlet and relative flow angle at the rotor outlet, are optimised simultaneously in order to achieve the highest turbine efficiency. While in the off-design model, the geometry of the turbine determined by the design stage is provided as input and the flow characteristics as well as the working fluid parameters are predicted.

The peripheral efficiency of the radial-inflow turbine can be expressed as:

$$\eta_u = 2\bar{u}_1 \cdot \left(\frac{\varphi \cdot \sqrt{1-\Omega} \cdot \cos \alpha_1 - \bar{D}_2^2 \cdot \bar{u}_1 + \bar{D}_2 \cdot \psi \cdot \cos \beta_2}{\left[\Omega + \varphi^2 \cdot (1-\Omega) - 2\varphi \cdot \sqrt{1-\Omega} \cdot \bar{u}_1 \cos \alpha_1 + \bar{D}_2^2 \cdot \bar{u}_1^2 \right]^{1/2}} \right) \quad (1)$$

Taking incidence loss, friction loss and leakage losses into consideration, the radial-inflow turbine efficiency is given by:

$$\zeta_1 = \frac{\left[w_1 \cdot \sin(\beta_1 - \beta_{1,opt}) \right]^2}{2 \cdot \Delta h_s}, \quad (2)$$

$$\zeta_f = \frac{K_f \cdot \frac{(\rho_1 + \rho_2)}{2} \cdot u_1^3 \cdot \left(\frac{D_1}{2} \right)^2}{2 \cdot m \cdot w_2^2 \cdot \Delta h_s}, \quad (3)$$

$$\zeta_1 = \frac{w_1^3 \cdot N_{rotor}}{8\pi \cdot \Delta h_s} \cdot \sqrt{\frac{0.4 \cdot 0.004 \cdot K_{c1} + 0.75 \cdot 0.00023 \cdot K_{c2}}{-0.3 \cdot (0.0004 \cdot 0.00023 \cdot K_{c1} \cdot K_{c2})}}, \quad (4)$$

$$\eta_T = \eta_u - \zeta_1 - \zeta_f - \zeta_1. \quad (5)$$

Heat exchanger models

Shell-and-tube heat exchangers are selected for the S-CO₂ cycle system, and the Bell-Delaware method [22] is used to

calculate heat transfer coefficients (HTCs) and pressure drops. The shell-side (heat source) HTC and pressure drop are given by:

$$\alpha_s = \alpha_i \frac{c_{p,s} G_s}{Pr_s^{2/3}} \left(\frac{\mu_s}{\mu_w} \right)^{0.14} j_c j_l j_b j_s j_r, \quad (6)$$

where j_c , j_l , j_b , j_s and j_r are correction factors accounting for combined effects of baffle cut and spacing, for baffle leakage effects, for bundle bypass flow, for variable baffle spacing in the inlet and outlet sections, and for adverse temperature gradient build-up in laminar flow:

$$\Delta P_s = \left[(N_b - 1) R_b R_l + 2 \left(1 + \frac{N_{tcw}}{N_{tcc}} \right) R_b R_s \right] \cdot \frac{2 f_s N_{tcc} G_s^2 \left(\frac{\mu_s}{\mu_w} \right)^{-0.14}}{\rho_s} + \frac{(2 + 0.6 N_{tcw}) G_w^2}{2 \rho_s} N_b R_l \quad (7)$$

The HTC of tube side with supercritical CO₂ working fluid is given by (Ptukhov-Krasnoshchekov-Protopopov correlation):

$$\alpha_t = \frac{\lambda}{d_i} \frac{(f/8) Re Pr}{\left[12.7 (f/8)^{0.5} (Pr^{2/3} - 1) + 1.07 \right]} \cdot \left(\frac{\bar{c}_p}{c_{p,bulk}} \right)^{0.35} \cdot \left(\frac{\lambda_{bulk}}{\lambda_{wall}} \right)^{-0.33} \cdot \left(\frac{\mu_{bulk}}{\mu_{wall}} \right)^{0.11} \quad (8)$$

The total HTC of heat exchanger is given by:

$$\frac{1}{U} = \frac{1}{\alpha_t} \cdot \frac{d_o}{d_i} + r_{ft} \cdot \frac{d_o}{d_i} + \frac{\delta_w}{\lambda_w} \cdot \frac{d_o}{d_m} + r_{fs} + \frac{1}{\alpha_s} \quad (9)$$

The heat exchanger area is given by:

$$A = \frac{Q}{U \cdot \Delta T} \quad (10)$$

where ΔT is the log mean temperature difference (LMTD) between the hot side and the cold side.

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 [23]. Therefore, the specific investment cost (SIC) is calculated by Eqs. (10)-(14) with the coefficients for each component summarised in Ref. [15].

$$C_{BM} = C_p^0 F_{BM} = C_p^0 (B_1 + B_2 F_M F_p), \quad (11)$$

$$\log(C_p^0) = K_1 + K_2 \log(X_i) + K_3 [\log(X_i)]^2, \quad (12)$$

$$\log(F_p^0) = C_1 + C_2 \log(p_i) + C_3 [\log(p_i)]^2, \quad (13)$$

$$C = \sum_i C_{BM} \frac{CEPCI_{2017}}{CEPCI_{2001}}, \quad (14)$$

$$SIC = \frac{C}{W_{net}}, \quad (15)$$

where i denotes different components, $CEPCI_{2001} = 397.0$ and $CEPCI_{2017} = 567.5$ [24], which are dimensionless numbers employed to updating capital cost required to erect a power-cycle system from a past date to a later time.

The levelised cost of electricity (LCOE), which represents the present value of electricity production price considering the

economic lifetime of a system and the costs incurred in the construction, operation and maintenance, is given by:

$$LCOE = \frac{C + \sum_{n=1}^N \frac{C_{O\&M}}{(1+i)^n}}{\sum_{n=1}^N \frac{P}{(1+i)^n}}, \quad (16)$$

where C is the initial investment cost of the system, $C_{O\&M}$ is the annual operating and maintenance cost (set as 1.65% of the initial investment cost [25]), i is the discount rate (considered as 5% [25]), P is the annual power generated by the system, with an annual operation hours (set as 8000 h) and N is the expected lifetime (totally 15 years in this study).

Conditions and assumptions

An ICE with a rated power of 1000 kW [26] is selected in this paper and the heat source conditions corresponding to engine load are shown in Table 1. The temperature of engine exhaust gases changes from 540 °C to 470 °C while the mass flow rate decreases from 1.56 kg/s to 0.72 kg/s when the engine condition changes from the rated one (100%) to a part-load of 40%, which indicates significant difference among the heat source conditions.

Table 1. Exhaust gases conditions corresponding ICE load.

Load	100%	90%	80%	70%	60%	50%	40%
Temperature (°C)	540	532	530	527	525	515	470
Mass flow rate (kg/s)	1.56	1.41	1.23	1.10	0.99	0.86	0.72

The model of the S-CO₂ for ICE waste-heat recovery used in this study was developed using in-house MATLAB codes, with working fluid properties acquired from NIST REFPROP [27]. The interior-point algorithm in MATLAB's *fmincon* function has been chosen as the solver to maximise the net power output under each heat source condition.

The main conditions and assumptions are given below:

(1) The minimum operating temperature and pressure of the S-CO₂ cycle system are 31.5 °C and 7.5 MPa, respectively, to ensure that the working fluid is in the supercritical state; the maximum operating pressure is 20 MPa.

(2) The pinch point temperature differences in all heat exchangers in the design stage are set to 10 °C.

(3) The compressor efficiency is assumed to be 0.8 at both design and off-design conditions, which will be explored in future work by implementing detailed design and off-design models.

(4) The heat sink (cooling water) temperature is set to 15 °C and the maximum mass flow rate of the heat sink is 10 kg/s.

(5) The temperature of engine exhaust gases after heat recovery is set to be >120 °C in order to avoid acid corrosion in the corresponding pipes and heat exchangers [28].

(6) Heat losses throughout the system are neglected.

(7) All processes take place at steady state conditions.

RESULTS AND DISCUSSION

Cycle parameters as well as turbine design parameters are optimised simultaneously in the design stage to achieve the maximum net power output under each heat source condition corresponding to different ICE operating load. In other words, separate design schemes in terms of cycle parameters and components are obtained for all possible heat source conditions shown in Table 1. The maximum net power output and the corresponding *SIC* are shown in Figure 3, indicating that performance of the optimal designs (from a thermodynamic perspective) is closely related to the given heat source conditions. More precisely, the maximum net power output obtained from the separate designs decreases from 166 kW to 59 kW when the ICE load changes from the rated value (100%) to a part-load of 40%, while the *SIC* increases from 4630 \$/kW to 8190 \$/kW.

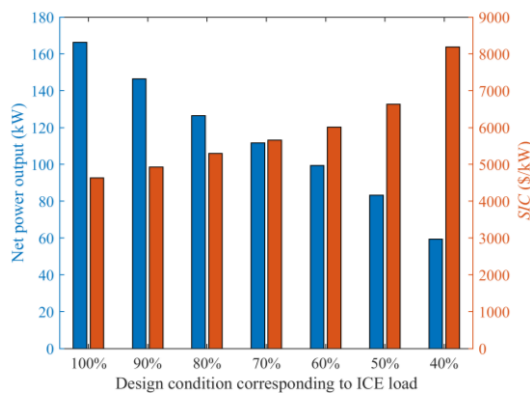


Figure 3: Net power output and corresponding *SIC* of the separate designs of S-CO₂ cycle system for ICE waste-heat recovery. Cycle parameters and turbine design parameters are optimised simultaneously to achieve the maximum net power output under each heat source condition corresponding to different ICE load shown in Table 1.

Detailed information of the separate designs for each heat source condition including turbine inlet and outlet conditions, CO₂ mass flow rate, areas of heat exchangers (recuperator, main heater and pre-cooler) and turbine efficiency are shown in Figs. 4-7, respectively. It can be seen from Figure 4 that the optimal turbine inlet and outlet pressures of the separate designs are nearly the same under different design conditions for the S-CO₂ cycle system to deliver the maximum net power output, while the optimised turbine inlet temperature decreases with the heat source temperature. Figure 5 shows that CO₂ mass flow rate corresponding to the separate designs decreases with the thermal load of the heat sources. It can be seen from Figure 6 that the main heater dominates in all designs as the density as well as the heat transfer coefficient of exhaust gases is much lower than the CO₂ working fluid, which also confirms that system compactness can be achieved as an advantage of CO₂-based systems thanks to the working fluid properties. Moreover, although the total heat exchanger area is larger with higher ICE load as well as more heat available from the engine exhaust

gases, the corresponding *SIC* is still lower due to the high power output (see Figure 3). Figure 7 shows that the turbine efficiency is within the range of 84% to 86%, indicating the optimisation enables high-efficiency design under various heat source and operating conditions.

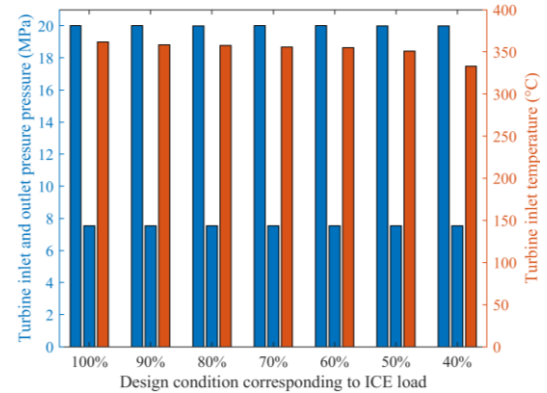


Figure 4: Turbine inlet and outlet conditions of the separate designs for each heat source condition corresponding to different ICE load.

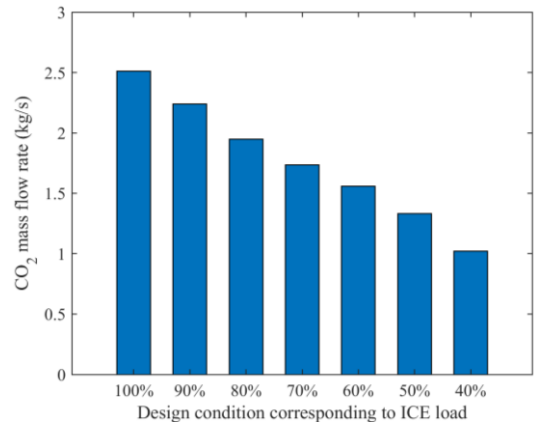


Figure 5: CO₂ mass flow rate of the separate designs for each heat source condition corresponding to different ICE load.

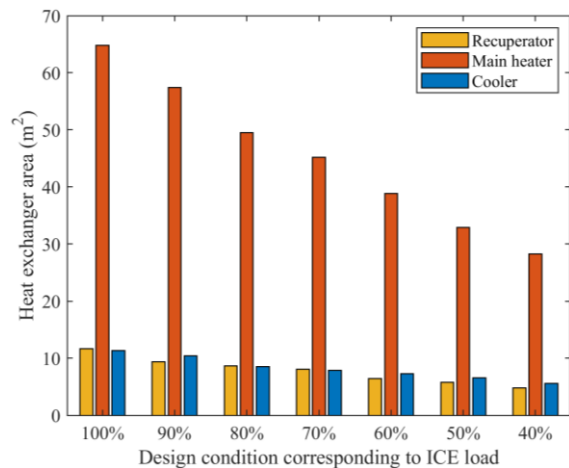


Figure 6: Heat exchanger areas of the separate designs for each heat source condition corresponding to different ICE load.

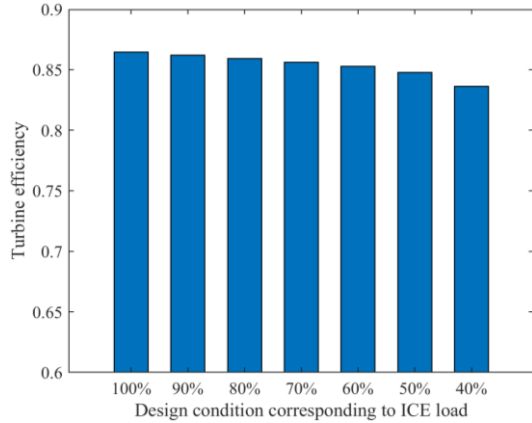


Figure 7: Turbine efficiency of the separate designs for each heat source condition corresponding to different ICE load.

In order to select the optimal design among the schemes shown above, off-design performance of each separate design under all possible heat source conditions are evaluated via detailed off-design models. System operating parameters in the off-design stage are also optimised to achieve the maximum power output, which corresponds to the optimal control strategy in practical applications. Figure 8 shows that all separate designs experience a decrease in power output when the ICE load changes to be lower as the heat input to the S-CO₂ cycle system decreases. The design scheme for the rated condition (100% ICE load) provides a higher net power output under most heat source conditions, while the difference becomes smaller with lower ICE load.

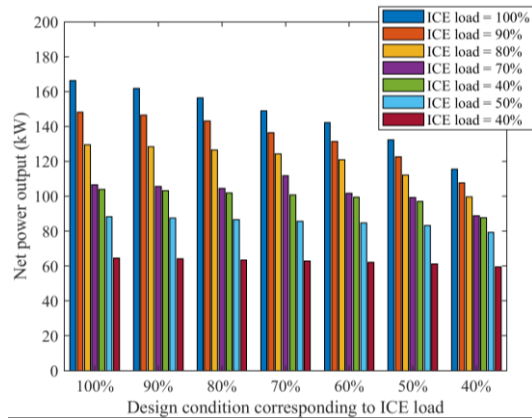


Figure 8: Off-design performance of the separate designs under all possible heat source conditions.

The seven ICE operating conditions listed in Table 1 are considered as actual variations and the probability of occurrence of the ICE operating conditions will vary with different end-users as well as demand profile. Two cases are assumed to indicate the variations, i.e., one considers the same probability of occurrence for all possible conditions (equal-weighted scenario) and the other accounts for different weights for all the conditions as reported in

Ref. [29], with 20.6%, 18.3%, 16.2%, 14.1%, 12.5%, 10.6% and 7.7% (different-weighted scenario), respectively.

It can be seen that the design corresponding to ICE rated load (100%) yields the maximum annual power output under both scenarios (922 MWh and 1330 MWh, respectively), and outperforms other designs significantly under the different-weighted scenario. The corresponding *LCOE* of the design at the rated heat source condition is the highest under the equal-weighted scenario (94 \$/MWh), while it is the lowest under the different-weighted scenario (65 \$/MWh). Therefore, the design at rated heat source condition is the optimal under the different-weighted scenario in terms of both thermodynamic and economic perspectives, while under the equal-weighted scenario, the design at the rated heat source condition is the optimal from the thermodynamic perspective, and the design corresponding to ICE load of 40% has the lowest *LCOE* of 75/MWh.

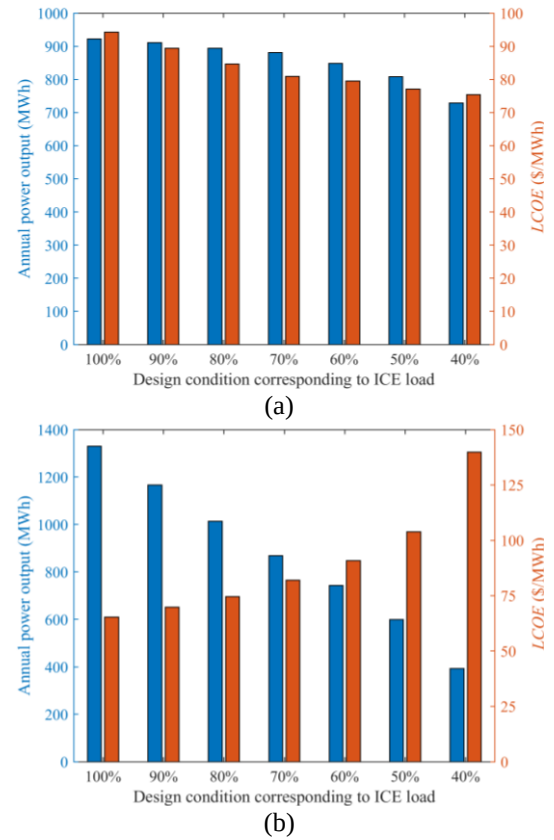


Figure 9: Annual power output and *LCOE* of separate design for each heat source condition corresponding different ICE load: (a) under equal-weighted scenario, and (b) under different-weighted scenario.

CONCLUSIONS

In this paper we explore the optimal design of S-CO₂ cycle systems for ICE waste-heat recovery considering heat source fluctuations arising from the ICE operating load. Separate designs are presented for each heat source condition to achieve the maximum net power output. The design corresponding to the ICE rated load (100%) provides the maximum net power output of

166 kW with a corresponding *SIC* of 4630 \$/kW, which is also the lowest among all design schemes. The off-design performance of all separate designs is evaluated at all possible heat source conditions and the design scheme for the ICE rated load is found to provide a higher net power output under most conditions. Two scenarios, i.e., equal-weighted and different-weighted, are considered in order to estimate the annual performance of the S-CO₂ cycle system. For the equal-weighted scenario, the design for the rated load ICE condition is the optimal from the thermodynamic perspective and the maximum annual power output reaches 922 MWh. The design for the 40% ICE load condition yields the lowest *LCOE* of 75 \$/MWh. The design for the rated load condition is the optimal for the different-weighted scenario with a maximum annual power output of 1330 MWh and the lowest *LCOE* of 65 \$/MWh. The advantage of the methodology presented herein relative to conventional approaches that only consider one specific design condition is that it enables us to avoid either over- or under-sizing of the S-CO₂ cycle system, which also allows us to gain a more complete understanding of the interplay between the bottoming heat recovery system and the ICE, while providing valuable guidance for further system optimisation and operation.

NOMENCLATURE

Symbols

B_i, C_i, K, F_M, F_P	constants for cost models
C	cost (\$)
c_p	specific heat capacity (J/kg·K)
d, D	diameter (m)
f	friction factor
j, R	correction factors
K_{c1}, K_{c2}	leakage loss factor
K_f	friction loss factor
G	mass flux (kg/m ² ·s)
h	enthalpy (J/kg)
i	discount rate
m	mass flow rate (kg/s)
N	number, lifetime (year)
p	pressure (Pa)
P	power (kW·h)
Pr	Prandtl number
Q	thermal load (W)
q	heat flux (W·m ²)
Re	Reynolds number
r_f	fouling resistance
R_p	surface roughness (m)
s	entropy (J/kg)
T	temperature (K)
u	peripheral velocity (m/s)
U	heat transfer coefficient (W/m ² ·K)
w	relative velocity (m/s)
W	power output (W)
X	component capacity

Greek symbols

α	heat transfer coefficient (W/m ² ·K)
α_1	absolute flow angle in rotor inlet (°)
β_1	relative flow angle in rotor inlet (°)
β_2	relative flow angle in rotor outlet (°)
η	efficiency
λ	thermal conductivity (W/m·K)
μ	viscosity (Pa/s)
ρ	density (kg/m ³)
ζ	loss factor
φ	nozzle velocity coefficient
ψ	rotor velocity coefficient
Ω	reaction degree

Subscripts

b	bundle bypass flow; baffle
BM	bare module
$bulk$	bulk temperature of fluid
f	friction loss
i	component
I	incidence loss
l	leakage loss
n	net power output
opt	optimal
$O\&M$	operating and maintenance
s	shell side, isentropic
t	tube side
T	turbine
tcc	tube rows in crossflow
tcw	tube rows in baffle window
u	peripheral
w	wall

Abbreviations

<i>CEPCI</i>	chemical engineering plant cost index
<i>HTC</i>	heat transfer coefficient
<i>ICE</i>	internal combustion engine
<i>LCOE</i>	levelised cost of electricity
<i>LMTD</i>	log mean temperature difference
<i>ORC</i>	organic Rankine cycle
<i>S-CO₂</i>	supercritical CO ₂
<i>SIC</i>	specific investment cost

ACKNOWLEDGEMENTS

This work was supported by the UK Engineering and Physical Sciences Research Council (EPSRC) [grant numbers EP/P004709/1, and EP/R045518/1]. Data supporting this publication can be obtained on request from cep-lab@imperial.ac.uk.

REFERENCES

- [1] Shu G, Zhao M, Tian H, Huo Y, Zhu W. Experimental comparison of R123 and R245fa as working fluids for waste heat recovery from heavy-duty diesel engine. *Energy* 2016;115:756–69.
- [2] Song J, Song Y, Gu C. Thermodynamic analysis and performance optimization of an organic Rankine cycle

- (ORC) waste heat recovery system for marine diesel engines. *Energy* 2015;82:976-85.
- [3] Li X, Song J, Yu G, Liang Y, Tian H, Shu G, Markides CN. Organic Rankine cycle systems for engine waste-heat recovery: Heat exchanger design in space-constrained applications. *Energy Convers Manag* 2019;199:111968.
 - [4] Le Brun N, Simpson M, Acha S, Shah N, Markides CN. Techno-economic potential of low-temperature, jacket-water heat recovery from stationary internal combustion engines with organic Rankine cycles: A cross-sector food-retail study. *Appl Energy* 2020;274:115260.
 - [5] Chatzopoulou MA, Markides CN. Thermodynamic optimisation of a high-electrical efficiency integrated internal combustion engine–Organic Rankine cycle combined heat and power system. *Appl Energy* 2018;226:1229-51.
 - [6] Song J, Li X, Ren X, Tian H, Shu G, Gu C, Markides CN. Thermodynamic and economic investigations of transcritical CO₂-cycle systems with integrated radial-inflow turbine performance predictions. *Appl Therm Eng* 2020;165:114604.
 - [7] Li X, Tian H, Shu G, Zhao M, Markides CN, Hu C. Potential of carbon dioxide transcritical power cycle waste-heat recovery systems for heavy-duty truck engines. *Appl Energy* 2019;250:1581-99.
 - [8] Iverson BD, Conboy TM, Pasch JJ, Kruijenga AM. Supercritical CO₂ Brayton cycles for solar-thermal energy. *Appl Energy* 2013;111:957-70.
 - [9] Li M, Zhu H, Guo J, Wang K, Tao W. The development technology and applications of supercritical CO₂ power cycle in nuclear energy, solar energy and other energy industries. *Appl Therm Eng* 2017;126:255-75.
 - [10] Ahn Y, Bae SJ, Kim M, Cho SK, Baik S, Lee JI, Cha JE. Review of supercritical CO₂ power cycle technology and current status of research and development. *Nuc Eng Technol* 2015;47:647-61.
 - [11] Ahmadi MH, Mehrpooya M, Pourfayaz F. Exergoeconomic analysis and multi objective optimisation of performance of a Carbon dioxide power cycle driven by geothermal energy with liquefied natural gas as its heat sink. *Energy Convers Manag* 2016;119:422-34.
 - [12] Sarkar J. Review and future trends of supercritical CO₂ Rankine cycle for low-grade heat conversion. *Renew Sust Energy Rev* 2015;48:434-51.
 - [13] Chen Y, Lundqvist P, Platell P. Theoretical research of carbon dioxide power cycle application in automobile industry to reduce vehicle's fuel consumption. *Appl Therm Eng* 2005;25:2041-53.
 - [14] Song J, Li X, Ren X, Gu C. Performance improvement of a preheating supercritical CO₂ (S-CO₂) cycle based system for engine waste heat recovery. *Energy Convers Manag* 2018;161:225-33.
 - [15] Song J, Li X, Wang K, Markides CN. Parametric optimisation of a combined supercritical CO₂ (S-CO₂) cycle and organic Rankine cycle (ORC) system for internal combustion engine (ICE) waste-heat recovery. *Energy Convers Manag* 2020;218:112999.
 - [16] Hou S, Wu Y, Zhou Y, Yu L. Performance analysis of the combined supercritical CO₂ recompression and regenerative cycle used in waste heat recovery of marine gas turbine. *Energy Convers Manag* 2017;151:73-85.
 - [17] Sharma OP, Kaushik SC, Manjunath K. Thermodynamic analysis and optimisation of a supercritical CO₂ regenerative recompression Brayton cycle coupled with a marine gas turbine for shipboard waste heat recovery. *Therm Sci Eng Prog* 2017;3:62-74.
 - [18] Wasserbauer CA, Glassman AJ. Fortran program for predicting off-design performance of radial-inflow turbines. NASA Technical Note 1975.
 - [19] Glassman AJ. Computer program for design analysis of radial-inflow turbines. NASA Technical Note 1976.
 - [20] Da Lio L, Manente G, Lazzaretto A. A mean-line model to predict the design efficiency of radial inflow turbines in organic Rankine cycle (ORC) systems. *Appl Energy* 2017;205:187-209.
 - [21] Zhou A, Song J, Li X, Ren X, Gu C. Aerodynamic design and numerical analysis of a radial inflow turbine for the supercritical carbon dioxide Brayton cycle. *Appl Therm Eng* 2018;132:245-55.
 - [22] K. Thulukkanam. Heat exchanger design handbook. Second Edition. CRC Press, 2013.
 - [23] Turton R, Bailie RC, Whiting WB, Shaeiwitz JA. Analysis, synthesis and design of chemical processes. Third Edition. Pearson Education, 2008.
 - [24] www.chemengonline.com/cepci-updates-january-2018-prelim-and-december-2017-final
 - [25] Shu G, Shi L, Tian H, Deng S, Li X, Chang L. Configurations selection maps of CO₂-based transcritical Rankine cycle (CTRC) for thermal energy management of engine waste heat. *Appl Energy* 2017;186:423-35.
 - [26] Wang X, Shu G, Tian H, Feng W, Liu P, Li X. Effect factors of part-load performance for various organic Rankine cycles using in engine waste heat recovery. *Energy Convers Manag* 2018;174:504-15.
 - [27] NIST. Reference Fluid Thermodynamic and Transport Properties Database (REFPROP), 2016.
 - [28] Bahadori A. Estimation of combustion flue gas acid dew point during heat recovery and efficiency gain. *Appl Therm Eng* 2011;31:1457-62.
 - [29] Li X, Shu G, Tian H. Integrating off-design performance in designing CO₂ power cycle systems for engine waste heat recovery. *Energy Convers Manag* 2019;201:112146.