This is a peer-reviewed, accepted author manuscript of the following research article: Koroglu, T, Cavus, I & Bayer, MU 2023, 'Marine engine efficiency improvement with supercritical-CO2 Rankine waste heat recovery', International Journal of Exergy, vol. 40, no. 3, pp. 347-364. https://doi.org/10.1504/IJEX.2023.129801

# Marine Engine Efficiency Improvement with Supercritical-$CO_2$ Rankine Waste Heat Recovery

Turgay Koroglu*, Irfan Cavus, Muhammed Umar Bayer

Department of Naval Architecture and Marine Engineering

Bandırma Onyedi Eylul University, 10200 Balıkesir-Türkiye

Email: tkoroglu@bandirma.edu.tr, ORCiD:0000-0001-9109-9066

Email: icavus@bandirma.edu.tr, ORCiD: 0000-0002-2537-7141

Email: mbayer@bandirma.edu.tr, ORCiD: 0000-0001-7801-7433

*Corresponding author

**Abstract:** The efficiency of marine energy systems gain more and more importance considering economic and environmental effects. Additional power is produced by annexation of a supercritical $CO_2$ Rankine cycle (s$CO_2$-RC) via utilization of the exhaust of a marine engine. A parametric study on s$CO_2$-RC is carried out to optimize objective functions such as ECOP, maximum net power output with respect to the outlet temperature of the exhaust stack, and the maximum pressure of the cycle. Then, energy and exergy analyses are applied to the system. Results show that the s$CO_2$-RC system improves thermal efficiency by 8.17% and provides a 7.54% better fuel economy, while exergy efficiency of the s$CO_2$-RC is 51.3% with a net power output of 321.7 kW and ECOP of 1.09. Hence, the results lead to the optimization order of the investigated system components for the improvement in overall efficiency, and the reduction of fuel consumption and environmental effects.

**Keywords**: Supercritical $CO_2$ Rankine cycle, Waste heat recovery, Exergy, Environmental effects, Fuel economy, ECOP

**Biographical Notes:** Turgay Koroglu is an Assistant Professor at Department of Naval Architecture and Marine Engineering in Bandirma Onyedi Eylul University. He received his BSc degree in Naval Architecture and Marine Engineering from Yildiz Technical University Istanbul, Türkiye and his PhD degree from Istanbul Technical University, Istanbul, Türkiye. He was a researcher in Marine and Maritime Institute, University of Southampton, UK as well as in the Institute of Chemical Engineering and Environmental Technology, Graz University of Technology, Austria. His areas of interest contain thermodynamic and exergy optimization of conventional and renewable energy systems, interactions of marine environment and marine energy systems, and thermo-economic optimization.

Irfan Cavus is a Research Assistant at Department of Naval Architecture and Marine Engineering in Bandirma Onyedi Eylul University. He graduated with BSc degree in Mechanical Engineering from Middle East Technical University, Ankara, Türkiye. He is an MSc student in Ankara Yildirim Beyazit University, Turkiye. His research interest is thermodynamics focusing on exergy analysis.

Muhammed Umar Bayer is a Research Assistant at Department of Naval Architecture and Marine Engineering in Bandirma Onyedi Eylul University. He graduated with MSc as well as BSc degree in Mechanical Engineering from Ankara Yildirim Beyazit University, Türkiye. He is a PhD student in the same university. His research interests are thermodynamic optimization and exergy-based analyses.

**Nomenclature**

**Abbreviations**

| | |
|---|---|
| BC | Brayton cycle |
| $CO_2$ | Carbon dioxide |
| ECOP | Ecological coefficient of performance |
| EES | Engineering equation solver |
| IMO | International Maritime Organization |
| LHV | Lower heating value (kJ) |
| LNG | Liquefied natural gas |
| MESMA | Autonomous submarine energy module (Module d'Energie Sous-Marin Autonome) |
| ORC | Organic Rankine cycle |
| PR-CTRC | $CO_2$ based transcritical Rankine cycle with preheater and regenerator |
| RC | Rankine cycle |
| $sCO_2$/SRC | Supercritical $CO_2$ and Steam Rankine combined cycle |
| $sCO_2$-RC | Supercritical $CO_2$ Rankine cycle |
| SFC | Specific fuel consumption (g/kWh) |
| SRC | Steam Rankine cycle |
| $tCO_2$-RC | Transcritical $CO_2$ Rankine cycle |
| WHR | Waste heat recovery |

**Symbols**

| | |
|---|---|
| $\dot{Q}$ | Heat (kW) |
| $\dot{W}$ | Power (kW) |
| $\dot{m}$ | Mass flow rate (kg/s) |
| $h$ | Specific enthalpy (kj/kg) |
| $\dot{E}$ | Energy (kW) |
| $\dot{E}x$ | Exergy (kW) |
| $T$ | Temperature (°C) |
| $s$ | Specific entropy (kJ/kgK) |
| $\dot{S}$ | Entropy (kW/K) |
| $\Delta$ | Difference |

**Greek Letter**

| | |
|---|---|
| ε | Exergy efficiency |
| $\eta$ | Energy efficiency |

**Subscripts**

| | |
|---|---|
| *o* | Dead (environmental) state |
| *out* | Outlet |
| *in* | Inlet |
| *net* | Net |
| f | Fuel |
| F | Exergy of fuel |
| D | Exergy destruction |
| P | Exergy of product |
| *i* | Stream *i* |
| *gen* | Generation |
| *tot* | Total |
| *th* | Thermal |
| *max* | Maximum |
| *pp* | Pinch point |

**Superscripts**

| | |
|---|---|
| *Q* | Heat flow |
| *W* | Work flow |

# 1 Introduction

Marine pollution is a driving force to decrease the emissions of a marine engine. There are several regulations and laws, which aim to lower the harm to the marine environment, put into action. Moreover, fuel prices rise and are expected to rise for a foreseeable future. Exhaust gas emissions of marine engines cause not only economic issues due to the unusable high energy content but also environmental issues due to the high temperature of released gases. To utilize residual energy and decrease the temperature of exhaust gases, several methods are applied. Among all, waste heat recovery (WHR) has become a more frequently used method to suppress economic and environmental concerns (İbrahim et al., 2020;Güneş and Karakurt, 2015;Koroglu and Sogut, 2018;Chen et al., 2006;Glover et al., 2015;Song et al., 2020). For more effective waste heat recovery, working fluid is one of the main criteria. Working fluid selection depends on parameters such as temperature and mass flow rate of exhaust gases; and output parameters such as net power output, system efficiency, and economic factors of the WHR system (Cengel and Boles, 2015). Besides, among the conventional working fluids such as steam and organic fluids, $CO_2$ comes forward as a novel working fluid with unique advantages such as economic benefits and simplicity (Karakurt, 2020;Karakurt et al., 2021;Kim et al., 2017).

Several studies have been done to examine the performance of WHR systems, which utilize $CO_2$ as a working fluid (Zhang et al., 2018;Liang et al., 2019;Bae et al., 2014;Yang et al., 2020;Mohammadi et al., 2020;Pan et al., 2020;Xu et al., 2019).

Song et al. (2020) conducted a study to examine the thermodynamic and economic performance of a combined supercritical $CO_2$ ($sCO_2$) cycle and organic Rankine cycle (ORC) system. The designed combined system consists of not only a topping $sCO_2$ cycle system that utilizes the waste heat of the engine jacket water and exhaust gases but also a bottoming ORC system that harvests the heat energy rejected from the topping cycle. As a result, it has been obtained that the combined system enhanced the power output up to 70% by the annexation of the ORC cycle to the $sCO_2$ cycle. Glover et al. (2015) worked on the performance optimization of a supercritical ORC WHR system regarding appropriate working fluid selection and operational parameters' effects on the efficiency and power output. It has been concluded that system efficiency of up to 5-23% can be achieved for the specified conditions. Moreover, fuel economy has improved between 10-30%. Sarkar (2015) presented an extensive and in-depth review of recent research about the $sCO_2$ Rankine cycle for low-grade heat conversion. The result shows that the $sCO_2$ Rankine cycle will be preferable more than the $sCO_2$ Brayton cycle for low-grade heat sources due to the better temperature matching of the researched system between the heat source and the working fluid. Chen et al. (2006) compared the transcritical $CO_2$ cycle that utilizes low-grade waste heat energy to an ORC, which uses R123 as a working fluid. It has been concluded that the transcritical $CO_2$ cycle contributes more power output than ORC with R123 for low-grade heat sources. Manente and Fortuna (2019) have designed three novel types of $sCO_2$ layouts to utilize heat sources as high as possible: dual recuperation, partial heating, and dual expansion, namely. They have proposed that, at 600 °C waste heat source temperature, 15%-3% more power generation is possible by using a single flow split with a dual expansion cycle compared to dual recuperated and partial heating cycles, respectively. Also, the mentioned cycle provides 40% more power than a traditional single recuperated cycle. Zhou et al. (2020) have modified the conventional $sCO_2$/$tCO_2$-RC (top and bottoming cycles, respectively) combined layout by using the top cycle's residual heat in the bottoming cycle; and adding a split flow branch to the $tCO_2$ cycle. According to the results, the modified $sCO_2$/$tCO_2$ layout provides 10.53% more net power output than the conventional one. Shu et al. (2016) have improved conventional $tCO_2$-RC by adding a preheater and regenerator to the basic system design. They compared the designed system with conventional $tCO_2$-RC (C/$tCO_2$-RC) and ORC. Their results show that the designed system provides 150% more net power output than C/$tCO_2$-RC. Moreover, they concluded that their design is better than ORC in terms of recycling heat energy from cooling water and exhaust gases. Chaudhary et al. (2018) have compared three WHR systems; $sCO_2$-RC, SRC, $sCO_2$/SRC (top and bottoming cycles). They have compared these systems in terms of net power generation and waste heat losses. $sCO_2$-RC shows the highest loss of waste heat as 40%. As the combined $sCO_2$/SRC cycle provides the highest net power generation, $sCO_2$-RC cycle provides very low net power generation.

Ecological coefficient of performance (ECOP) is a key parameter to determine better performance criteria in terms of entropy generation, thermal efficiency, and investment cost (Ust et al., 2005). Ust et al. (2016) have been conducted a performance optimization based on ECOP. Their results show that a design at $ECOP_{max}$ provides lower entropy generation, higher thermal efficiency, and lower investment cost. However, it leads to lower power output than a design at $\dot{W}_{\max}$ condition (Ust et al., 2006). Wu et al. (2021) analyzed an irreversible Diesel cycle in terms of ECOP among other functions. They stated that ECOP and maximum thermal efficiency both decrease, and net power output firstly increases and then decreases with increasing piston speed ratio. Ust et al. (2016) conducted an ECOP-based performance optimization on regenerative and cogeneration gas turbine systems to determine the optimal design. Their results show that, at maximum ECOP conditions, the optimal pressure ratio is higher than the optimal exergy efficiency, and power output conditions. Gonca (2018) conducted an optimization study on a Rankine cycle via exergy efficiency, ECOP, and effective power. He stated that component pressures are significantly effective on system performance;

increasing condenser pressure leads ECOP, power output, and exergy efficiency to decrease. Recently, ECOP has been applied to marine systems to optimize the system and its performance. Ozsari (2022) conducted a thermodynamic analysis on MESMA system which is used as a propulsion system in submarines. The system is examined in terms of ECOP to determine environmentally friendly operating conditions. He proposed that the methanol fuel is better than ethanol in terms of ECOP. Karakurt et al. (2022). carried out a study on the pressurized air starting system and utilized ECOP to compare three different system models and stated that the first model has the highest ECOP value due to its low exergy destruction.

In this study, LNG powered Rolls-Royce BL35:40L9PG 9-cylinder marine engine with 3940 kW power is considered (Rolls-Royce, 2016). Obtaining more power output by recovering the heat energy of exhaust gases is the main objective of the study to reach the aim of decreasing both the environmental and economic burden of marine engine fuel consumption to imply the IMO regulations. The literature review reveals that the $sCO_2$-RC is more compact and efficient in low-grade waste heat recovery. Moreover, ships are limited in space as well as simple systems are important to operate and maintain onboard. Furthermore, very limited studies have been found on the application of the $sCO_2$-RC onboard ships. Therefore, a simple supercritical $CO_2$ Rankine cycle system is designed with a regenerator to transfer the heat of the expander outlet gases into pump/compressor outlet gases to make the system more efficient as well as to utilize as high as possible from the energy, which is given by the exhaust gases to the WHR system. The system is optimized to deliver not only the maximum power output but also the highest efficiency. Afterward, the exergy analysis is applied to reveal how much of the exergy is released to the atmosphere before and after the addition of the WHR system. Lastly, exergy destruction of each component regarding the $sCO_2$ Rankine WHR system is calculated with fuel and product exergy, and exergy efficiency. Results could be used to improve the system as well as to eliminate the economic and environmental concerns.

## 2 Method

Analyzing the thermal energy conversion systems is important to decide whether a system should be considered an efficient design. The first law of thermodynamics is utilized to determine the energy efficiency of a system. According to the first law of thermodynamics, if the potential and kinetic energy are neglected, the conservation of energy can be formulated as follow (Cengel and Boles, 2015).

$$\dot{Q}_{net} - \dot{W}_{net} = \Sigma \dot{m}_{out} h_{out} - \Sigma \dot{m}_{in} h_{in} \tag{1}$$

The net heat transfer that enters the system and the net produced power in the system are denoted as $\dot{Q}_{net}$ and $\dot{W}_{net}$, respectively, while $\dot{m}$ is mass flow rate, $h$ is specific enthalpy, and subscripts *in* and *out* denote inlet and outlet, respectively.

The energy of stream $i$, $\dot{E}_i$, could be calculated as (Cengel and Boles, 2015),

$$\dot{E}_i \cong \dot{m}_i h_i \tag{2}$$

Total energy efficiency, $\eta_{tot}$, of the system is the indicator of how much of the given heat is transferred into the net power output, $\dot{W}_{net}$, and shown by (Bejan et al., 1995),

$$\eta_{tot} = \dot{W}_{net}/\dot{Q}_{in} \tag{3}$$

However, the first law does not help to reveal the usable information of a system to be improved. Evaluating the first and the second laws of thermodynamics together would bring exergy analysis to the table. Exergy analysis is a well-known method to reveal the reasons, positions, and amounts of the irreversibilities within the system. The exergy of stream $i$, $\dot{E}x_i$, which is defined as a maximum useful work during a thermodynamic process with its environment, denoted as "$_0$", can be calculated as below (Bejan et al., 1995).

$$\dot{E}x_i = \dot{m}_i ex_i \cong \dot{m}_i[(h_i - h_0) - T_0(s_i - s_0)] \tag{4}$$

$ex_i, s$ and $T$ are specific exergy, specific entropy, and temperature, respectively. Exergy destruction is the result of why the exergy is not conserved due to the inefficiencies within the investigated system during the processes. The heat exergy and work are denoted by $\dot{E}x^Q$ and $\dot{E}x^W$, respectively. The balance equation is given by (Bejan et al., 1995),

$$\dot{E}x^Q - \dot{E}x^W = \Sigma\dot{m}_{out}ex_{out} - \Sigma\dot{m}_{in}ex_{in} + \Sigma\dot{E}x_D \tag{5}$$

$$\dot{E}x_F = \dot{E}x_P + \dot{E}x_D \tag{6}$$

Exergy of fuel, product, and destruction are represented as $\dot{E}x_F$, $\dot{E}x_P$, and $\dot{E}x_D$, respectively. Exergy rates of fuel and product for each component could be investigated in detail in the literature (Bejan et al., 1995).

Exergy efficiency, ε, represents the amount of the exergy of fuel, which is converted into a product as a ratio (Cengel and Boles, 2015).

$$\varepsilon = \dot{E}x_F/\dot{E}x_P \tag{7}$$

The thermal efficiency of a power production system that is related to fuel consumption could be calculated as,

$$\eta_{\text{th}} = \frac{\dot{W}_{net}}{\dot{m}_f \cdot LHV} \tag{8}$$

The specific fuel consumption, SFC, of the system could be described as the consumed fuel,$\dot{m}_f$, over the power output as follow (Pulkrabek, 2007).

$$SFC = \dot{m}_f/\dot{W}_{\text{net}} \tag{9}$$

Ecological coefficient of performance (ECOP) can be defined as the ratio of net power output over total exergy destruction which could also be calculated as the product of absolute temperature, $T_0$ , and total entropy generation, $\dot{S}_{gen}$. It could also be considered an objective function for the optimization of power production systems (Ust et al., 2005). It is proposed that a design with a maximum of the ECOP objective function provides the best compromise between the power output, thermal efficiency, investment cost, and environmental effect (Ust et al., 2006).

$$ECOP = \dot{W}_{net}/T_0\dot{S}_{gen} \tag{10}$$

## 3 System Description

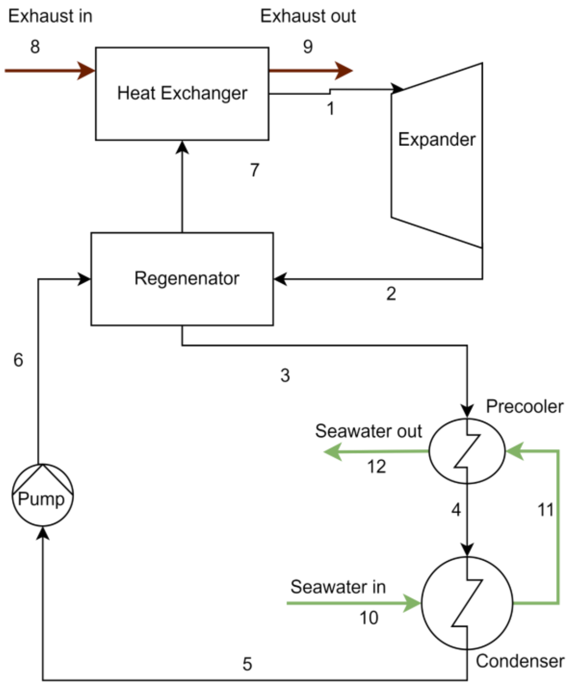
The literature reveals that the simple supercritical (transcritical) $CO_2$ Rankine cycle with a regenerator has similar efficiency in comparison with the cascade and split $sCO_2$ Rankine cycles (Kim et al., 2017). Therefore, the $sCO_2$ Rankine cycle is designed based on the simple cycle in this study and annexed to the exhaust system of Rolls-Royce B35:40L9PG, which uses natural gas as the main fuel (Rolls-Royce, 2016). Typical specifications of the mentioned main engine are given in Table 1.

**Table 1** Specifications of the investigated main engine

| | | | |
|---|---|---|---|
| Engine Speed (RPM) | 750 | Exhaust Temperature ($T_{Exh,}$ °C) | 395 |
| Power (kW) | 3940 | Exhaust Mass Flow Rate ($\dot{m}_{Exh,}$ kg/s) | 5.83 |
| Number of cylinders | 9 | Air Consumption (kg/h) | 20400 |
| Cylinder bore (mm) | 350 | Gas Consumption at MCR (kg/h) | 660 |
| Piston stroke (mm) | 400 | Turbocharger type (ABB) | TPL-65 VTG |
| Mean effective pressure (bar) | 18.2 | Lower Heating Value (MJ/Nm$^3$) | 36 |

The design of the simple $sCO_2$ Rankine cycle includes six components, namely a heat exchanger, an expander, a regenerator, a precooler, a condenser, and a pump as shown in Figure 1.

**Figure 1** Schematic of the simple $sCO_2$ Rankine cycle




The exhaust gas of the main engine enters the heat exchanger at 395 °C while transferring some of its energy to the $CO_2$ within. The exhaust gas is modeled as air since it mostly consists of air because of the high air-fuel ratio (Cengel and Boles 2015). The pressurized $CO_2$ gas gains heat and leaves the heat exchanger at 380 °C, with a pinch temperature difference of 15 °C. Afterward, the supercritical working fluid expands through the turbine and produces power to be utilized. Expanded $CO_2$ has still relatively high temperature and energy, therefore, the heat is transferred to the pumped/compressed $CO_2$. Then, $CO_2$ continues its way to pass through the precooler to cool down more before entering the condenser. In the condenser, subcritical $CO_2$ changes from saturated $CO_2$ vapor into the saturated liquid while releasing its heat into the seawater. The seawater is assumed to be taken from nature at 20 °C and modeled as water to be analyzed. The saturated liquid, then, is pumped and compressed until the high pressure of the cycle. It is important to note that while pressuring the subcritical $CO_2$, first it is liquid until the transcritical state, around the critical point. The working fluid turns into vapor, and later gas, after passing the critical point, hence it is compressed. As mentioned before, it passes through the regenerator, and energy is transferred from the outlet gas of the turbine. Lastly, the $sCO_2$ continues to the heat exchanger to complete the cycle.

For the analysis of the system, some assumptions have been made. The isentropic efficiency of the pump is assumed as 0.60. The effectiveness of the regenerator and expander isentropic efficiency are 0.90 and 0.80, respectively (Min, Sohn, and Yoon 2017). It is assumed that the pressure does not change in the heat exchanger, regenerator, precooler, and condenser due to their neglectable effect (Koroglu and Sogut, 2018). The seawater outlet is accepted as 26 °C to prevent the crossflow of the transferred heat in the condenser and the preheater.

Table 2 shows the balance equations of exergy and energy to analyze the mentioned system. The exergy balance equation of each component was designed regarding the exergy of fuel and product approach. Therefore, the left-hand side represents the exergy of fuel while the right-hand side denotes the sum of the exergy of product and the exergy destruction. Energy balances are given to conserving the energy transfer from one point to another.

**Table 2** Energy and Exergy Balance Equations

| Component | Exergy Balance Equations | Energy Balance Equations |
|---|---|---|
| Expander | $\dot{Ex}_1 - \dot{Ex}_2 = \dot{W}_T + \dot{Ex}_{D,T}$ | $\dot{E}_1 - \dot{E}_2 = \dot{W}_T$ |
| Precooler | $\dot{Ex}_3 - \dot{Ex}_4 = \dot{Ex}_{12} - \dot{Ex}_{11} + \dot{Ex}_{D,Pre}$ | $\dot{E}_3 - \dot{E}_4 = \dot{E}_{12} - \dot{E}_{11}$ |
| Condenser | $\dot{Ex}_4 - \dot{Ex}_5 = \dot{Ex}_{11} - \dot{Ex}_{10} + \dot{Ex}_{D,C}$ | $\dot{E}_4 - \dot{E}_5 = \dot{E}_{11} - \dot{E}_{10}$ |
| Pump | $\dot{W}_P = \dot{Ex}_6 - \dot{Ex}_5 + \dot{Ex}_{D,P}$ | $\dot{W}_P = \dot{E}_6 - \dot{E}_5$ |
| Regenerator | $\dot{Ex}_2 - \dot{Ex}_3 = \dot{Ex}_7 - \dot{Ex}_6 + \dot{Ex}_{D,Reg}$ | $\dot{E}_2 - \dot{E}_3 = \dot{E}_7 - \dot{E}_6$ |
| Heat Exchanger | $\dot{Ex}_8 - \dot{Ex}_9 = \dot{Ex}_1 - \dot{Ex}_7 + \dot{Ex}_{D,Hex}$ | $\dot{E}_8 - \dot{E}_9 = \dot{E}_1 - \dot{E}_7$ |

### *3.1 Validation*

The system is designed and simulated using Engineering Equation Software (EES) (Klein, 2018). In literature, there are limited studies on the trans/supercritical $CO_2$ Rankine cycle. To validate the designed system exact topology of the system could not be found due to its genuineness. However, similarities are taken into consideration to validate the design. The simulation is validated using the key data and operating conditions as in the study of Kim et al. (2017) which are given in Table 3.

**Table 3** Operating conditions of the validated study (Kim et al., 2017)

| | | | |
|---|---|---|---|
| Turbine inlet temperature (°C) | 391 | Turbine and pump isentropic efficiencies | 0.8 |
| Cycle maximum/minimum pressures (kPa) | 23000/ 5730 | Regenerator effectiveness | 0.9 |
| Exhaust mass flow rate (kg/m$^3$) | 69.8 | Exhaust inlet/outlet temperatures (°C) | 538/180 |

The result of the validation is given in Table 4. The present results of the simulation by applying the operating conditions of the Kim et al. (2017) are in good agreement according to the absolute errors which are lower than 1%.

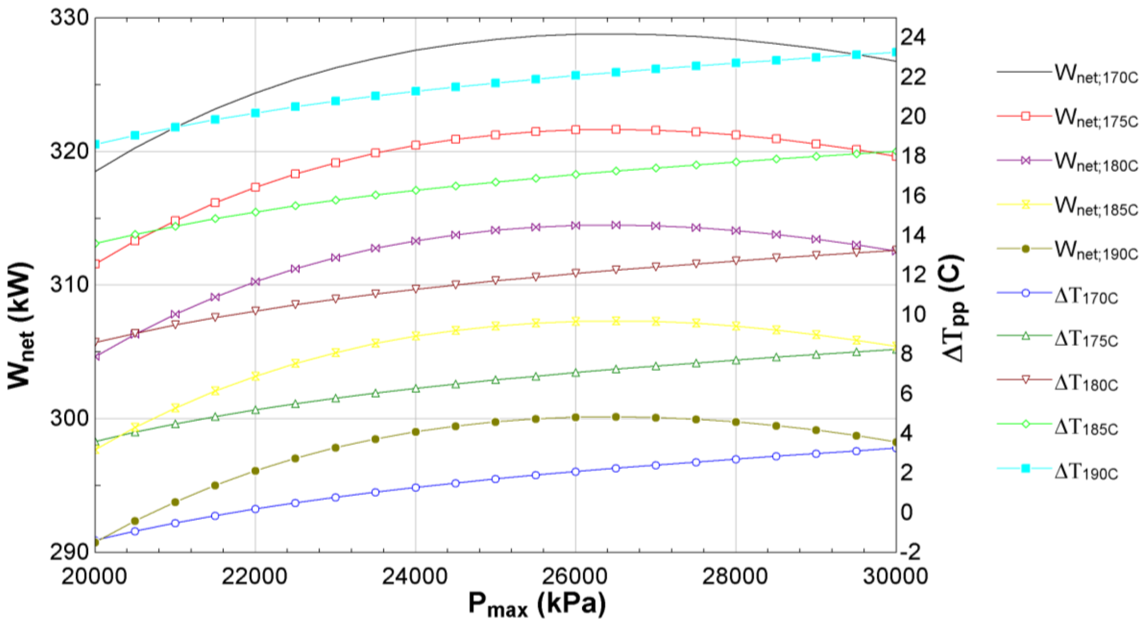
**Table 4** Comparison of the calculated results in the present paper and (Kim et al., 2017)

| | $\dot{W}_{net}$ (kW) | $\eta_{tot}$ | $\dot{Q}_{max}$ (kW) |
|---|---|---|---|
| Reference | 8129 | 0.287 | 40893 |
| Present | 8122.39 | 0.287 | 41288 |
| Absolute Error (%) | 0.08 | 0 | 0.97 |

## 4. Results and Discussion

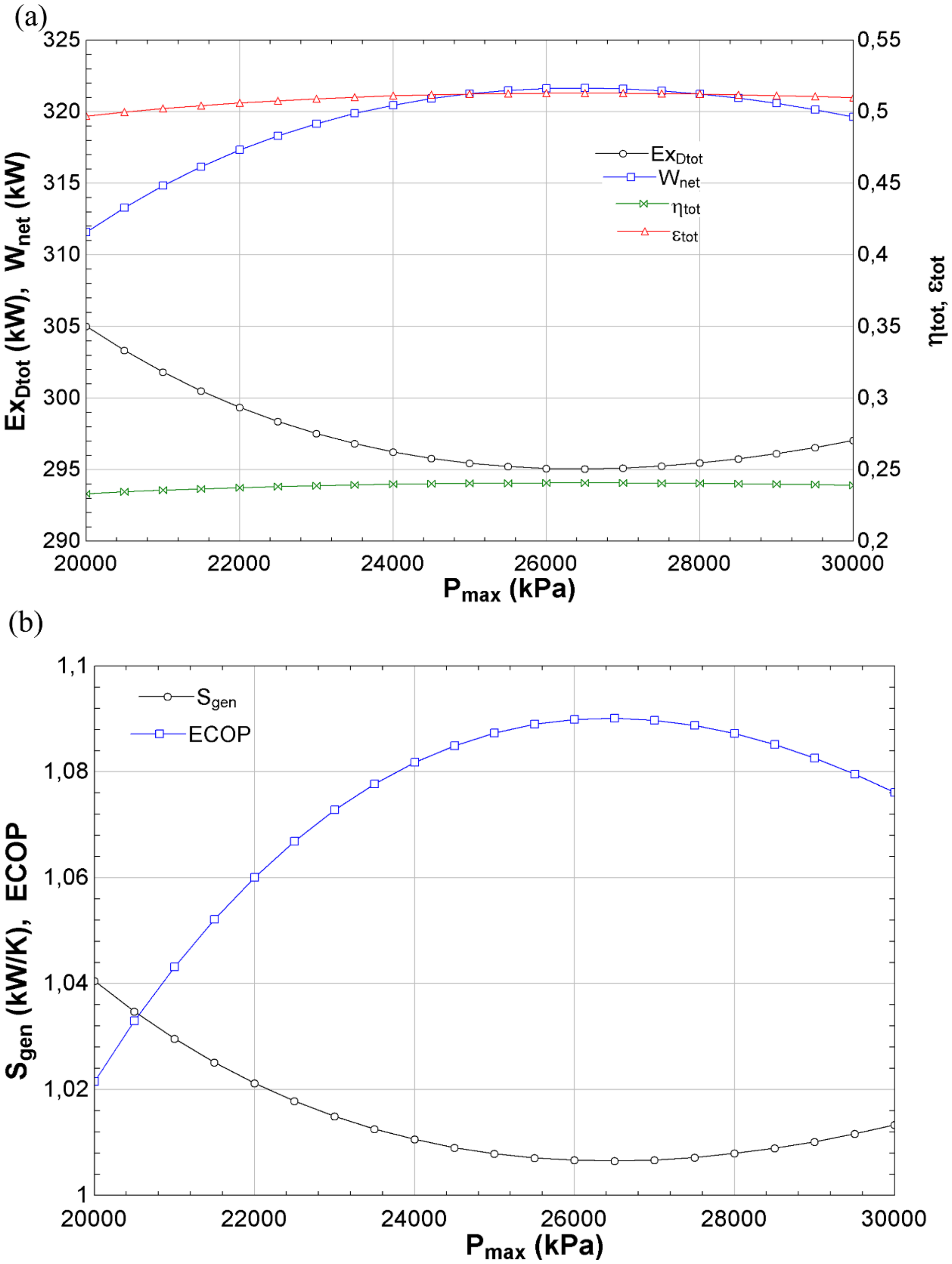
Concerning the schematic representation of the design, first, the minimum temperature of exhaust gas, which leaves the WHR system is optimized by considering the temperature difference in the heat exchanger, maximum net power output, thermal efficiency, and expander inlet pressure. Thence, the optimization procedure is performed by iterating the exhaust gas outlet temperature from 190 °C to 170 °C as seen in Figure 2. As the temperature decreases, the net power output increases as expected. However, the temperature difference (ΔT) between the hot exhaust gas outlet and cold working fluid inlet temperatures in the heat exchanger is not acceptable (less than 4 °C) below 175 °C. Consequently, the optimum exhaust gas outlet temperature is selected as 175 °C.

**Figure 2** Variation of net power output and temperature difference




Later, the maximum pressure of the cycle is required to be optimized with respect to maximization and minimization of the objective functions namely the net power output, exergy destruction, cycle efficiency, exergy efficiency, ECOP, and entropy generation by iterating the pressure within a certain range, while controlling the pinch point temperature difference of the heat exchanger, ΔT. The variations of the objective functions could be seen in Figures 3(a) and (b). The figures indicate that there is one optimum pressure level that all objective functions compromise.

**Figure 3** Variations of objective functions a) total exergy destruction, net power output, energy, and exergy efficiencies b) entropy generation and ECOP with respect to the maximum pressure.




The detailed results of the objective functions are shown in Table 5**.** It is observed that the optimization of objective functions yields maximum pressure of 26443.40 kPa. On the other hand, the variation of the net power output is neglectable between 26443.40 and 26500 kPa (321.659 kW-321.658 kW). For realization purposes of the cycle, it is accepted as 26500 kPa with a neglectable error. The overall WHR system efficiency is similar in the given detailed pressure range. Moreover, the temperature difference between hot and cold streams is greater than 4 °C in the heat exchanger, ΔT, as well as the regenerator.

**Table 5** Optimization results of the maximum pressure of the cycle

| $P_1$ (kPa) | $\dot{W}_{net}$ (kW) | $\eta_{tot}$ | ΔT (°C) | ECOP | $\dot{S}_{gen}$ (kW/K) | $\dot{E}x_{D_{tot}}$ (kW) | $\varepsilon_{tot}$ |
|---|---|---|---|---|---|---|---|
| 20000 | 311.589 | 0.233109 | 3.618 | 1.02152 | 1.04051 | 305.026 | 0.496981 |
| 25500 | 321.491 | 0.240518 | 6.909 | 1.08898 | 1.00707 | 295.223 | 0.512766 |
| 26000 | 321.623 | 0.240616 | 7.09 | 1.08890 | 1.00663 | 295.093 | 0.512986 |
| 26500 | 321.658 | 0.240643 | 7.26 | 1.09016 | 1.00651 | 295.057 | 0.513043 |
| 27000 | 321.604 | 0.240602 | 7.421 | 1.08977 | 1.00669 | 295.111 | 0.512956 |
| 27500 | 321.465 | 0.240498 | 7.576 | 1.08879 | 1.00716 | 295.249 | 0.512734 |
| 30000 | 319.652 | 0.239141 | 8.273 | 1.07611 | 1.01328 | 297.044 | 0.509842 |

According to the selected expander inlet pressure and exhaust gas outlet temperature, the system is constituted, and the obtained data are shown in Table 6.

**Table 6** Thermodynamic properties of the investigated system

| State | $\dot{m}$ (kg/s) | T (°C) | P (kPa) | ex (kJ/kg) | State | $\dot{m}$ (kg/s) | T (°C) | P (kPa) | ex (kJ/kg) |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 4.622 | 380 | 26500 | 413.42 | 7 | 4.622 | 167.70 | 26500 | 282.49 |
| 2 | 4.622 | 254 | 6892 | 283.38 | 8 | 5.83 | 395 | 101.325 | 138.56 |
| 3 | 4.622 | 85.27 | 6892 | 219.81 | 9 | 5.83 | 175 | 101.325 | 31.02 |
| 4 | 4.622 | 28 | 6892 | 208.69 | 10 | 40.45 | 20 | 101.325 | 0 |
| 5 | 4.622 | 28 | 6892 | 206.26 | 11 | 40.45 | 22.50 | 101.325 | 0.04 |
| 6 | 4.622 | 69.66 | 26500 | 235.17 | 12 | 40.45 | 26 | 101.325 | 0.25 |

Under optimal conditions, the net power output, and the total thermal efficiency of the WHR system are calculated as 321.658 kW and 0.24064, respectively. The theoretical efficiency of the marine engine is calculated as 0.4638. Producing more power with the WHR system leads to a rise of 8.17% of the overall thermal efficiency with a theoretical efficiency of 0.5017. As a result, specific fuel consumption decreased from 167.5 g/kWh to 154.87 g/kWh, and fuel economy improved by 7.54%. In addition, the amount of $CO_2$ and other greenhouse gases released into the environment decreases with lower fuel consumption.

Exergy of the exhaust gas that is 808 kW, is released into the atmosphere without a WHR, therefore this wasted exergy results in increasing not only the fuel consumption but also environmental pollution. By utilizing the $sCO_2$-WHR system, some of the exhaust gas exergy could be recovered into usable energy and negative environmental effects could be prevented. Moreover, the exergy analysis of the WHR system is performed to determine the inefficiencies within the system and its components, as well as their environmental effects. Results of the exergy analysis, given in Table 7, reveal the exergy destruction, fuel exergy, product exergy, and exergy efficiency of each component and total WHR system.

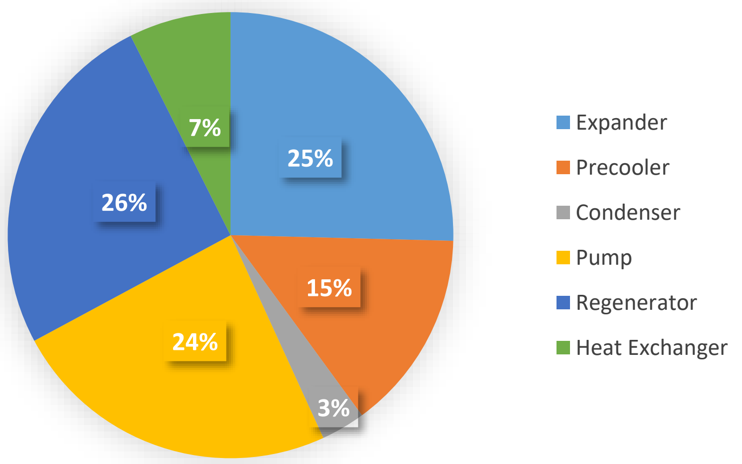
**Table 7** Exergy analysis of the simple $sCO_2$ Rankine cycle

| Component | $\dot{E}x_F$ (kW) | $\dot{E}x_P$ (kW) | $\dot{E}x_D$ (kW) | $\varepsilon_k$ |
|---|---|---|---|---|
| Expander | 601 | 526.10 | 74.96 | 0.8753 |
| Precooler | 51.41 | 8.452 | 42.95 | 0.1644 |
| Condenser | 11.24 | 1.794 | 9.448 | 0.1596 |
| Pump | 204.4 | 133.6 | 70.77 | 0.6538 |
| Regenerator | 293.8 | 218.7 | 75.1 | 0.7444 |
| Heat Exchanger | 627 | 605.1 | 21.83 | 0.9652 |
| Total | 627 | 321.7 | 295.06 | 0.513 |

Total exergy destruction is determined as 295.06 kW, which is the sum of all components' exergy destructions. It is obvious that the WHR system decreases the exergy loss of the power production system. However, exergy loss could not be eliminated by the WHR system because of occurring exergy destruction in each component and releasing exhaust gas exergy into the atmosphere. The exhaust gas outlet exergy is calculated as 181 kW at 175 °C, while the ejected seawater carries approximately 10 kW of exergy to be lost. Consequently, the total exergy loss of the $sCO_2$-WHR system is determined as 486.3 kW. Compared to the power system without an appended WHR (808 kW), reverted exergy loss (321.7 kW) is eliminated by the WHR system, which is nearly 40% of the total exergy recovery.

Table 7 shows that 51.3% of total fuel exergy is converted into product exergy, which means that there is still some unused energy. Exergy destruction occurs in each component due to the irreversibility and gives ideas to designers for component improvement. Exergy destruction rates in percentiles are shown in Figure 4.

**Figure 4** Exergy destruction ratios of components

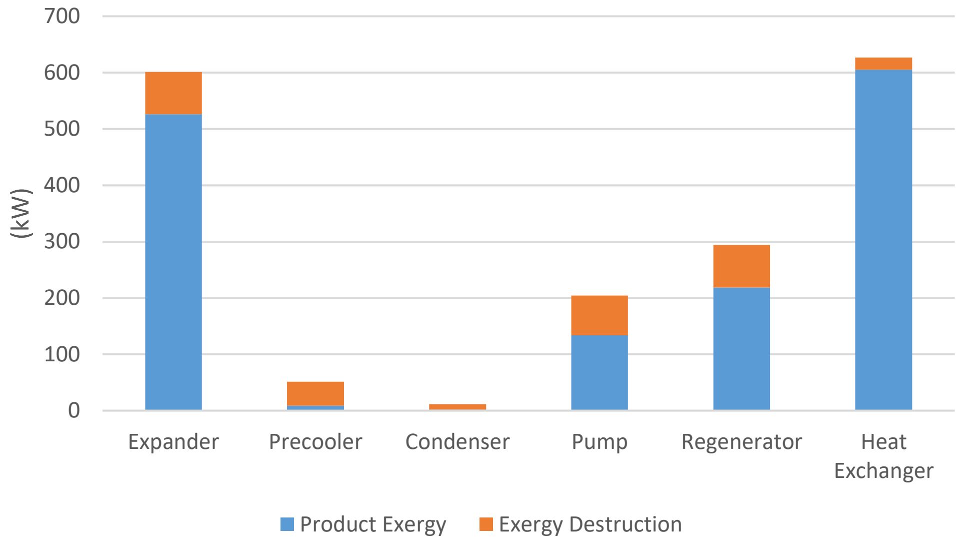



According to Figure 4, the top three have slightly different exergy destruction percentages. Among them, the regenerator has the highest exergy destruction due to the high amount of temperature differences between the inlets and the outlets of hot and cold streams (86.3 °C). Moreover, it is related to the dramatic temperature drop of the hot stream (254~85.27 °C). It is well-known that exergy destruction is directly proportional to the finite temperature difference, therefore the system must be improved to lower the exergy destruction as double pressure, split, or cascade systems. These innovations would help to lower the temperature difference, thus the exergy destruction of the component. On the other hand, it will bring an extra economic burden, thus it should be optimized. The expander is the component with the second highest exergy destruction because of not only the isentropic efficiency of the expander but also the relatively higher fuel exergy. It is a good approach to lower the friction, and leakages on the expander to improve the isentropic efficiency. The following component is the pump, which has an isentropic efficiency of 0.60 and pressurizes the working fluid to a high-pressure level by passing the critical point, which is caused by not only pressuring the working fluid but compressing it as well. Compression increases the temperature of the working fluid which leads to an increase in the specific volume of the component. It might be better to cool the component to work efficiently.

The precooler has relatively high exergy destruction compared with the heat exchanger and condenser because of high temperature drop (85.27~28 °C) and high temperature difference (59.27 °C). Some of the heat potential of the precooler might be evaluated to preheat the working fluid, concerning the temperature differences of the working fluid between the pump/compressor outlet and precooler inlet. Therefore, the temperature elevation could be stepped down. Even though the highest fuel and product exergies belong to the heat exchanger, which transfers heat from the high-temperature exhaust gases to the working fluid, the temperature differences on each side of the heat exchanger are low (15 and 7.3 °C). That concludes the relatively low exergy destruction in the heat exchanger. The condenser has the minimum destruction among all components. There is only a phase change at constant temperature and the cooling fluid, the seawater, is in a similar temperature range, therefore the exergy destruction results in the lowest.

Exergy destructions are the key to decreasing the environmental effects as well as the economic issues. On the other hand, it would be better to evaluate exergy destructions together with the fuel and product exergies. Figure 5 shows the results of exergy analysis for each component regarding their product and destruction exergies. The total of the mentioned exergies will result in the exergy of fuel for each component. It is obvious that the condenser has the lowest exergy destruction, but its product exergy is way lower. Thus, it has the lowest exergy efficiency. It is expected due to the cooling of the working fluid in a similar range of temperature on both sides of the component. Similarly, the precooler in the same range has almost identical efficiency to the condenser. On the other hand, this component has the potential to be utilized as mentioned above. It could be stated that these two components have the highest potential to decrease their exergy destruction by far a better design. However, it is arguable to invest in the mentioned components due to the low exergy recovery potentials. On the other hand, the heat exchanger utilizes almost all the fuel exergy that leads to the highest exergy efficiency with almost no room for improvement in this component. Nevertheless, it should be investigated deeply that it might have a better opportunity to recover more exergy destruction than the condenser. This could be done by increasing the expander inlet temperature. Overall, the exergy efficiencies of other components in between are queued in ascending order as the pump, regenerator, and expander, respectively. The exergy efficiency gives information to evaluate the system and its components regarding the improvement potentials. It is important to determine the exergy efficiencies of the components parallel to the exergy destructions to see where to focus on to recover the exergy destruction. That concludes, lower exergy efficiency might lead to more improvement potential.

**Figure 5** Exergy of product and exergy destruction of each component, sum yields exergy of fuel




*4.1 Literature Comparison*

The results of the present study are compared to the studies in the literature. The systems in the literature have a wide variety of temperatures, net power output, and exergy destructions, therefore obtained results of the non-dimensional objective functions, namely energy efficiency, exergy efficiency, and ECOP are evaluated for comparison as seen in Table 8. It should be noted that the adjacent ECOP results in the table are calculated by the values given in the corresponding studies.

**Table 8** Comparison of different studies with respect to the objective functions.

| | Present Study | Kim et al. (2017) | | | Shu et al. (2016) | |
|---|---|---|---|---|---|---|
| | | Simple | Cascade | Split | Simple | Preheated-regenerative |
| Energy efficiency | 0.240643 | 0.287 | 0.282 | 0.290 | 0.08 | 0.15 |
| Exergy efficiency | 0.513043 | 0.549 | 0.584 | 0.626 | 0.15 | 0.35 |
| ECOP | 1.09016 | 1.21874 | 1.40238 | 1.67401 | 0.19666 | 0.63776 |

For a fair comparison, it should be stated that the study of Shu et al. (2016) utilizes the exhaust gas of a gasoline engine with the power output of 43.8 kW with an exhaust mass flow rate and temperature of 202.6 kg/h and 1050 K (777 °C), respectively. Moreover, their PR-CTRC design also recovers waste heat of engine coolant with the addition of a preheater and regenerator/recuperator. They assumed that the isentropic efficiency of the turbine and the pump are 0.70 and 0.80 respectively, while their pinch temperature difference in the heat exchanger is 30 K, and maximum pressure is selected as 14760 kPa. Turbine inlet temperatures are different for each cycle as well.

On the other side, the study of Kim et al. (2017) utilizes a recuperator for all their designed cycles. Their values are given in Table 4. They also assumed the pinch temperature difference as 30 K and turbine inlet temperatures are different for each system.

It could be observed from Table 8 that, the more complex the system topology yields the better performance in their boundaries. Moreover, the results of Kim et al. (2017) for a basic $sCO_2$-RC system are better than the present study for all objective functions, while those of Shu et al. (2016) have the lowest values in comparison. However, it should be noted that the turbine inlet temperature of Kim et al. (2017) is higher than in the present study. Moreover, they have a similar pressure range, and its effect, as shown in Figure 3 and Table 5, is relatively lower on the energy efficiency, exergy efficiency, and ECOP results. In contrast, the exhaust inlet temperature, and the turbine inlet temperature of Shu et al. (2016) are the highest, it does not have a recuperator for the simple system. Even though the preheater and recuperator are included in the system, its pressure is lower among all. Therefore, the values are the lowest.

# 5 Conclusion

In this study, a natural gas-fueled marine engine with a power of 3940 kW is considered. It is aimed to recover waste heat of engine exhaust gas by adding a WHR system to achieve a low specific fuel consumption, thus low financial burden as well as less $CO_2$ emission. The system is validated via the data from the literature. The energy of the exhaust gas is utilized in the designed system via the heat exchanger to produce more power with the same amount of fuel, hence, increasing overall efficiency. Not only the exhaust gas outlet temperature but also the maximum pressure of the WHR system is optimized to reach maximum net power output, energy efficiency, thermal efficiency, ECOP, and minimum exergy destruction as well as entropy generation. According to the optimization results, it is stated that the upper pressure of the cycle of 26500 kPa, and exhaust outlet temperature of 175 °C are feasible as the optimum conditions.

Specific fuel consumption of the power system is reduced by 7.54% and overall system efficiency is enhanced by 8.17% with $sCO_2$-RC. Further, the wasted exergy of the engine exhaust gas is decreased by 40%. Hence, it is shown that approximately half of the exhaust gas exergy could be converted into power in the $sCO_2$-RC WHR system. A detailed exergy analysis for each component of the WHR system is conducted to identify possible improvements. In terms of fuel exergy, it is seen that the maximum and the minimum values are obtained in the heat exchanger and condenser as 627 kW and 11.24 kW, respectively. The maximum exergy destruction occurs in the regenerator as 75.1 kW due to high temperature difference between hot and cold streams. It is followed by the expander (74.96 kW), pump (70.77 kW), precooler (42.95 kW), the heat exchanger (21.83 kW), and condenser (9.448 kW) in sequence. The heat exchanger has the maximum exergy efficiency of 0.9652 while the condenser has the minimum of 0.1596. In between these components, there is the expander (0.8753), regenerator (0.7444), pump (0.6538), and precooler (0.1644), respectively.

The entire WHR system has 627 kW fuel exergy, 321.7 kW product exergy, and 295 kW exergy destruction in total. Consequently, the WHR system has exergy and energy efficiencies of 0.513 and 0.2406, respectively. It could be said that there is still a large portion of energy wasted from engine exhaust gas even though usage of the $sCO_2$-RC WHR system.

The results also yielded that the maximum pressure of the system in a close range does not affect the objective function results, such as the net power output, ECOP, energy, and exergy efficiencies as much. Therefore, similar results could be gained with relatively lower expander inlet pressure. This could be validated by the comparison of the literature results. On the other hand, it is seen that the outlet temperature of the exhaust gas has more effect on the net power output. Moreover, it is observed the pinch temperature difference in the heat exchanger influences the net power output. Furthermore, it is noticed that the point that optimizes efficiency, net power output, and ECOP intersects. Furthermore, the designed system disclosed that ECOP could be evaluated as a reference comparison value for different states and operational conditions of various systems.

In conclusion, the addition of a $sCO_2$-RC WHR system is a good option to recover the exhaust energy of a marine engine. It is also obtained that applying exergy analysis to reveal exergy destruction that is a way to determine the inefficiencies within the system, is a better way to evaluate the improvement potentials of the power systems. However, it would be better to utilize exergy efficiency along with the destruction to result in more accurate ways to improve the systems.

Lastly, it is clear that the released exhaust gas still has a relatively high temperature even after the WHR system. Hence, it should be aimed to recover this part and carry out an exergy analysis as well as an economic investigation on the improved WHR system with the annexation of an organic Rankine cycle for future studies.

**References**


Bae, S. J., et al. Hybrid system of Supercritical Carbon Dioxide Brayton cycle and carbon dioxide rankine cycle combined fuel cell. Turbo Expo: Power for Land, Sea, and Air, 2014. American Society of Mechanical Engineers, V03BT36A004.

Bejan, A., Tsatsaronis, G.andMoran, M. J. 1995. *Thermal design and optimization,* John Wiley & Sons.

Cengel, Y. A.andBoles, M. A. 2015. Thermodynamics: An Engineering Approach McGraw-Hill Education. *New York*.

Chaudhary, A., Mulchand, A.andDave, P. Feasibility Study of Supercritical CO2 Rankine Cycle for Waste Heat Recovery. Proceedings. In Proceedings of the 6th International Symposium-Supercritical CO2 Power Cycles, Pittsburgh, PA, USA, 2018. 27-29.

Chen, Y., et al. 2006. A comparative study of the carbon dioxide transcritical power cycle compared with an organic Rankine cycle with R123 as working fluid in waste heat recovery. *Applied thermal engineering,* 26**,** 2142-2147.

Glover, S., et al. 2015. Simulation of a multiple heat source supercritical ORC (Organic Rankine Cycle) for vehicle waste heat recovery. *Energy,* 93**,** 1568-1580.

Gonca, G. 2018. Thermo-ecological performance analysis of a double-reheat Rankine cycle steam turbine system (RCSTS) with open and close feed water heaters. *International Journal of Exergy,* 25**,** 117-131.

Güneş, Ü.andKarakurt, A. 2015. Exergy And Economic Analysis Of Dual Pressure Waste Heat Recovery Boiler. *International Journal of Advances in Mechanical and Civil Engineering,* 2.

İbrahim, K., Karakurt, A. S.andYasin, Ü. 2020. Investigation of waste heat energy in a marine engine with transcritical organic Rankine cycle. *Journal of Thermal Engineering,* 6**,** 282-296.

Karakurt, A. S. 2020. Comparative maximum power density analysis of a supercritical CO2 Brayton power cycle. *Journal of Thermal Engineering,* 6**,** 50-57.

Karakurt, A. S., Ozel, I. F.andİskenderlİ, S. 2021. Performance analyses and optimization of a regenerative supercritical carbon dioxide power cycle with intercooler and reheater. *Seatific,* 1**,** 1-6.

Karakurt, A. S., Ozsari, I.andBashan, V. 2022. Exergetic performance analysis of high pressure air systems on ships. *International Journal of Exergy,* 37**,** 74-86.

Kim, Y. M., Sohn, J. L.andYoon, E. S. 2017. Supercritical CO2 Rankine cycles for waste heat recovery from gas turbine. *Energy,* 118**,** 893-905.

Klein, S. 2018. EES. F-chart software professional V10. (2018). ed.

Koroglu, T.andSogut, O. S. 2018. Conventional and advanced exergy analyses of a marine steam power plant. *Energy,* 163**,** 392-403.
Liang, Y., et al. 2019. Theoretical analysis of a regenerative supercritical carbon dioxide Brayton cycle/organic Rankine cycle dual loop for waste heat recovery of a diesel/natural gas dual-fuel engine. *Energy Conversion and Management,* 197**,** 111845.
Manente, G.andFortuna, F. M. 2019. Supercritical CO2 power cycles for waste heat recovery: A systematic comparison between traditional and novel layouts with dual expansion. *Energy Conversion and Management,* 197**,** 111777.
Mohammadi, K., Ellingwood, K.andPowell, K. 2020. A novel triple power cycle featuring a gas turbine cycle with supercritical carbon dioxide and organic Rankine cycles: Thermoeconomic analysis and optimization. *Energy Conversion and Management,* 220**,** 113123.
Özsari, İ. 2022. Denizaltılarda Havadan Bağımsız Tahrik Sistemi Olarak Kullanılan MESMA Sisteminin Termodinamik Performans Analizi. *Gemi ve Deniz Teknolojisi***,** 59-74.
Pan, M., et al. 2020. Theoretical analysis and comparison on supercritical CO2 based combined cycles for waste heat recovery of engine. *Energy Conversion and Management,* 219**,** 113049.
Pulkrabek, W. W. 2007. *Engineering fundamentals of the internal combustion engine,* Prentice Hall.
Rolls-Royce 2016. Project Guide: Bergen Engine Type B35:40 Gas. .
Sarkar, J. 2015. Review and future trends of supercritical CO2 Rankine cycle for low-grade heat conversion. *Renewable and Sustainable Energy Reviews,* 48**,** 434-451.
Shu, G., et al. 2016. An improved CO2-based transcritical Rankine cycle (CTRC) used for engine waste heat recovery. *Applied Energy,* 176**,** 171-182.
Song, J., et al. 2020. 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 Conversion and Management,* 218**,** 112999.
Ust, Y., Sahin, B.andCakir, M. 2016. Ecological coefficient of performance analysis and optimisation of gas turbines by using exergy analysis approach. *International journal of exergy,* 21**,** 39-69.
Ust, Y., et al. 2006. Ecological coefficient of performance analysis and optimization of an irreversible regenerative-Brayton heat engine. *Applied Energy,* 83**,** 558-572.
Ust, Y., Sahin, B.andSogut, O. S. 2005. Performance analysis and optimization of an irreversible dual-cycle based on an ecological coefficient of performance criterion. *Applied energy,* 82**,** 23-39.
Üst, Y., Sahin, B.andKodal, A. 2005. Ecological coefficient of performance (ECOP) optimization for generalized irreversible Carnot heat engines. *Journal of the Energy Institute,* 78**,** 145-151.
Wu, H., et al. 2021. Power, efficiency, ecological function and ecological coefficient of performance optimizations of irreversible Diesel cycle based on finite piston speed. *Energy,* 216**,** 119235.
Xu, J., et al. 2019. Perspective of S− CO2 power cycles. *Energy,* 186**,** 115831.
Yang, H., et al. 2020. Performance investigation of solar tower system using cascade supercritical carbon dioxide Brayton-steam Rankine cycle. *Energy Conversion and Management,* 225**,** 113430.
Zhang, Y., et al. 2018. Low-grade heat utilization by supercritical carbon dioxide Rankine cycle: Analysis on the performance of gas heater subjected to heat flux and convective boundary conditions. *Energy Conversion and Management,* 162**,** 39-54.
Zhou, A., et al. 2020. Improvement design and analysis of a supercritical CO2/transcritical CO2 combined cycle for offshore gas turbine waste heat recovery. *Energy,* 210**,** 118562.