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Exergy Analysis of Supercritical CO_2 System for Marine Diesel Engine Waste Heat Recovery Application

Abstract

In this research is performed an exergy analysis of supercritical CO_2 system which uses various waste heat flows from marine diesel engine to produce additional mechanical power. The performed exergy analysis contains whole system as well as each system component individually. The observed system produces useful mechanical power equal to 2299.47 kW which is transferred to the main propulsion propeller shaft. Additionally produced mechanical power by using waste heat only will reduce marine diesel engine fuel consumption and exhaust gas emissions. Main cooler has the highest exergy destruction of all system components and simultaneously the lowest exergy efficiency in the observed system, equal to 32.10% only. One of the possibilities how main cooler exergy efficiency can be increased is by decreasing water mass flow rate through the main cooler and simultaneously by increasing water temperature at the main cooler outlet. Observed system has five heat exchangers which are involved in the CO_2 heating process, and it is interesting that the last CO_2 heater (exhaust gas waste heat exchanger) increases the CO_2 temperature more than all previous four heat exchangers. Whole analyzed waste heat recovery supercritical CO_2 system has exergy destruction equal to 2161.68 kW and exergy efficiency of 51.54%. In comparison to a similar CO_2 system which uses waste heat from marine gas turbine, system analyzed in this paper has approximately 12% lower exergy efficiency due to much lower waste heat temperature levels (from marine diesel engine) in comparison to temperature levels which occur at the marine gas turbine exhaust.

Keywords: Exergy analysis, Supercritical CO_2 system, Waste heat recovery, Marine diesel engine

1. Introduction

Diesel engines are today the dominant producers of mechanical power used for the propulsion in a shipping industry [1, 2]. Due its dominancy, many researchers are involved in the analysis and optimization of various diesel engines used in the marine sector [3, 4]. Tracking diesel engine operating parameters, instead of standard measurements, today is often performed by developing various simulation models which can accurately and precisely track its operation in the real exploitation conditions [5-8]. One of the most important goals in the field of marine diesel engines, especially in a recent time, is reducing its fuel consumption (and consequentially its exhaust gas emissions) with an aim to satisfy current legislation [9-11].

Researchers and scientists have developed various techniques, methods and processes for reducing fuel consumption and/or exhaust gas emissions from marine diesel engines [12-14]. It is proved that many of such techniques, methods and processes have a very beneficial impact on the diesel engine operation process and more importantly, they reduce the harmful impact of the diesel engines on the environment [15-17].

One of the options for reducing diesel engine fuel consumption and emissions is using a various upgrades and modifications which can also increase the efficiency of the whole system (diesel engine + upgrade) [18, 19]. Such upgrades can be performed on the diesel engine [20] or can be installed as an additional component with some kind of connection to diesel engine [21].

Recently, in the literature can be found propositions that standard marine mechanical power producers (diesel engines or gas turbines) should be upgraded with various waste heat recovery systems, which will not reduce fuel consumption and exhaust gas emissions only, they will also increase overall plant efficiency [22, 23]. Often proposed waste heat recovery systems for diesel engines or gas turbines are supercritical CO_2 systems [24, 25]. It should be highlighted that supercritical CO_2 systems must be much differently composed in regards to main mechanical power producer (diesel engine or gas turbine). At the gas turbine outlet combustion gases have very high temperature levels, while diesel engines have various waste heat fluids which temperature levels notably differ and they are much lower in comparison to gas turbines [26, 27]. Therefore, when supercritical CO_2 system is applied along with the diesel engine, it should be expected cascade process of CO_2 heating.

This paper presents an exergy analysis of a waste heat recovery supercritical CO_2 system which for its operation uses heat from three different diesel engine waste heat fluids (cooling jacket water, scavenge air and exhaust gas). Using mentioned waste heat from three different diesel engine fluids allows production of additional mechanical power which is transferred to a main propulsion propeller shaft. In that way, the analyzed supercritical CO_2 system will reduce cumulative mechanical power produced by the diesel engine and delivered to the main propulsion propeller shaft what will simultaneously reduce diesel engine fuel consumption and exhaust gas emissions.

It is performed calculation of exergy power inlets, outlets, exergy destructions and exergy efficiencies of each system component and of the whole observed system. Performed exergy analysis detects the most problematic component in the observed waste heat recovery supercritical CO_2 system, and one possibility of that component improvement is presented and described. The obtained exergy efficiency of the whole system is compared to a similar supercritical CO_2 system which uses waste heat from the marine gas turbine.

2. Description and operating characteristics of the analyzed waste heat recovery supercritical CO_2 system

General scheme of the observed waste heat recovery supercritical CO_2 system is presented in Figure 1. In Figure 1 can also be seen operating points (marked with the numbers) required for the performed exergy analysis.

Analyzed system is composed of six heat exchangers, one turbocompressor and one turbine. Heat exchangers are arranged according to available heat levels contained in various waste heat flows from marine diesel engine. Turbocompressor (TC) increases the CO_2 pressure (simultaneously TC increases CO_2 temperature) and delivers it to Cooling Jacket Water waste heat exchanger (CJW). In the CJW waste heat contained in water from diesel engine cooling jacket is transferred to CO_2 what increases CO_2 temperature. After CJW is mounted the first Recuperator (R1) which is actually CO_2-CO_2 heat exchanger (CO_2 of higher temperature – operating points from 9 to 10, Figure 1, transfer heat to CO_2 of lower temperature – operating points from 3 to 4, Figure 1). After R1, CO_2 which passes from TC to turbine (TURB) increases temperature by heat transfer in Scavenge Air waste heat exchanger (SA). In the SA heat is transferred from the scavenge air after charger (turbocharger assembly) to CO_2 (in that way scavenge air is cooled, what decreases water flow in air cooler mounted after SA and before diesel engine in the air flow line). After SA, CO_2 passes through another Recuperator (R2) which is a second CO_2-CO_2 heat exchanger (CO_2 of higher temperature – operating points from 8 to 9, Figure 1, transfer heat to CO_2 of lower temperature – operating points from 5 to 6, Figure 1). The final increase in CO_2 temperature before its expansion in the turbine occurs in Exhaust Gas waste heat exchanger (EG). In the EG exhaust gas after the turbine (turbocharger assembly) transfer heat to CO_2 and after the EG CO_2 has the highest temperature in the observed supercritical CO_2 system. CO_2 of the highest temperature (operating point 7, Figure 1) expands through the turbine (TURB), and the turbine produces mechanical power due to CO_2 expansion.

One part of cumulatively produced mechanical power in TURB is used for the TC drive (TURB and TC are mounted on the same shaft through which TC gets mechanical power from the TURB for its drive), while mechanical power surplus produced by the TURB (useful mechanical power) is transferred to the main propulsion propeller shaft. In that way, the observed supercritical CO_2 system reduces mechanical power produced by the marine diesel engine and simultaneously reduces diesel engine fuel consumption and exhaust gas emissions.

The last heat exchanger necessarily needed in the observed supercritical CO_2 system is Main Cooler (MC) mounted before TC. MC is used for the final CO_2 cooling before its entrance to TC (operating point 1, Figure 1). Before TC, CO_2 has the lowest temperature in the entire observed system. In the MC, heat is transferred from the CO_2 to cooling water. As can be found in the literature [27], MC operation has significant influence on the operation of whole observed supercritical CO_2 system, so a special attention in performed exergy analysis will be placed on this heat exchanger.

The observed CO_2 system is supercritical because CO_2 pressure in the entire system (Table 4) is higher than CO_2 critical pressure (CO_2 critical pressure is equal to 73.773 bar [28]). Instead of CO_2, in such system can be used other gases (helium, neon, argon, nitrogen, etc.). However, it should be highlighted that any other gas in comparison to CO_2 will have different thermodynamic properties and the observed system will not operate with the same operating parameters in each operating point from Figure 1. One of the possibilities in further research of this system can be investigation of operating parameters and system performances when other gases are used [29].

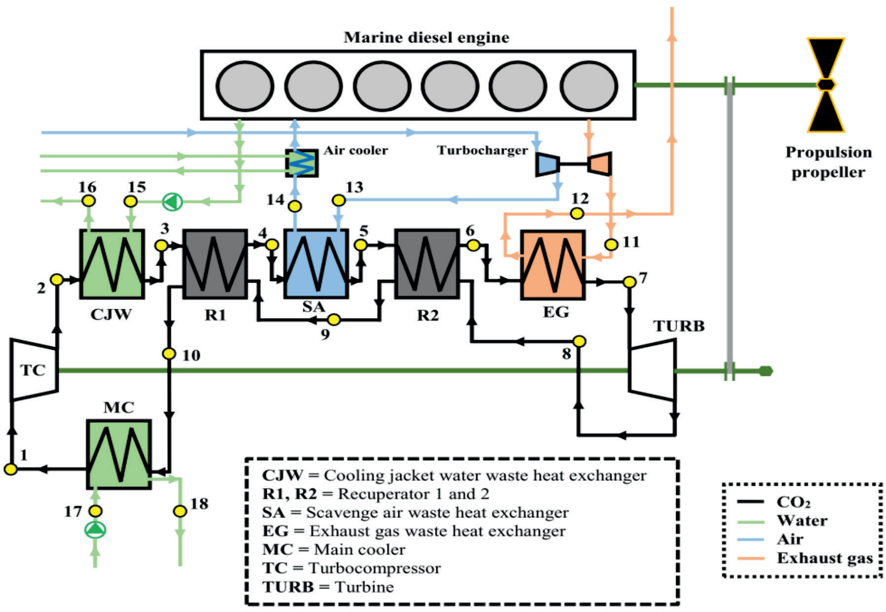

Figure 1 – General scheme of the observed waste heat recovery supercritical CO_2 system with marked operating points required for the exergy analysis

Exergy parameters of the whole observed system are obtained according to Figure 2 (operating point markings are related to Figure 1). The whole observed system has four exergy power inlets: exhaust gas inlet, scavenge air inlet, cooling jacket water inlet and main cooling water inlet. Whole observed system also has five exergy power outlets: exhaust gas outlet, scavenge air outlet, cooling jacket water outlet, main cooling water outlet and the fifth exergy power outlet is useful produced mechanical power. Mentioned elements are required and sufficient for proper exergy analysis of the whole observed waste heat recovery supercritical CO_2 system. It should also be highlighted that additional control of the performed calculations will be performed in a way that whole system exergy destruction (obtained by using Figure 2) will be compared with the sum of exergy destructions of all system components – if the calculations are correct, these two values will be identical.

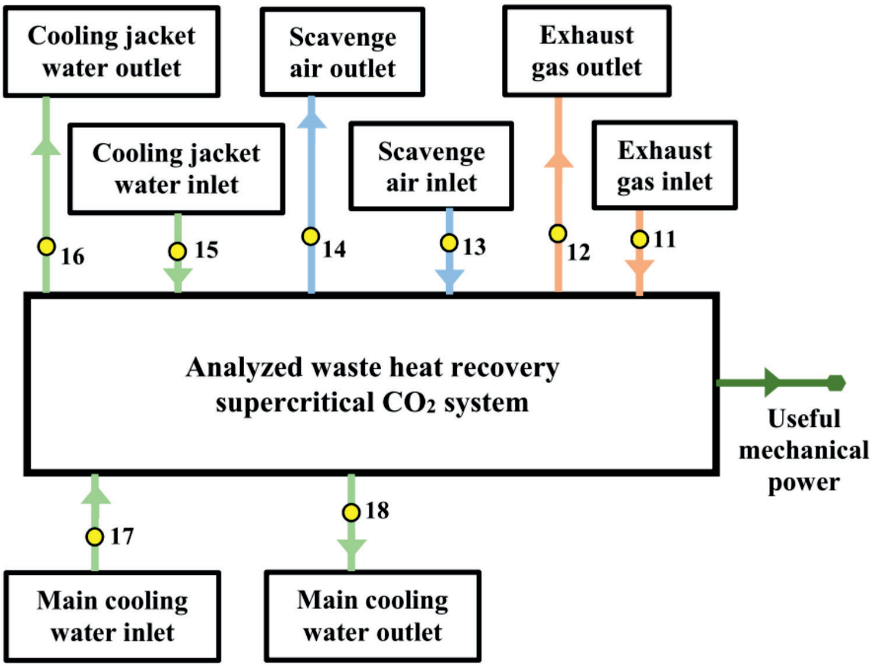

Figure 2 - Scheme for exergy analysis parameters definition of the whole observed waste heat recovery supercritical CO_2 system

3. Exergy analysis and equations

Exergy analysis has a baseline in the second law of thermodynamics [30, 31]. Unlike energy analysis which is based on the first law of thermodynamics and which did not consider parameters of the ambient [32, 33], exergy analysis takes into consideration pressure and temperature of the ambient in which observed component or a system operates [34, 35]. Therefore, exergy analysis offers an additional possibility in comparison to energy analysis – by using exergy analysis can be investigated the ambient parameters change and its influence on the observed component or a system destructions and efficiencies [36, 37].

In a recent literature, various researchers prefer the usage of exergy analysis instead of energy analysis [38-40]. In addition, exergy analysis can be a baseline for more complex analyses of any observed component or a system [41-43].

The additional exergy analysis benefit is that it is actually a "black box" method, which means that it did not require the detail inner structure of observed component – sufficient elements are fluid, power and heat flows only (to any component and from any component) [44, 45].

3.1. General exergy equations and balances

In the exergy analysis exists a few equations and balances which should always be satisfied, regardless of the observed component or a system and regardless of the observed operating regimes [46, 47]. These general equations and balances will always be satisfied also in this analysis. The general exergy balance equation for a component or a system in steady state, according to [48] is:

$$\dot{X}_{\mathrm{heat}} - P = \sum \dot{E}x_{\mathrm{out}} - \sum \dot{E}x_{\mathrm{in}} + \dot{E}x_{\mathrm{D}} \,. \tag{1}$$

In Eq. 1, $\dot{E}x$ is a total fluid exergy flow, which is defined as [49]:

$$\dot{E}x = \dot{m} \cdot \varepsilon \,, \tag{2}$$

while $\dot{X}_{\mathrm{heat}}$ is an exergy heat transfer at the temperature T, which definition can be found in [50] and presented by an equation:

$$\dot{X}_{\mathrm{heat}} = \sum (1 - \frac{T_0}{T}) \cdot \dot{Q} \,. \tag{3}$$

Moreover, in Eq. 2, ε is fluid specific exergy which can be defined by an equation [51]:

$$\varepsilon = \left(h - h_0\right) - T_0 \cdot \left(s - s_0\right). \tag{4}$$

In standard component or system operation, mass flow rate leakage of any fluid stream did not occur, so the valid mass flow rate balance is [52]:

$$\sum \dot{m}_{\text{in}} = \sum \dot{m}_{\text{out}} . \tag{5}$$

General exergy efficiency definition for any component or a system can be found in [53] and presented by an equation:

$$\eta_{\text{ex}} = \frac{\text{cumulative exergy outlet}}{\text{cumulative exergy inlet}} . \tag{6}$$

It should be highlighted that above presented a general exergy efficiency equation (Eq. 6) can take many different forms, what depends on the observed component or a system operation characteristics and specificities.

3.2. Equations for the exergy analysis of observed waste heat recovery supercritical CO_2 system and its components

Equations required for the exergy analysis of each component and whole observed waste heat recovery supercritical CO_2 system are presented in this section. Each component and whole system require equations for the calculation of the exergy power inlet, exergy power outlet, exergy destruction and exergy efficiency [54]. Markings in all equations presented in this section are related to operating points presented in Figure 1.

Exergy analysis equations of all heat exchangers from the observed waste heat recovery supercritical CO_2 system are defined according to recommendations from the literature [55, 56] and presented in Table 1.

Table 1 - Exergy analysis equations of all heat exchangers from the observed waste heat recovery supercritical CO_2 system

Component	Exergy power inlet	Eq.	Exergy power outlet	Eq.
CJW	$\dot{E}x_{\mathrm{in,CJW}} = \dot{E}x_{15} - \dot{E}x_{16}$	(7)	$\dot{E}x_{\mathrm{out,CJW}} = \dot{E}x_{3} - \dot{E}x_{2}$	(13)
R1	$\dot{E}x_{\mathrm{in,R1}} = \dot{E}x_{9} - \dot{E}x_{10}$	(8)	$\dot{E}x_{\mathrm{out,R1}} = \dot{E}x_{4} - \dot{E}x_{3}$	(14)
SA	$\dot{E}x_{\mathrm{in,SA}} = \dot{E}x_{13} - \dot{E}x_{14}$	(9)	$\dot{E}x_{\mathrm{out,SA}} = \dot{E}x_{5} - \dot{E}x_{4}$	(15)
R2	$\dot{E}x_{\mathrm{in,R2}} = \dot{E}x_{8} - \dot{E}x_{9}$	(10)	$\dot{E}x_{\mathrm{out,R2}} = \dot{E}x_{6} - \dot{E}x_{5}$	(16)
EG	$\dot{E}x_{\mathrm{in,EG}} = \dot{E}x_{11} - \dot{E}x_{12}$	(11)	$\dot{E}x_{\mathrm{out,EG}} = \dot{E}x_{7} - \dot{E}x_{6}$	(17)
MC	$\dot{E}x_{\mathrm{in,MC}} = \dot{E}x_{10} - \dot{E}x_{1}$	(12)	$\dot{E}x_{\mathrm{out,MC}} = \dot{E}x_{18} - \dot{E}x_{17}$	(18)
Component	**Exergy destruction**	**Eq.**	**Exergy efficiency**	**Eq.**
CJW	$\dot{E}x_{\mathrm{D,CJW}} = \dot{E}x_{\mathrm{in,CJW}} - \dot{E}x_{\mathrm{out,CJW}}$	(19)	$\eta_{\mathrm{ex,CJW}} = \dfrac{\dot{E}x_{\mathrm{out,CJW}}}{\dot{E}x_{\mathrm{in,CJW}}}$	(25)
R1	$\dot{E}x_{\mathrm{D,R1}} = \dot{E}x_{\mathrm{in,R1}} - \dot{E}x_{\mathrm{out,R1}}$	(20)	$\eta_{\mathrm{ex,R1}} = \dfrac{\dot{E}x_{\mathrm{out,R1}}}{\dot{E}x_{\mathrm{in,R1}}}$	(26)
SA	$\dot{E}x_{\mathrm{D,SA}} = \dot{E}x_{\mathrm{in,SA}} - \dot{E}x_{\mathrm{out,SA}}$	(21)	$\eta_{\mathrm{ex,SA}} = \dfrac{\dot{E}x_{\mathrm{out,SA}}}{\dot{E}x_{\mathrm{in,SA}}}$	(27)
R2	$\dot{E}x_{\mathrm{D,R2}} = \dot{E}x_{\mathrm{in,R2}} - \dot{E}x_{\mathrm{out,R2}}$	(22)	$\eta_{\mathrm{ex,R2}} = \dfrac{\dot{E}x_{\mathrm{out,R2}}}{\dot{E}x_{\mathrm{in,R2}}}$	(28)
EG	$\dot{E}x_{\mathrm{D,EG}} = \dot{E}x_{\mathrm{in,EG}} - \dot{E}x_{\mathrm{out,EG}}$	(23)	$\eta_{\mathrm{ex,EG}} = \dfrac{\dot{E}x_{\mathrm{out,EG}}}{\dot{E}x_{\mathrm{in,EG}}}$	(29)
MC	$\dot{E}x_{\mathrm{D,MC}} = \dot{E}x_{\mathrm{in,MC}} - \dot{E}x_{\mathrm{out,MC}}$	(24)	$\eta_{\mathrm{ex,MC}} = \dfrac{\dot{E}x_{\mathrm{out,MC}}}{\dot{E}x_{\mathrm{in,MC}}}$	(30)

Observed waste heat recovery supercritical CO_2 system has one mechanical power consumer (TC) and one mechanical power producer (TURB). To obtain all exergy analysis parameters of mechanical power consumer, producer and of the whole system it is necessary to calculate produced, used and useful mechanical power. Each mechanical power is calculated according to the literature [57]. Mechanical power consumed by TC is:

$$P_{\mathrm{TC}} = \dot{m}_1 \cdot (h_2 - h_1)\,, \tag{31}$$

while mechanical power produced by the TURB is:

$$P_{\mathrm{TURB}} = \dot{m}_7 \cdot (h_7 - h_8)\,. \tag{32}$$

Useful mechanical power is the mechanical power surplus produced by the TURB, which is not used for the TC drive. Useful mechanical power can be delivered from the observed waste heat recovery supercritical CO_2 system to any mechanical power consumer (in the observed system useful mechanical power is delivered to the main propulsion propeller shaft, Figure 1). Useful mechanical power produced in the observed supercritical CO_2 system is:

$$P_{\text{Useful}} = P_{\text{TURB}} - P_{\text{TC}} \,. \tag{33}$$

All equations for the calculation of mechanical power producer and mechanical power consumer exergy parameters from the observed waste heat recovery supercritical CO_2 system are arranged according to instructions from the literature [58, 59] and presented in Table 2.

Table 2 - Exergy analysis equations of turbine and turbocompressor from the observed waste heat recovery supercritical CO_2 system

Component	**Exergy power inlet**	**Eq.**	**Exergy power outlet**	**Eq.**
TURB	$\dot{E}x_{\text{in,TURB}} = \dot{E}x_7$	(34)	$\dot{E}x_{\text{out,TURB}} = \dot{E}x_8 + P_{\text{TURB}}$	(36)
TC	$\dot{E}x_{\text{in,TC}} = \dot{E}x_1 + P_{\text{TC}}$	(35)	$\dot{E}x_{\text{out,TC}} = \dot{E}x_2$	(37)
Component	**Exergy destruction**	**Eq.**	**Exergy efficiency**	**Eq.**
TURB	$\dot{E}x_{\text{D,TURB}} = \dot{E}x_{\text{in,TURB}} - \dot{E}x_{\text{out,TURB}}$	(38)	$\eta_{\text{ex,TURB}} = \dfrac{P_{\text{TURB}}}{\dot{E}x_7 - \dot{E}x_8}$	(40)
TC	$\dot{E}x_{\text{D,TC}} = \dot{E}x_{\text{in,TC}} - \dot{E}x_{\text{out,TC}}$	(39)	$\eta_{\text{ex,TC}} = \dfrac{\dot{E}x_2 - \dot{E}x_1}{P_{\text{TC}}}$	(41)

All required exergy analysis equations related to the whole observed waste heat recovery supercritical CO_2 system are composed according to Figure 2 and presented in Table 3. Each equation presented in Table 3 is composed according to the literature [60-62].

Table 3 - Equations for exergy analysis of the whole observed waste heat recovery supercritical CO_2 system

/	Whole system	Eq.
Exergy power inlet	$\dot{E}x_{\mathrm{in,WS}} = \dot{E}x_{11} + \dot{E}x_{13} + \dot{E}x_{15} + +\dot{E}x_{17}$	(42)
Exergy power outlet	$\dot{E}x_{\mathrm{out,WS}} = \dot{E}x_{12} + \dot{E}x_{14} + \dot{E}x_{16} + \dot{E}x_{18} + P_{\mathrm{Useful}}$	(43)
Exergy destruction	$\dot{E}x_{\mathrm{D,WS}} = \dot{E}x_{\mathrm{in,WS}} - \dot{E}x_{\mathrm{out,WS}}$	(44)
	$\dot{E}x_{\mathrm{D,WS}} = \sum \dot{E}x_{\mathrm{D,all\ system\ components}} = \dot{E}x_{\mathrm{D,CJW}} + \dot{E}x_{\mathrm{D,R1}} + \dot{E}x_{\mathrm{D,SA}} + \dot{E}x_{\mathrm{D,R2}} + \dot{E}x_{\mathrm{D,EG}} + \dot{E}x_{\mathrm{D,MC}} + \dot{E}x_{\mathrm{D,TURB}} + \dot{E}x_{\mathrm{D,TC}}$	(45)
Exergy efficiency	$\eta_{\mathrm{ex,WS}} = \dfrac{P_{\mathrm{Useful}}}{\dot{E}x_{\mathrm{in,WS}} - \dot{E}x_{\mathrm{out,WS}} + P_{\mathrm{Useful}}}$	(46)

4. Operating parameters required for the waste heat recovery supercritical CO_2 system exergy analysis

For the exergy analysis of the whole observed waste heat recovery supercritical CO_2 system as well as for the exergy analysis of each its component are required pressures, temperatures and mass flow rates of all fluids in each operating point from Figure 1 and Figure 2. The most of these operating parameters are found in [27] and presented in Table 4, while the others are assumed according to the recommendations from the literature [63] or calculated from the energy balances. Specific enthalpies and specific entropies in each operating point from Figure 1 are calculated by using NIST-REFPROP 9.0 software [28] from known fluid stream pressure and temperature (for each fluid stream). Each fluid stream specific exergy is calculated by using Eq. 4. For the specific exergies calculation must be defined the base ambient state [64]. The base ambient state can be selected arbitrarily and is related to the pressure and temperature of the ambient in which observed component or a system operates. In this analysis, the base ambient state is defined by the ambient pressure of 1 bar (100 kPa) and the ambient temperature of 15 °C (288.15 K).

Table 4 - Fluid stream properties and operating parameters of the observed waste heat recovery supercritical CO_2 system [27]

O.P.*	Medium	Temperature (°C)	Pressure (bar)	Mass flow rate (kg/s)	Specific enthalpy (kJ/kg)	Specific entropy (kJ/kg·K)	Specific exergy (kJ/kg)**
1	**CO_2**	20.22	75.04	38.79	248.99	1.1567	199.32
2	**CO_2**	42.62	259.99	38.79	277.40	1.1784	221.47
3	**CO_2**	74.68	259.47	38.79	345.73	1.3846	230.39
4	**CO_2**	114.48	259.15	38.79	432.30	1.6205	248.98
5	**CO_2**	142.99	253.10	38.79	489.77	1.7667	264.31
6	**CO_2**	153.44	252.78	38.79	508.36	1.8111	270.13
7	**CO_2**	276.45	248.52	38.79	688.09	2.1862	341.76
8	**CO_2**	164.54	76.82	38.79	600.40	2.2112	246.85
9	**CO_2**	148.34	76.35	38.79	581.81	2.1690	240.44
10	**CO_2**	79.72	75.91	38.79	495.24	1.9451	218.40
11	**Exhaust gas**	325.54	2.40	41.85	732.02	4.3445	175.13
12	**Exhaust gas**	164.52	2.30	41.85	565.44	4.0329	98.34
13	**Air**	193.73	2.55	41.10	595.28	4.0693	117.70
14	**Air**	140.57	2.35	41.10	541.04	3.9694	92.24
15	**Water**	80.00	1.30	40.07	335.08	1.0755	26.77
16	**Water**	64.22	1.25	40.07	268.93	0.8839	15.83
17	**Water**	15.00	2.50	152.29	63.22	0.2244	0.15
18	**Water**	30.00	2.30	152.29	125.94	0.4367	1.71

** O.P. = Operating Point (according to operating points presented in Figure 1)*
*** Specific exergies are calculated for the base ambient state*

According to fluid operating parameters presented in Table 4, it can be seen why the Main Cooler (MC) operation has a high influence on the performance of the whole observed waste heat recovery supercritical CO_2 system. In the MC, the difference in specific exergies between cooling water outlet and inlet (operating points 18 and 17, Figure 1) is extremely low because cooling water operating parameters in the MC are very close to the ambient state. Therefore, a general observation of the system operating parameters presented in Table 4 can lead to conclusion that MC will be problematic component and that any system improvements or modifications should involve this component.

5. Exergy analysis results of waste heat recovery supercritical CO_2 system and discussion

The essential element in exergy analysis of mechanical power producers and consumers, as well as in the exergy analysis of the whole system is mechanical power. In the observed system exist one mechanical power consumer (turbocompressor) which consumes mechanical power equal to 1102.02 kW, Figure 3. The turbine is only mechanical power producer in the observed system and it produces mechanical power equal to 3401.50 kW (due to CO_2 expansion through the turbine). The main turbine goal is to ensure sufficient mechanical power for the turbocompressor drive, while the rest of produced mechanical power will be delivered to any mechanical power consumer (useful mechanical power).

According to fluid properties and operating parameters presented in Table 4, produced mechanical power surplus by the turbine (useful mechanical power) in the observed system is 2299.47 kW. Useful mechanical power obtained from the waste heat recovery supercritical CO_2 system will be delivered to the main propulsion propeller shaft, Figure 1. Useful mechanical power produced by the observed supercritical CO_2 system will reduce diesel engine produced mechanical power (and simultaneously diesel engine fuel consumption and exhaust gas emissions will be reduced).

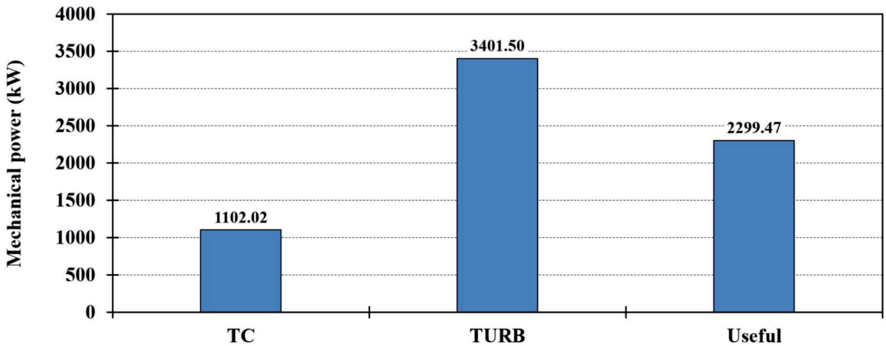

Figure 3 - Produced (turbine), used (turbocompressor) and useful mechanical power from the observed waste heat recovery supercritical CO_2 system

Exergy power inlets and outlets to each component (and from each component) of the observed waste heat recovery supercritical CO_2 system are presented in Figure 4. For each component, the exergy power inlet must be higher than exergy power outlet (the difference between the inlet and outlet of each component is component exergy destruction).

From Figure 4 is clear that all heat exchangers in the observed system (CJW, R1, SA, R2, EG and MC) have notably lower exergy power inlets and outlets in comparison to mechanical power consumer and mechanical power producer (TC and TURB). The turbine has the highest exergy power inlets and outlets of all system components (turbine exergy power inlet is equal to 13256.87 kW while the turbine exergy power outlet is equal to 12976.81 kW), Figure 4.

Observation of heat exchangers only shows that the highest exergy power inlet and outlet has heat exchanger which operates with the fluids of the highest temperature. That heat exchanger is EG in which diesel engine exhaust gas transfer heat to the CO_2

and increase its temperature to the highest possible level, before CO_2 expansion in the turbine.

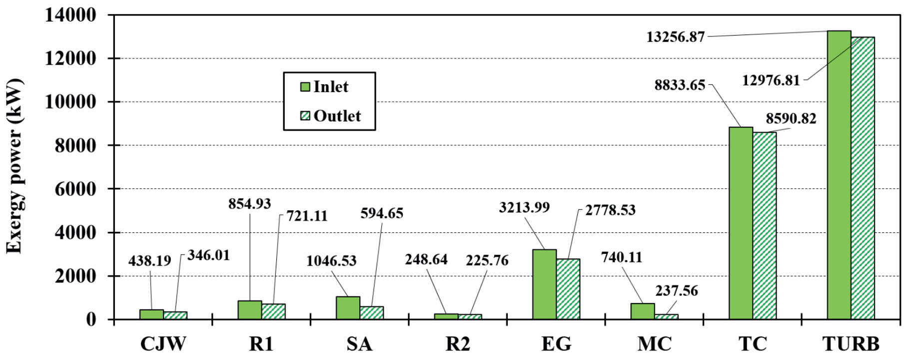

Figure 4 - Exergy power inlet and outlet of all components from the observed waste heat recovery supercritical CO_2 system (CJW - Cooling Jacket Water waste heat exchanger; R1- Recuperator 1; SA - Scavenge Air waste heat exchanger; R2 - Recuperator 2; EG – Exhaust Gas waste heat exchanger; MC – Main Cooler; TC – Turbocompressor; TURB – Turbine)

Exergy destruction of each component from the observed waste heat recovery supercritical CO_2 system is presented in Figure 5.

As expected, the highest exergy destruction equal to 502.55 kW has MC, followed by SA (451.88 kW) and EG (435.46 kW). High exergy destruction in MC can be explained by the fact that cooling water to which the heat is transferred from CO_2 has parameters close to the ambient state and very small specific exergy difference (cooling water at the MC outlet in comparison to cooling water at the MC inlet), what results with high MC exergy destruction. Also, MC has high temperature difference between water at the MC outlet and CO_2 at the MC inlet.

High exergy destruction of SA can be explained by relatively high temperature differences between air at the SA inlet and CO_2 at the SA outlet (around 50 °C). Similar temperature difference as in the SA occurs in EG between combustion gas inlet and CO_2 outlet what is also a reason for increased exergy destruction. The much lower temperature difference between the fluids occur in R1, CJW and R2, and proportionally that three heat exchangers have much lower exergy destruction in comparison to EG, SA and MC. The lowest exergy destruction of all heat exchangers is found in R2 (equal to 22.89 kW).

TURB as a mechanical power producer has higher exergy destruction in comparison to TC. Exergy destructions of both TURB and TC are higher in comparison to heat exchangers R1, CJW and R2, but lower in comparison to heat exchangers EG, SA and MC, Figure 5.

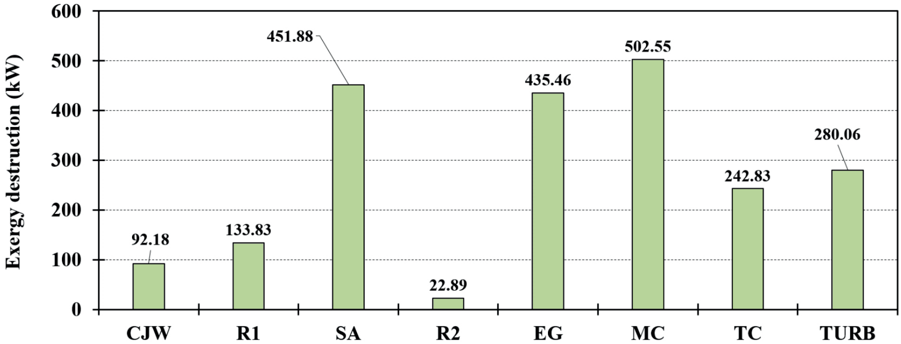

Figure 5 - Exergy destruction of all components from the observed waste heat recovery supercritical CO_2 system (CJW - Cooling Jacket Water waste heat exchanger; R1- Recuperator 1; SA - Scavenge Air waste heat exchanger; R2 - Recuperator 2; EG – Exhaust Gas waste heat exchanger; MC – Main Cooler; TC – Turbocompressor; TURB – Turbine)

The exergy efficiency of each component from the observed waste heat recovery supercritical CO_2 system is presented in Figure 6.

While observing heat exchangers, R2 which has the lowest exergy destruction of all heat exchangers has also the highest exergy efficiency, equal to 90.80%. Simultaneously, MC which has the highest exergy destruction has the lowest exergy efficiency, equal to 32.10% only. EG which has high exergy destruction has at the same time satisfying exergy efficiency of 86.45%. EG exergy efficiency is higher in comparison to exergy efficiency of CJW and R1 (CJW and R1 have notably lower exergy destruction in comparison to EG). Due to high exergy destruction, SA has a quite low exergy efficiency equal to 56.82%.

It should be highlighted that the components which operating parameters are the closest to the ambient state (the closest to the ambient pressure and temperature) has usually low exergy efficiency, due to low differences between inlet and outlet specific exergies. In this research, that components are SA and MC. The confirmation of this statement can be found in [64] where is confirmed that in the marine condensate/feedwater heating system, heater which is the closest to the main condenser (and which operating parameters are the closest to the ambient state) has the lowest exergy efficiency and is highly influenced by the change in the ambient parameters.

Regardless of higher exergy destruction, TURB has notably higher exergy efficiency in comparison to TC (TURB exergy efficiency equals to 92.39% in comparison to exergy efficiency of TC which is equal to 77.97%).

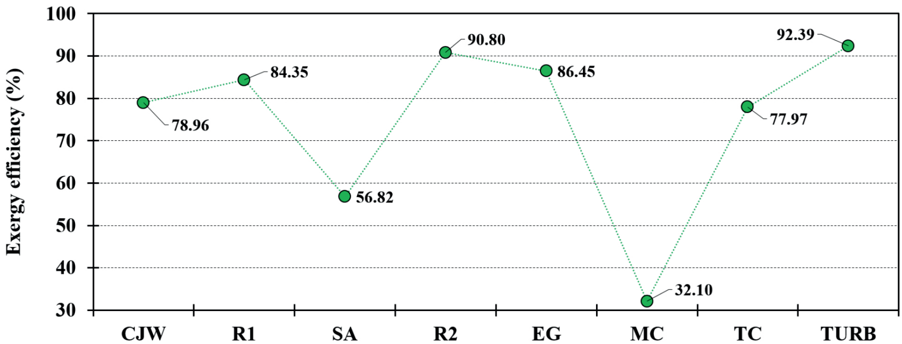

Figure 6 - Exergy efficiency of all components from the observed waste heat recovery supercritical CO_2 system (CJW - Cooling Jacket Water waste heat exchanger; R1- Recuperator 1; SA - Scavenge Air waste heat exchanger; R2 - Recuperator 2; EG – Exhaust Gas waste heat exchanger; MC – Main Cooler; TC – Turbocompressor; TURB – Turbine)

Exergy analysis of the observed waste heat recovery supercritical CO_2 system shows that the most problematic component in this system is Main Cooler (MC) which has the highest exergy destruction and the lowest exergy efficiency of all system components. This analysis also proved the fact mentioned in [27] that MC performance has the dominant influence on the whole system operation. In addition to [27], this analysis also shows that SA performance can be notably improved due to its high exergy destruction and relatively low exergy efficiency.

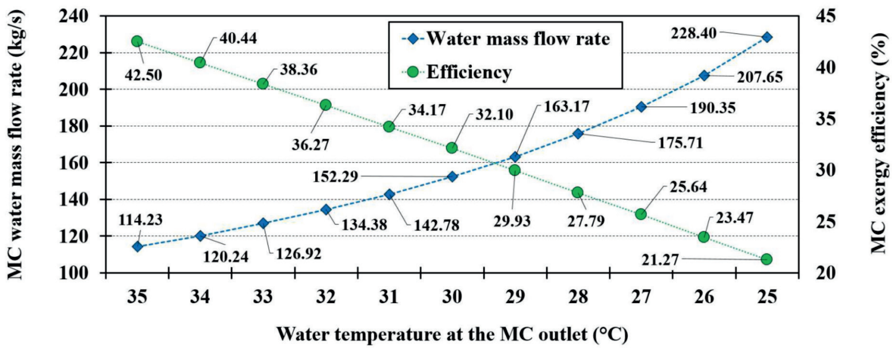

Figure 7 - The change in MC water mass flow rate and exergy efficiency during the change in water temperature at the MC outlet

From previously presented results is clear that the exergy analysis of supercritical CO_2 system detects the Main Cooler (MC) as the most problematic component. Therefore, in this paper is shown one of the possible solutions how the MC performance can be improved.

If it is assumed that the cooling water temperature at the MC outlet is not limited, then the MC exergy efficiency can be increased by decreasing water mass flow rate through MC and simultaneously by increasing water temperature at the MC outlet, Figure 7. In the proposed process cooling water temperature at the MC inlet remains always the same (and equal to 15 °C), Table 4.

In Table 4 can be found that current water temperature at the MC outlet is 30 °C which corresponds to water mass flow rate through MC of 152.29 kg/s and MC exergy efficiency of 32.10%. If water temperature at the MC outlet increases to 35 °C, the water mass flow rate through MC will be reduced to 114.23 kg/s and MC exergy efficiency will increase to 42.50%. Decrease in water temperature at the MC outlet will result with an increase in the water mass flow rate and in MC exergy efficiency reduction, Figure 7. It should be highlighted that the presented one is just one of at least few possibilities how the MC exergy efficiency can be increased (along with a simultaneous decrease in MC exergy destruction).

For the observed waste heat recovery supercritical CO_2 system is interesting to observe the CO_2 temperature increase in all five heat exchangers between TC and TURB, Figure 8. Operating points presented in Figure 8 are related to the operating points from Figure 1 and Table 4.

In the first heat exchanger (CJW), cooling jacket water increases CO_2 temperature for 32.06 °C. After CJW, the first recuperator (R1) increases the CO_2 temperature for 39.8 °C, while SA increases CO_2 temperature for 28.51 °C. Second recuperator (R2) increases the CO_2 temperature for only 10.45 °C, while the last heat exchanger (EG) increases the CO_2 temperature more than all previous four heat exchangers (EG increases CO_2 temperature for 123.01 °C), Figure 8. Therefore, it is important to note that the last heat exchanger (EG) has the dominant influence in the CO_2 heating process.

Analyzed supercritical CO_2 system exploits waste heat from the marine diesel engine in phases and uses waste heat from three fluids (cooling jacket water, air and exhaust gas) for a cascade CO_2 heating. Along with fluids from marine diesel engine, a heat contained in a CO_2 after its expansion in a turbine is effectively used in two recuperators for additional heating purposes with an aim to minimize cumulative heat loss from the diesel engine and the supercritical CO_2 system.

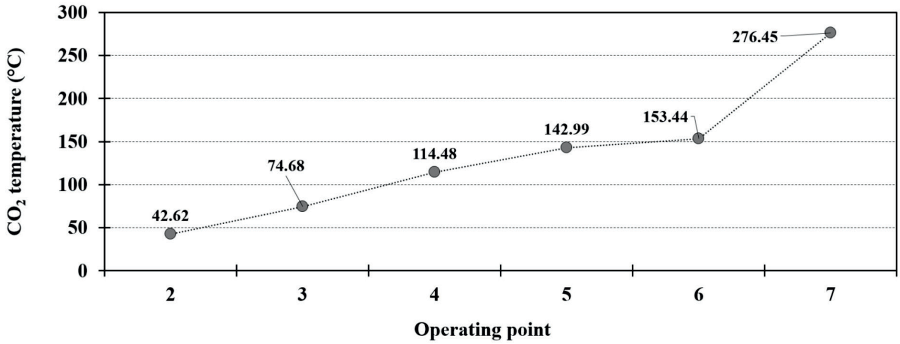

Figure 8 - Increase in CO_2 temperature through all heat exchangers between turbocompressor and turbine

Exergy power inlet, outlet and destruction of the whole observed waste heat recovery supercritical CO_2 system are presented in Figure 9. Whole observed system has exergy destruction equal to 2161.68 kW. Moreover, whole system exergy efficiency is equal to 51.54%. Obtained exergy efficiency is sufficient to prove the applicability of this system in the marine sector by using various waste heat temperature levels from a diesel engine.

When the waste heat recovery supercritical CO_2 system analyzed in this paper is compared with similar system which uses waste heat from marine gas turbine [26] it can be concluded that the whole system observed in this paper has approximately 12% lower exergy efficiency. The reason of lower exergy efficiency can be found in a fact that system observed in this paper uses much lower temperature levels (from marine diesel engine) in comparison to temperature levels which occur at the marine gas turbine exhaust.

Finally, with an aim to control performed calculations, the whole system exergy destruction (calculated by using Figure 2) is compared with a sum of exergy destructions of all system components. Whole system exergy destruction is identical in both calculated ways, which confirm the accuracy of used method.

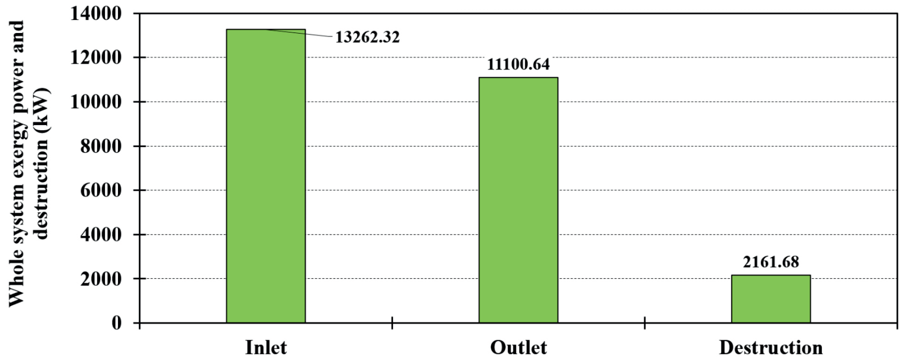

Figure 9 – Exergy power inlet, outlet and destruction of the whole observed waste heat recovery supercritical CO_2 system

Further research related to the observed waste heat recovery supercritical CO_2 system will be based on the application of various artificial intelligence methods and algorithms [65-68] with an aim to observe improvement possibilities. The main goal will be to perform optimization by decreasing exergy destruction and increasing exergy efficiency of each component and whole system.

6. Conclusions

This research presents an exergy analysis of supercritical CO_2 system which uses various waste heat flows from a marine diesel engine with an aim to produce additional mechanical power. Analysis is performed for each individual component from the observed system, as well as for the whole system. The problematic components which operation can and should be improved are detected. For the most problematic component is presented one possibility how its exergy efficiency can be increased (with simultaneous decrease of its exergy destruction). The main conclusions of the performed analysis are:

- The observed waste heat recovery supercritical CO_2 system produces useful mechanical power equal to 2299.47 kW. This useful mechanical power is delivered to the main propulsion propeller shaft what will reduce marine diesel engine fuel consumption and exhaust gas emissions.

- All heat exchangers in the observed system (CJW, R1, SA, R2, EG and MC) have notably lower exergy power inlets and outlets in comparison to TC and TURB.

- The highest exergy destruction of all system components, equal to 502.55 kW has MC, followed by SA (451.88 kW) and EG (435.46 kW). These three heat exchangers have high exergy destructions due to high temperature differences between fluids which exchange heat in each of them.

- MC which has the highest exergy destruction of all system components has the lowest exergy efficiency in the observed system, equal to 32.10% only. Due to high exergy destruction, also SA has a quite low exergy efficiency equal to 56.82%. Any system improvement should firstly be based on these two heat exchangers.

- If the cooling water temperature at the MC outlet is not limited, then the MC exergy efficiency can be increased (along with simultaneous MC exergy destruction decrease), by decreasing water mass flow rate through MC and simultaneously by increase of water temperature at the MC outlet.

- Five heat exchangers (CJW, R1, SA, R2 and EG) are involved in the CO_2 heating process between turbocompressor and turbine. It is interesting that the last heat exchanger (EG) which uses exhaust gas for CO_2 heating, increases CO_2 temperature more than all previous four heat exchangers (EG increases CO_2 temperature for 123.01 °C).

- Whole analyzed waste heat recovery supercritical CO_2 system has exergy destruction equal to 2161.68 kW, while its exergy efficiency is 51.54%. Whole system exergy efficiency is sufficient to prove its applicability in the marine sector by using various waste heat temperature levels from a diesel engine.

- In comparison to a similar CO_2 system which uses waste heat from marine gas turbine exhaust, it can be concluded that the waste heat recovery supercritical CO_2 system analyzed in this paper has approximately 12% lower exergy efficiency. The reason of such occurrence can be found in a fact that the system analyzed in this paper uses much lower waste heat temperature levels (from marine diesel engine) in comparison to temperature levels which occur at the marine gas turbine exhaust.

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NOMENCLATURE

Latin Symbols:

$\dot{E}x$	total fluid exergy flow, kW
h	specific enthalpy, kJ/kg
$\dot{m}$	mass flow rate, kg/s
P	mechanical power, kW
$\dot{Q}$	energy transfer by heat, kW
s	specific entropy, kJ/kg·K
T	temperature, °C or K
$\dot{X}_{\mathrm{heat}}$	exergy transfer by heat, kW

Greek symbols:

ε	specific exergy, kJ/kg
η_{ex}	exergy efficiency, -

Subscripts:

0	base ambient state
D	exergy destruction
in	inlet
out	outlet
WS	whole system

Abbreviations:

CJW	Cooling Jacket Water waste heat exchanger
EG	Exhaust Gas waste heat exchanger
MC	Main Cooler
R1	Recuperator (the first one)
R2	Recuperator (the second one)
SA	Scavenge Air waste heat exchanger
TC	Turbocompressor
TURB	Turbine

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