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Optimizing Energy Recovery from Marine Diesel Engines: A Thermodynamic Investigation of Supercritical Carbon Dioxide Cycles

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P A P E R I N F O

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A B S T R A C T

Since the population and economic activities have increased energy demands, researchers and scientists have turned to recover wasted energy in various systems. This study aims to maximize energy recovery from marine diesel engines through heat exchange in four different types of wasted streams (exhaust gas, engine coolant, and engine oil coolant). In this research, four cycles of carbon dioxide critical recovery have been designed and modeled to utilize wasted heat energy from marine diesel engines (MAN B&W L35MC6-TII). The EES engineering software has been used for mathematical calculations. In each cycle, the effect of various parameters such as compressor outlet pressure, compressor inlet temperature, and turbine inlet temperature on output power, exergy efficiency, and compressor power consumption has been investigated. The results of this study indicate that the use of a heat recovery cycle in a diesel engine prevents the loss of a significant amount of energy in the engine. Additionally, increasing the fluid temperature at the compressor inlet reduces the output power and exergy efficiency in all recovery cycles. Increasing the fluid temperature at the turbine inlet reduces the compressor power consumption, increases the output power, and enhances the exergy efficiency in all recovery cycles. When the engine body coolant is used in the recovery cycle, the output power and energy system efficiency increase. Moreover, the cycle includes a generator to recover the heat output of carbon dioxide from the turbine. According to the findings of this study, despite the highest turbine output power (611.5 kW) belonging to the exhaust gas recovery system with two heat exchangers, the recovery system with a single heat exchanger has the highest usable output power (228.3 kW). This system had the highest energy and exergy efficiency of 17.72% and 12.85%, respectively.

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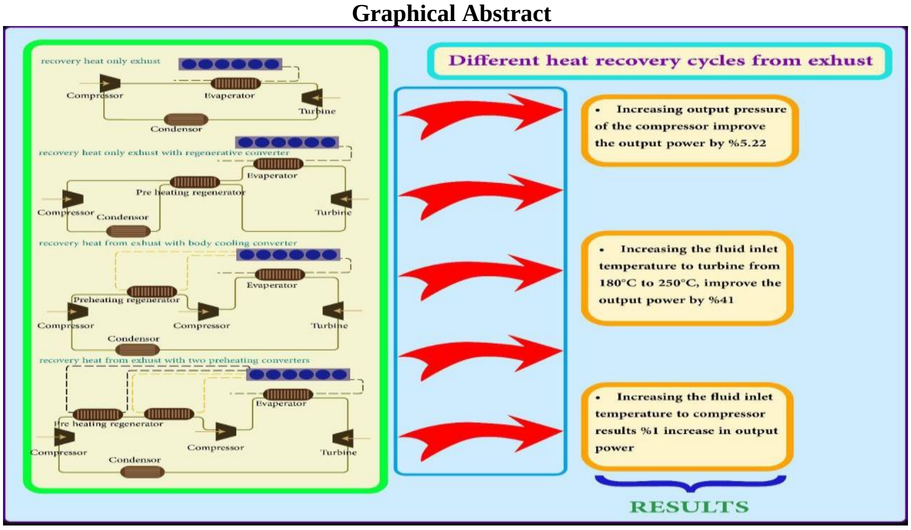
Graphical Abstract



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Highlights

- In this study, four heating recovery cycles of carbon dioxide were processed with EES.
- Modeling four critical recovery heat generator cycles for carbon dioxide
- Analysis and comparison of effective parameters of the heat recovery cycle
- The exhaust recovery with a regenerative converter yields an output power of 228.3 kW.
- The highest turbine output power (611.5 kW) exhaust recovery cycle with two converters

NOMENCLATURE

BR	V	T_{out}	Temperature outlet
$\dot{m}_c$	Mass flow	T_{in}	Temperature inlet
h_s	Sritical isentropic enthalpy	W_{net}	Net power output
η	Efficiency	**Greek Symbols**	
Q	Heat transfered	ξ	Efficiency of heat exchangers
C_p	Specific heat capacity constant pressure		

1. INTRODUCTION

According to International Maritime Organization's third study of greenhouse gas emissions from 2007 to 2012, approximately 3% of greenhouse gas emissions (corresponding to 1 billion tons of emissions per year) were associated with ships. Additionally, ships produce 13% and 15% of global NOx and SOx emissions annually (1). The diesel engine is the most common engine used in vehicles for energy production. About 96% of ships use diesel power cycles to generate their energy. Diesel engines cannot be replaced soon due to their high energy density, low fuel costs, and high thermal efficiency. The fuel efficiency of large diesel engines is about 50%. As waste heat, the remaining energy is transferred to the environment.

Heat recovery systems can convert the remaining wasted energy to electrical and mechanical energy, providing propulsion force and auxiliary power without increasing fuel consumption.

The supercritical carbon dioxide power cycle specializes in heat recovery systems. Feher (2) designed this system in 1967. High thermal efficiency, smaller components, and smaller scale of the entire parts of the system are potential advantages of this cycle. Supercritical conditions are required for this cycle to work. CO_2 works as a critical condition working fluid at 31 ºC, 7.35 MPa; which improves heat transfer between the working fluid and the heat source.

S-CO_2 power cycle analysis and optimization have been studied in a variety of ways to date, including the use of different types of heat sources (3, 4), heat exchangers of high efficiency (5, 6), cycle design and optimizations (7-9) and aerodynamics for turbomachines (10, 11). Researchers have conducted a variety of studies on S-CO_2 power cycles.

Vickers (12) has analyzed and optimized different settings of the S-CO_2 Rankine cycle and compared them with other working fluids plus, the development of the initial sample, the design of components, and challenges. According to Crespi et al. (13), S-CO_2 power cycles possess numerous advantages and applications. They also compared different S-CO_2 cycle designs. Li et al. (14) studied the recent development process of the S-CO_2 power cycle and its applications across various industries. Shi et al. (15) studied and classified the concepts of cycle, layout, fuel, applications, and different operational conditions.

The S-CO_2 cycle was improved by Singh and Pedersen (16) while adding a regenerator and a preheater to extract energy from exhaust gases and engine coolant (17). Based on simulation results, this new engine configuration has a net power of 43.8 kW to 9.0 kW, an increase of 150% over the base configuration (3.6 kW). Using waste heat from the exhaust and cooling fluid, Yu et al. (18) evaluated different CO_2 critical power cycle configurations for a 1.7-liter engine with natural gas fuel and a nominal power of 80 kW. Results show that internal heat exchangers and preheaters increase the system's thermal efficiency and increase electrical output by up to 18 kW.

Mikielewicz and Mikielewicz (19) claimed the S-CO_2 cycle has a relative improvement of about 5% compared to the organic cycle. Based on their comparison of non-critical and supercritical cycles, Schuster et al. (20) found supercritical cycles to have an 8% relative efficiency increase due to lower exergy reduction.

Yu et al. (18) examined and analyzed the carbon dioxide cycle with a ship's engine as part of a thermodynamic and economic study. Their results showed that the system was 44.42% and 39.05% efficient in terms of energy and energy efficiency. Their report states, the cost of produced energy is \$9.28/GJ. Their results indicate, the inlet temperature of the turbine does not affect the efficiency of the recovery system, but its work decreases with an increase in temperature.

Li and Wang (21) investigated the thermodynamic performance of a hybrid cascade carbon dioxide cycle for recovering waste heat. based on their results, the output power of HCCC is 17.02% higher than the series cascade carbon dioxide cycle. They also reported the specific heat conductance of HCCC is 7-11% lower than the series cascade carbon dioxide cycle (SCCC).

The performance of an organic Rankine cycle system for recovery of waste heat was assessed. Ie was tested with 5 working fluids with low ozone depletion potential and low global warming potential. based on their finding, butane has the best performance among others. the overall power in the aforementioned system reaches 1048kW and the efficiency of the system is 36.47%.

Qin et al. (22) studied a novel system for waste heat recovery. their system consisted of combination of supercritical CO_2 recompression Brayton cycle and transcritical CO_2 refrigeration cycle. Their results showed that higher inlet pressure of the high-pressure compressor leads to enhancement of thermal performance of system; however, it decreases exergy performance of the system. Other valuable works have been conducted in the fields of thermal and heat systems, thermodynamics, thermal cycles, and renewable energies, which can be referred to in literature (23-26).

The main source of thermal energy loss in marine transportation, particularly in diesel engines, is the exhaust gases, which constitute about 50% of the total fuel energy. Approximately half of this energy is transferred to the environment without performing any useful work. The most significant sources of heat loss are the exhaust gases from combustion. After that, heat losses in the cooling fluid and heat losses in the lubricating fluid are the subsequent contributors. Improving engine efficiency through the absorption of waste heat, whether through heat recovery systems or by utilizing energy for heating services, can lead to reduced fuel consumption and lower greenhouse gas emissions into the atmosphere.

Heat recovery systems can utilize the remaining waste energy for the production of electrical and mechanical power, which can be used to provide propulsion and auxiliary power in marine systems without increasing fuel consumption. In recent years, many researchers have focused on increasing heat recovery and maximizing the use of thermal energy through the carbon dioxide (CO_2) transcritical cycle. This study specifically investigates heat recovery from a marine diesel engine (MAN B&W L35MC6-TII) using a transcritical CO_2 cycle. The chosen cycle is highlighted for its higher efficiency, compatibility with various waste heat sources in the discussed diesel engine within the temperature range of the transcritical CO_2 cycle, and the compact and space-efficient equipment required for marine vessels.

Present study also aims to achieve maximum energy recovery by using various heat exchangers on different streams (exhaust gas, engine cooling fluid, cooling air, and lubricating oil). To attain the highest power output, four transcritical CO_2 cycles with different components are discussed and investigated in this study. Additionally, the impact of changes in various parameters, such as compressor outlet pressure, compressor inlet temperature, and turbine inlet temperature, on power output, energy efficiency, and exergy of the system is explored for each cycle. It is worth noting that the thermodynamic modeling of these cycles has been performed using engineering software, EES.

By employing thermodynamic modeling in the Engineering Equation Solver (EES) software, the study aims to provide valuable insights into the feasibility and efficiency of transcritical CO_2 cycles for waste heat recovery in marine applications. The ultimate goal is to contribute to the ongoing efforts in sustainable marine transportation by improving energy efficiency, reducing fuel consumption, and minimizing environmental impact.

2. PROBLEM DESCRIPTION

Diesel engines play a crucial role in power generation for various transportation vehicles, especially in the maritime industry, where 96% of ships meet their energy needs through diesel power cycles. Due to their high energy density, low fuel costs, and superior thermal efficiency compared to alternatives, diesel engines remain irreplaceable within foreseeable timeframes. Large-scale diesel engines typically exhibit an efficiency of around 50%, with the remaining released energy transferred to the environment as waste heat. Improving engine efficiency, whether through heat recovery systems or utilizing energy for heating services, can reduce fuel consumption and greenhouse gas emissions.

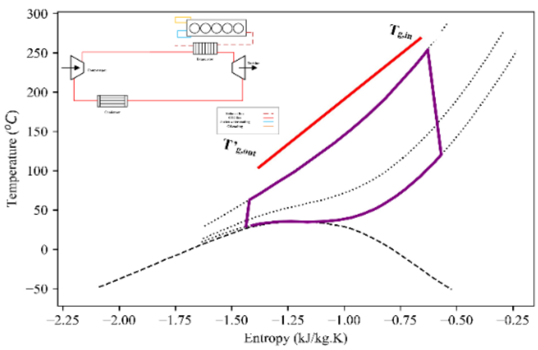
One of the byproducts of combustion in exhaust emissions is carbon dioxide, with temperatures exceeding 100 degrees Celsius. Approximately, 50% of the total fuel energy is transferred to the environment without performing any useful work. Consequently, critical carbon dioxide recovery systems are employed to convert wasted energy into electrical and mechanical energy. Heat recovery systems can be designed to operate on a single heat source or a combination of various heat sources. Depending on the waste heat source, the recovery system offers three alternatives: (1) utilizing only exhaust gases for producing water at a suitable temperature; (2) employing exhaust gases and cooling water from the engine body; and (3) utilizing all waste heat sources for maximum recovery. For maritime applications, heat recovery systems must possess high efficiency, compatibility with ship operational specifications, easy integration with other power systems on board, and compact equipment due to space and weight constraints.

A MAN B&W L35MC6-TII turbocharged linear 6-cylinder engine is studied in this paper. The exhaust outlet and the cooling of the engine body have been studied with the heat recovery system. An overview of the engine output parameters and environmental conditions are summarized in Table 1. Fuel is assumed to burn completely in the combustion chamber. On the basis of this result, 15.2% of the exhaust's mass gases will be carbon dioxide, 6% water vapor, 73% nitrogen, and 5.8% oxygen. The exhaust outlet temperature cannot be less than 100°C based on the dew point temperature of combustion products. In this study, pure water is considered to be the cooling fluid of the engine body. As a result of this research, the amount of heat removed from engine cooling fluid, lubricating oil of the body, cooling air entering the engine, and exhaust gas from the engine has been calculated under constant environmental conditions.

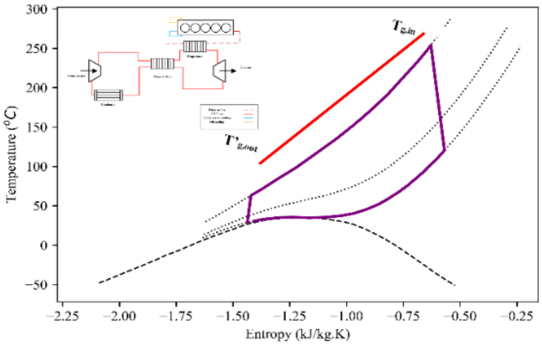
Various heat recovery cycle designs have been studied and put forward in this study. Figure 1 illustrates how it increases the energy extracted from the engine system through the converter and heat recovery in the engine cooling and fluid return routes. The exhaust gas in system A is used as a source of energy harvesting. To

TABLE 1. Engine operating environment and nominal power

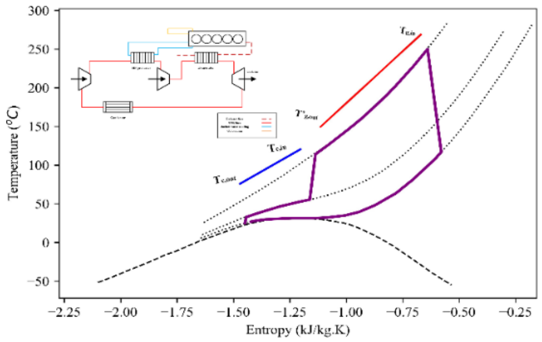
Parameter	Value
Max power	3900 kW
Max round	210 rpm
Power at point M	3315 kW
Engine speed at point M	189.0 rpm
optimal power	3315 kW
Engine speed at the optimum point	189.0 rpm
Power at point s	2652 kW
temperature	20 °C
Turbocharger inlet temperature	18 °C
Ambient pressure	1013 mbar
Exhaust back pressure	300 mm WC



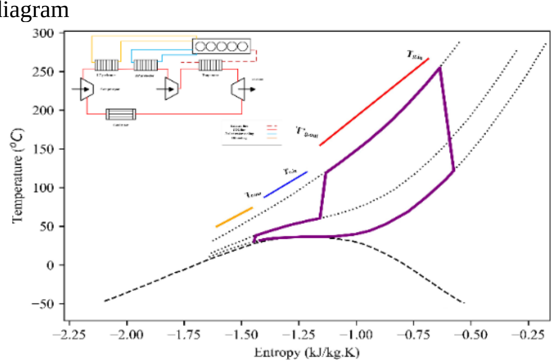
a) Thermal recovery cycle in the temperature-entropy diagram



b) Thermal recovery cycle in the temperature-entropy diagram



c) Thermal recovery cycle in the temperature-entropy diagram



d) Thermal recovery cycle in the temperature-entropy diagram

Figure 1. Diagram of examined cycles

improve heat recovery efficiency, heat exchangers and heat recovery are used in B&P&D systems.

3.GOVERNING THE EQUATIONS

To calculate the dimensions of the heat exchanger and pumps in the system, it is essential to calculate the amount of heat loss in the system. The output parameters of the engine at maximum power and maximum speed under standard laboratory conditions are provided in the literature (27). However, it should be noted that the engine does not always operate under maximum conditions, depending on environmental conditions.

To calculate the heat loss when the engine operates at the power and speed corresponding to point M (P, n), the following relationship can be used:

$$Q_{air\%} = (P_M / P_{L1})^{1.68} \times (n_M / n_{L1})^{-0.83} \times k_O \quad (1)$$

$$k_O = 1 + 0.27 \times (1 - P_O / P_M) \quad (2)$$

In this study, it is assumed that $P_o = P_M$ (the power at point M relative to the optimal power of the system), so k_o is equal to 1.

To calculate the value of $Q_{air\%}$, two parameters $P_{M\%}$ and $n_{M\%}$ are needed, which represent the relative power to the maximum power (L1) and the relative speed to the maximum speed, respectively. These values can be calculated using the following relationships: Additionally, the equation related to calculating the optimal power to point M power ratio is also provided.

$$P_{M\%} = P_M / P_{L1} \quad (3)$$

$$n_{M\%} = n_M / n_{L1} \quad (4)$$

$$P_{O\%} = P_O / P_M \quad (5)$$

The value of the heat loss coefficient for the working fluid can be calculated either from the following relationship or from the relevant literature (27), using the relative power and speed.

$$Q_{jw\%} = e^{(-0.0811 \times \ln(n_{M\%})) + 0.8072 \times \ln(P_{M\%}) + 1.2614} \quad (6)$$

The heat loss coefficient of the working fluid can be calculated using the following relationship:

$$Q_{lub\%} = 67.3009 \times \ln(n_{M\%}) + 7.6304 \times \ln(P_{M\%}) + 245.0714 \quad (7)$$

The amount of heat loss in the operational state of the engine for each of the above-mentioned loss systems is obtained by multiplying the maximum state values under laboratory conditions by the heat loss coefficient representing them. The following relationships illustrate this concept:

$$Q_{air,M} = Q_{air.L1} \times Q_{air\%} \quad (8)$$

$$Q_{jw,M} = Q_{jw.L1} \times Q_{jw\%} \quad (9)$$

$$Q_{lub,M} = Q_{lub.L1} \times Q_{lub\%} \quad (10)$$

The value of the fluid flow rate (sea water), which absorbs the waste heat in the above-mentioned methods, is obtained from the following equations:

$$V_{cw,air,M} = V_{cw,air.L1} \times Q_{air\%} \quad (11)$$

$$V_{cw,jw,M} = V_{cw,jw.L1} \times Q_{jw\%} \quad (12)$$

$$V_{cw,lub,M} = V_{cw,lub.L1} \times Q_{lub\%} \quad (13)$$

In the above equations, the value of V represents the fluid flow rate. The quantities and exhaust gas temperature for the diesel engine used in this study under laboratory conditions are provided in literature (27). However, to determine the amount and temperature of the exhaust fluid under operational and environmental conditions, parameters such as:

- knowing the maximum continuous rating of the engine (point M).
- Engine power and speed at point M.
- ambient conditions and the backpressure of the exhaust gas.
- knowing the point of continuous service of the engine (point S).

are necessary. To calculate the temperature and volume of the exhaust gas in real conditions, the following relationships are used:

$$M_{exh} = M_{L1} \times \frac{P_M}{P_{L1}} \times \left(1 + \frac{\Delta m_{M\%}}{100}\right) \times \left(1 + \frac{\Delta M_{amb\%}}{100}\right) \times \left(1 + \frac{\Delta m_{s\%}}{100}\right) \times \frac{P_{s\%}}{100} \pm 5\% \quad (14)$$

$$T_{exh} = T_{L1} + \Delta T_M + \Delta T_o + \Delta T_{amb} + \Delta T_s \pm 15^\circ C \quad (15)$$

Which M_{exh} and T_{exh} represent the value and temperature of the exhaust gas, respectively. T_{L1} and M_{L1} indicate the temperature and volume of the exhaust gas at the maximum state.

ΔT_o represents the changes in exhaust temperature if point M is below the optimal engine point.

$$\Delta T_O = -0.3 \times (100 - P_{O\%} \times 100) \quad (16)$$

In Equations 14 and 15, $\Delta m_{M\%}$ and ΔT_M represent the relative changes to the maximum values in the quantity and temperature of the exhaust gases. These values can be obtained from the corresponding formulas and also from the relevant figures in literature (27).

$$\Delta m_{M\%} = 14 \times \ln(P_M / P_{L1}) - 24 \times \ln(n_M / n_{L1}) \quad (17)$$

$$\Delta T_M = 15 \times \ln(P_M / P_{L1}) + 45 \times \ln(n_M / n_{L1}) \quad (18)$$

The quantities $\Delta M_{amb\%}$ and ΔT_{amb} in Equations 14 and 15 represent the variations in the exhaust output quantity and the temperature change at the outlet relative to the standard iso-ambient state, and they can be calculated using the following formulas:

$$\Delta M_{amb\%} = -0.41 \times (T_{air} - 25) + 0.03 \times (p_{bar} - 1000) + 0.19 \times (T_{CW} - 25) - 0.011 \times (\Delta p_M - 300)\% \quad (19)$$

$$\Delta T_{amb} = 1.6 \times (T_{air} - 25) - 0.01 \times (p_{bar} - 1000) + 0.1 \times (T_{CW} - 25) - 0.05 \times (\Delta p_M - 300) \quad (20)$$

In Equations 14 and 15, the quantities $\Delta M_{s\%}$ and ΔT_s, respectively, represent the relative changes in mass and temperature of the fluid at the exhaust outlet from the point with load Ps, under conditions where the system is not operating at maximum power. To calculate these quantities, based on literature (27) can be consulted, and the following equations can also be utilized:

$$\Delta m_{s\%} = 37 \times (P_s / P_M)^3 - 87 \times (P_s / P_M)^2 + 31 \times (P_s / P_M) + 19 \quad (21)$$

$$\Delta T_s = 280 \times (P_s / P_M)^2 - 410 \times (P_s / P_M) + 130 \quad (22)$$

Figure 1 illustrates the T-s diagram for critical carbon dioxide thermal recovery cycles. The following relationships govern all cycles of this study:
In stages 1-2, the critical fluid pressure is increased in the compressor.

$$\dot{W}_{comp} = \frac{\dot{m}_{co_2}(h_{2s} - h_1)}{\eta_{comp}} \quad (23)$$

After compression, ith_{2s} represents the critical isentropic enthalpy of carbon dioxide and η_{comp} is equal to the efficiency of the compressor used to compress the gas. In steps 2-3, critical carbon dioxide exchanges heat with the cooling fluid of the engine body through the converter, increasing its temperature.

$$Q_{preh,LT} = \dot{m}_{co_2}(h_{3`} - h_2) = \dot{m}_{jw}.C_{pjw}.(T_{jw,in} - T_{jw,out}).\xi \quad (24)$$

If the system has preheated with low temperature

$$Q_{preh,LT} = \dot{m}_{co_2}(h_3 - h_{3`}) = \dot{m}_{lub}.C_{plub}.(T_{lub,in} - T_{lub,out}).\xi \quad (25)$$

$T_{jw,out}$ What is the exhaust gas temperature after the evaporative exchanger? This study ξ represents the efficiency of heat exchangers. To prevent acid formation in exhaust gases $T_{jw,out}$ should be higher than the dew point.

$$Q_{preh,reg} = \dot{m}_{co_2}(h_4 - h_3) = \dot{m}_{co_2}(h_6 - h_7).\xi \quad (26)$$

Based on the following correlation, the output temperature difference in heat recovery is 6°C.

$$\text{Min}\big((T_6 - T_4), (T_7 - T_3)\big) = \Delta T \quad (27)$$

Evaporating heat exchangers (processes 4-5) follow the same equation as preheating exchangers at increasing temperatures.

$$Q_{evap} = \dot{m}_{co_2}(h_5 - h_4) = \dot{m}_{co_2}.C_{p\,gas}(T_{gas,in} - T_{gas,out}).\xi \quad (28)$$

According to the following equation, the output temperature difference in the regenerative heat exchanger is 6°C.

$$\text{Min}\big((T_{gas,in} - T_5), (T_{gas,out`} - T_4)\big) = \Delta T \quad (29)$$

For all cycles, process 5-6 reduces pressure and temperature in the turbine using the following relationship:

$$\dot{W}_{turbine} = \dot{m}_{co_2}(h_5 - h_{6s}).\eta_{turbine} \quad (30)$$

This equation h_{6s} represents the isentropic enthalpy of the system's working fluid and $\eta_{turbine}$ turbine efficiency. The following equation can be used to calculate the power consumption of Process 7-1, which involves cooling the working fluid in the condenser.

$$Q_{cond} = \dot{m}_{co_2}(h_7 - h_1).\xi \quad (31)$$

From the following equation, we can calculate the net output power of the system:

$$W_{net} = W_{turbine} - W_{pump} \quad (32)$$

The following equation calculates the thermal efficiency of the system:

$$\eta_{thermal} = W_{net}/(Q_c - W_{net}) \quad (33)$$

4. ASSUMPTIONS AND CONDITIONS

A list of design parameters and critical carbon dioxide thermal recovery cycles is provided in Table 2. A minimum temperature close to the carbon dioxide critical point has been chosen so that the fluid at the compressor inlet always remains critical. In this wprk, the maximum cycle pressure of 35 MPa was considered. A constant inlet temperature 250ºC was used. EES software has been used to model the thermodynamics of the carbon dioxide critical cycle. The following factors have been taken into account when modeling this cycle:

- Each heat exchanger has a pressure drop of 0.1 MPa

TABLE 2. Engine operating environment and nominal power

Parameter	Amaount
Minimum cycle temperature	25 ºC
Maximum cycle pressure	35MPa
Compressor efficiency	0.8
Turbine efficiency	0.8
Output temperature difference	6
Heat exchanger efficiency	0.95
Maximum cycle temperature	250 ºC

- Carbon dioxide reaches a stable state at the output of each heat exchanger
- Each converter has a 95% thermal efficiency
- Pipeline pressure drops during the cycle are ignored

There has been no consideration of changes in potential energy and kinetic during the cycle.

5. RESULTS

Using a diesel engine (MAN B&W L35MC6-TII), four thermal recovery cycles with carbon dioxide have been investigated to achieve the highest recovery power.

A variety of parameters, such as the temperature of the fluid entering the compressor, the temperature entering the turbine, and the pressure leaving the compressor, have been studied in the following sections to determine the influence of these parameters on a variety of cycle parameters, including cycle power, compressor power consumption, and cycle efficiency.

In order to verify the accuracy of the results obtained through the method used in this study and to compare the present work with a similar study, a comparative analysis was conducted with the study by Yu et al. (28). Yu et al. (28) conducted a theoretical study on the thermodynamic perspective of the carbon dioxide critical cycle. During this study, they examined four different forms of the carbon dioxide critical cycle. Figures 2 illustrate the comparison between the current research and the study mentioned above. According to Figures 2, it is clear that the results show good consistency with each other. The average error of mathematical calculations is also 5%.

Figure 3 illustrates the effects of a change in compressor output pressure and inlet temperature of 25 °C on turbine, compressor, and system output power. When the output pressure of the compressor is increased from 28 to 35 MPa, both the power consumption of the compressor and the turbine's production power increase. However, the turbine's production power is growing faster than the compressor's power consumption. Consequently, the output power of the system increases as the output pressure increases. Increasing the compressor output pressure results in higher system output power because this pressure rise leads to increased turbine power. As the compressor output pressure increases, fluid enters the turbine, and the turbine operates more efficiently due to the pressure boost, generating more power. This enhanced turbine power ultimately contributes to an increase in the overall system output power. The power consumption of the compressor increases from 116.4 to 185.2 kW when the output pressure of the compressor is boosted from 28 to 35 MPa. Meanwhile, the turbine's output increases from 292.4 to 349.6 kW. As the pressure increases, the output power of the system increases by 5.22%.

Under the influence of temperature changes in the working fluid input to the compressor, Figure 4 shows the changes in turbine output power, compressor power consumption, and system output power. It can be seen that as the inlet temperature of the compressor increases, the power consumption of the compressor and the production power of the turbine increase. As a result of the change in the properties of carbon dioxide, as the critical point approaches, an increasing behavior of compressor and turbine power changes from linear to logarithmic. Elevating the compressor's inlet fluid temperature contributes to a boost in the system's output power by augmenting fluid pressure. This heightened pressure enhances the turbine's efficiency, leading to increased power output and consequently raising the overall system's output power. Based on the results of this study, the compressor's power consumption grew from 164.4 to 220.3 kW when the inlet temperature was raised from 25 to 31 °C; however, the turbine's power output grew from 349.6 kW to 406.4 kW when the inlet temperature was changed from 25 to 31 °C. Lastly, the output power of the recovery system increased by a very small amount (>1%).

The effect of changes in turbine inlet temperature on the performance parameters of the system was investigated as shown in Figure 5. Based on the data

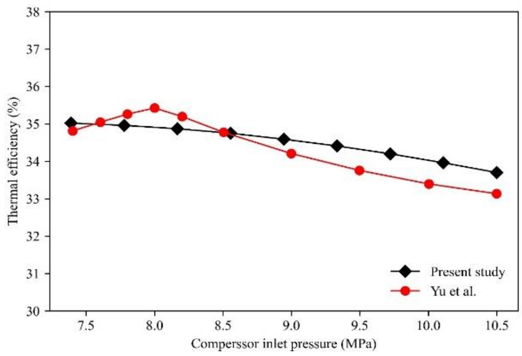


Figure 2. Comparison of the Present Study with the Work of Yu et al. [29]

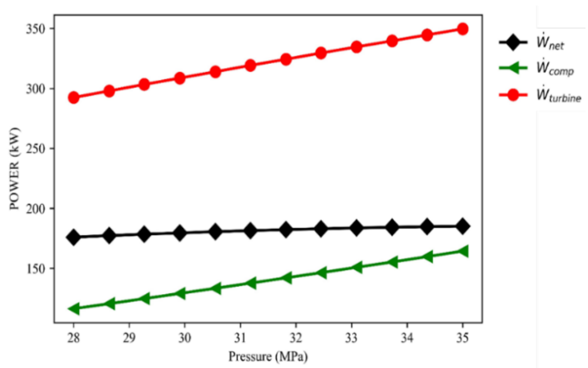


Figure 3. The effect of increasing compressor pressure on turbine power, compressor power, and system output power

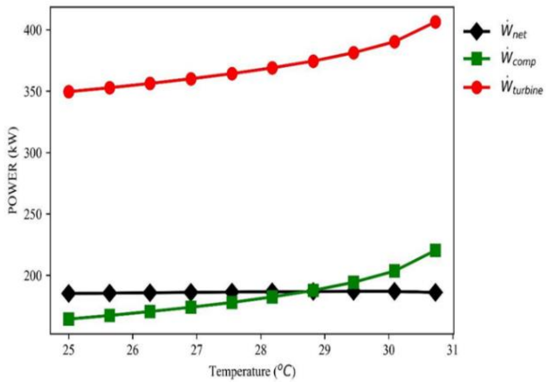


Figure 4. Effects of increasing compressor inlet temperature on turbine power, compressor power, and system useful power

found in this research, an increasing in the inlet temperature to the turbine has a significant effect on the production power of the turbine. However, the power consumption of the compressor decreases (since the rate of heat transfer in the heat exchanger is constant, the outlet temperature of the compressor increases at a constant pressure. It was found that it leads to a decrease in the work of the compressor), which contributes to the output work of the recovery system. By increasing the inlet temperature of the turbine from 180 ° to 250 °C, the production power of the turbine and the power consumption of the compressor decrease by 79.2 and 25.1 kW. As a result, the output power of the system rises by 41%.

In Figure 6, the effect of inlet temperature on working fluid mass rate, system output power, and system energy efficiency and efficiency has been investigated at different pressure ratios. Based on Figure 6-a, it can be seen that with an increase in the inlet temperature of the compressor at various pressures, the mass flow rate entering the system increases because the temperature rise reduces the fluid density; allowing a fixed volume of fluid to enter the compressor, resulting

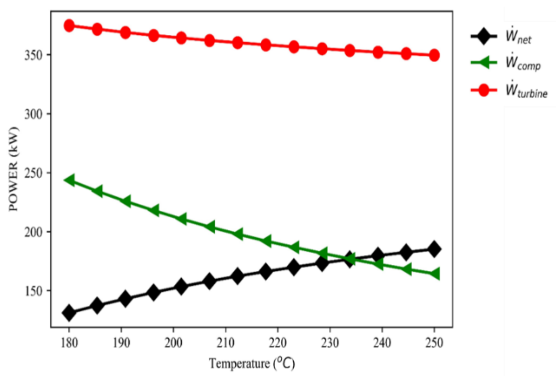
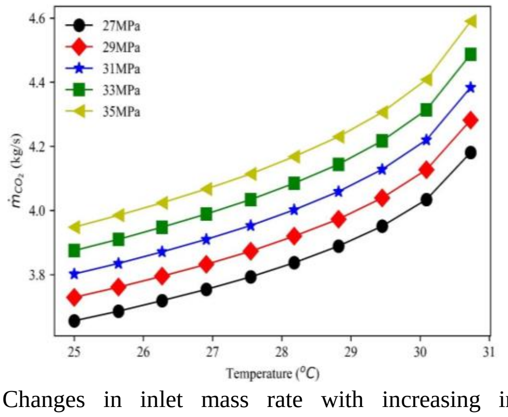


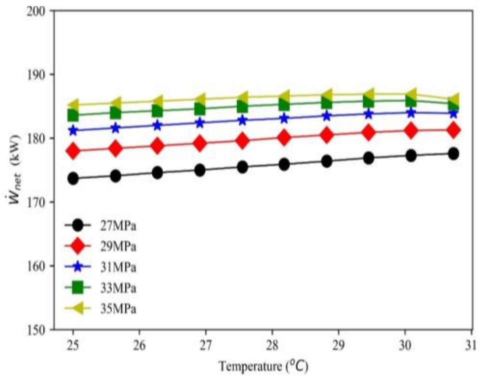
Figure 5. Effects of changes in turbine inlet temperature on turbine power, compressor, and system output power

in an increase in the mass flow rate (given the constant inlet temperature of the turbine, the turbine's power production depends on the system's mass flow rate). Although the fluid mass flow rate also increases with an increase in the pump's outlet pressure, this increase, unlike the temperature-induced mass flow rate rise, occurs at a relatively constant rate. Additionally, the mass flow rate increases as it approaches the critical temperature as the physical properties of carbon dioxide undergo significant changes near the critical point. According to Figure 6-b, the output power of the system changes as a function of the changes in the system inlet temperature in the ratio of different compressor pressures. It can be seen that the output power of the system increases as the inlet temperature to the compressor increases. While the output work of the system increases as the compressor pressure increases, the rate at which the output work increases decreases as the compressor pressure increases. This phenomenon can be attributed to the temperature rising, leading to an increase in mass flow rate. The elevated mass flow rate contributes to improve turbine efficiency, ultimately culminating in a higher overall power output for the system. According to the results obtained in this research, by increasing the inlet temperature to the compressor from 25 to 31 °C, the output power of the system with pressures of 27, 29, 31, 33 and 35 Mpa, respectively, increases by 5.01, 4.90, 2. 5, 1.5, and 1.4%, respectively. The changes in the thermal efficiency of the system from the point of view of energy are illustrated in Figure 6-c. The thermal efficiency of the system increases when the inlet temperature of the compressor is increased from 25 to 31 °C at low pressures (27-29 MPa). Up to 30 °C, this trend continues at high pressures (29-35 MPa). In the ratio of high pressures, however, increasing the inlet temperature from 30 to 31 °C causes a decrease in thermal efficiency. In general, increasing the compressor pressure increases the system's thermal efficiency. As a result of this research, the system with the highest thermal efficiency has a pressure of 35 MPa and an inlet temperature of 30 °C (14.51%). There is also a low thermal efficiency of the system at a pressure of 27 MPa at a temperature of 25 °C (13.48%). Figure 6-d depicts the change in exergy efficiency as the inlet temperature increases to the compressor under different pressure ratios. As expected, the exergy efficiency of the system increases as the inlet temperature increases. Despite the fact that this increase rate occurs almost continuously at low pressures (27, 29, and 31 MPa), at high pressures (33 and 35 MPa), the exergy efficiency of the system decreases as it approaches the critical temperature. However, in general, the energy efficiency of the system increases with an increase in compressor pressure. In spite of the fact that with an increase in pressure, the rate of increase in energy efficiency decreases (the incremental slope of the graph decreases). The increase in exergy efficiency with rising compressor inlet temperatures can be attributed to improve thermodynamic performance. Exergy efficiency depends on factors such as temperature, pressure, and fluid properties. At low pressures (27, 29, and 31 MPa), the continuous rise in exergy efficiency indicates that the system benefits from higher temperatures. However, at higher pressures (33 and 35 MPa), the diminishing efficiency as it approaches the critical temperature suggests a more complex interaction between temperature and pressure. In general, exergy efficiency tends to increase with higher compressor pressures, showcasing the system's ability to convert input energy into useful work more effectively. This cycle exhibits the highest exergy efficiency at an inlet temperature of 30 °C and a pressure of 35 MPa (10.67%). Furthermore, as can be seen, the system reaches an exergy efficiency of 9.86% at a compressor pressure of 27 MPa and an inlet temperature of 25 °C, which is the cycle with the lowest efficiency.

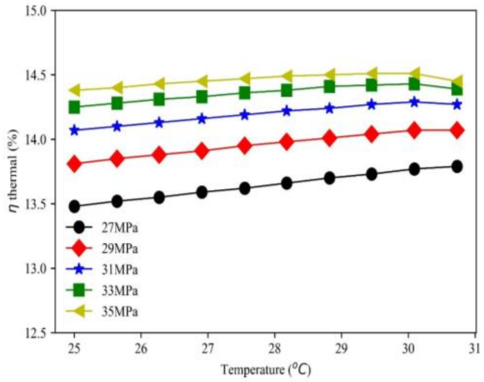
Figure 7 illustrates the effects of an increase in turbine inlet temperature on compressor power consumption, turbine production power, and turbine output power. Increasing the turbine inlet temperature from 180 to 250 °C results in a reduction in compressor power consumption, as shown in Figure 7-a. By raising the compressor pressure, however, will intensify the compressor's work, as expected. In fact, with an increase in the turbine's inlet temperature, the workload required by the compressor decreases. In addition, it should be noted that as the compressor pressure ratio increases, the reduction rate of the turbine's consumed work increases. In Figure 7-b, the effect of increasing turbine inlet temperature on turbine production power is illustrated at various pressures. As with the compressor, an increase in the inlet temperature reduces the turbine's power output. In contrast, increasing the compressor pressure boosts the turbine's production power. The reduction rate of turbine production power increases with an increase in compressor pressure. According to this study, lowering



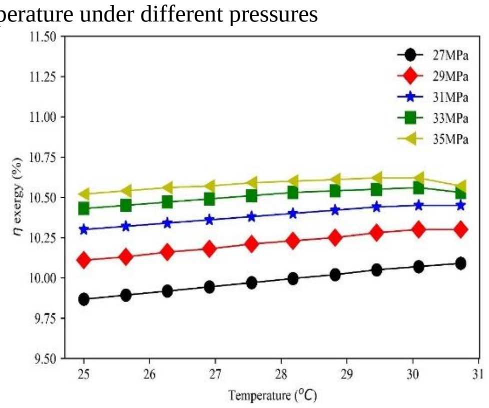
a) Changes in inlet mass rate with increasing inlet temperature under different pressures



b) Changes in output power with increasing inlet temperature under different pressures



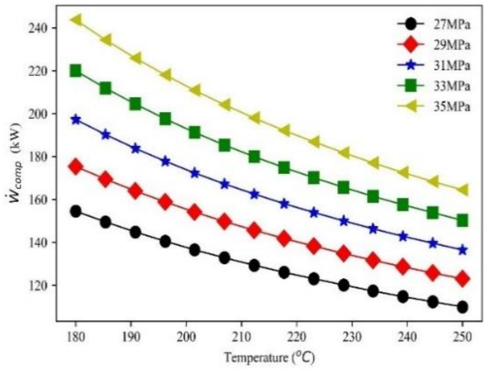
c) Changes in thermal efficiency with increasing inlet temperature under different pressures



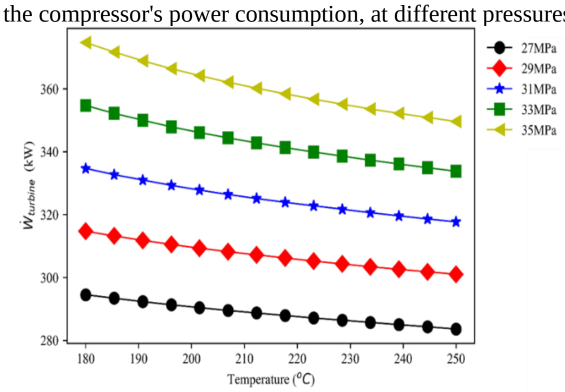
d) Changes in exergy efficiency with increasing inlet temperature under different pressures

Figure 6. Effects of changing the inlet temperature of the compressor on the power and energy efficiency as well as the exergy of the system at different pressure ratios

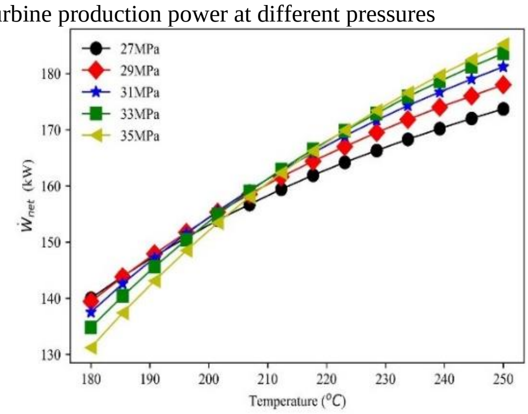
the turbine inlet temperature from 180 to 250 °C reduces turbine production power by 25.1 and 10.9 kW at 35 and 27 MPa pressures, respectively. Figure 7-C illustrates the changes in the output power of the system caused by an increase in the turbine inlet temperature. It can be seen that the output power of the system increases as the inlet temperature of the turbine increases. By increasing the compressor pressure ratio, this trend is intensified. Increasing the turbine temperature results in an increase in the thermal energy of the fluid, reducing energy losses. Considering the total system power is equal to the difference between generated and consumed power, the overall system power increases as a result of this temperature increase. According to the results of this study, increasing the turbine inlet temperature from 180 to 250 °C at pressures of 35 and 27 MPa results in an increase in the output power of the system of 54 and 33.7 kW, respectively.



a) the effect of increasing the turbine inlet temperature on the compressor's power consumption, at different pressures



b) The effect of increasing turbine inlet temperature on turbine production power at different pressures

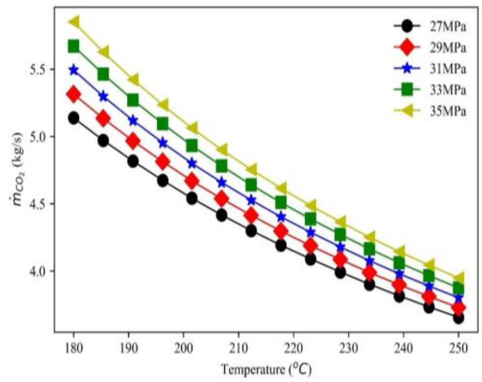


c) The effect of increasing turbine inlet temperature on output power at different pressures

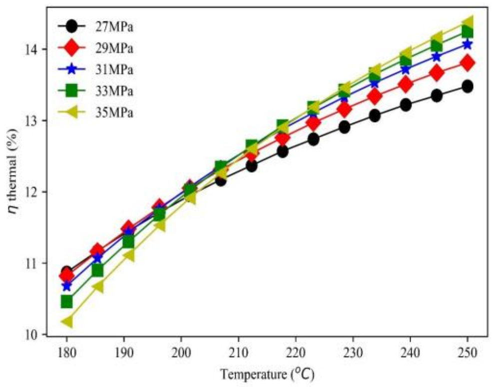
Figure 7. Effects of different turbine inlet temperatures on the production and consumption power as well as the net power of the system at different pressure ratios

Figure 8 illustrates the changes in the working fluid mass rate, energy efficiency, and exergy of the system when the turbine inlet temperature is elevated in the ratio of different compressor pressures. As shown in Figure 8-a, as the inlet temperature of the turbine increases, the mass-fluid working rate of the system decreases. In spite of the fact that the flow rate of the compressor increases with an increase in output pressure, the mass fluid rate decreases with the increase in temperature due to an increase in the input pressure to the compressor. Despite the fact that the flow rate increases with the increase in the output pressure of the compressor, the mass fluid rate decreases with the increase in temperature as a result of the increase in the input pressure of the compressor. An increase in the inlet temperature of the turbine results in an increase in the internal coolant volume of the fluid. This increase in volume leads to a decrease in fluid density. Additionally, the temperature increase causes an expansion of the fluid volume and faster movement of molecular details within the fluid. These combined factors result in a reduction in the mass flow rate of the fluid. Figure 8-b illustrates the changes in output thermal efficiency as the turbine input temperature increases at different compressor pressures. At low pressure ratios, an increase in the inlet temperature of the turbine has a much smaller effect on the output thermal efficiency of the system. As the compressor pressure increases, the effects of increasing the turbine inlet temperature on the output work of the system are intensified. Thus, the output thermal efficiency of the system at a compressor pressure of 27 MPa increases by approximately 2.6% when the inlet temperature of the turbine is raised from 180 to 250 °C. As the turbine inlet temperature increases, the output thermal efficiency of the system increases by 4.2% at a compressor pressure of 35 MPa. Figure 8-c illustrates the changes in the exergy efficiency of the system as the turbine inlet temperature increases in relation to different compressor pressures. The exergy efficiency of the system increases as the inlet temperature of the turbine increases. Additionally, increasing the compressor pressure increases the effect of increasing the inlet temperature of the turbine in addition to increasing the energy efficiency. The change in the system's exergy efficiency with an increase in the turbine inlet temperature occurs due to the increase in the thermal energy of the fluid. With an increase in the turbine's inlet temperature, the thermal energy of the fluid entering the turbine increases. This increase in thermal energy results in an increase in the fluid's volume, leading to an improvement in the system's exergy efficiency. This improves the exergy efficiency of the system at a pressure of 27 MPa by 1.9% with an increase in temperature, while the exergy efficiency of a system at a pressure of 35 MPa increases by 3.12% with an increase in temperature. During this cycle, the majority of the exergy efficiency occurs at the inlet temperature of 250 °C. The compressor pressure is 35 MPa, which is equal to 10.52%.

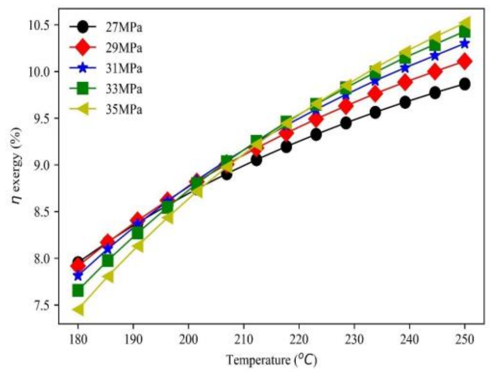
An investigation of the recovery cycles has been conducted in this research to select the system that produces the highest output power with the best system conditions for engine waste heat recovery. This comparative study is shown in Figure 9.



a) Effects of turbine inlet temperature on mass rate at different pressures

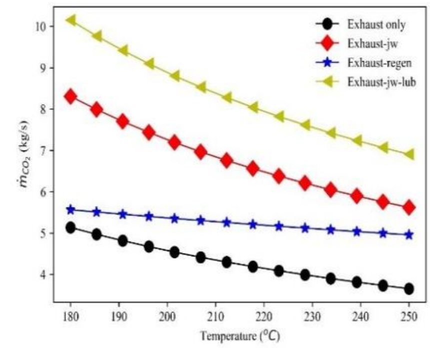


b) Effects of turbine inlet temperature on energy efficiency at different pressures

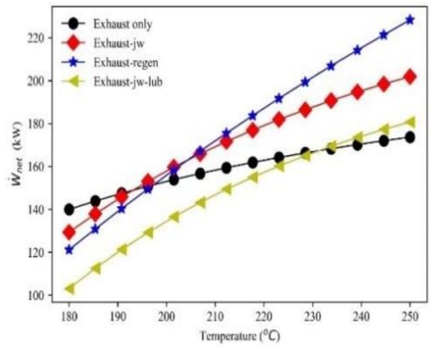


c) Effects of turbine inlet temperature on exergy efficiency at different pressures

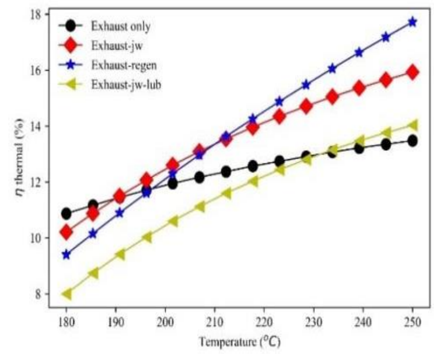
Figure 8. Effects of temperature changes at the inlet to the turbine on the energy efficiency and exergy of the system at different pressure ratios



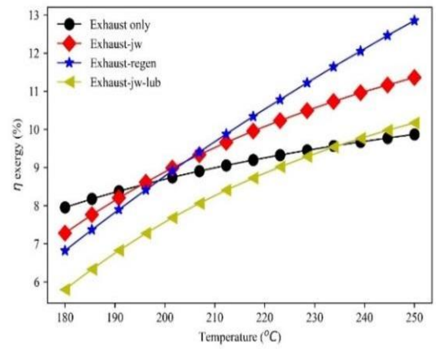
a) Effects of pump inlet temperatures on recovery cycle mass flow rates



b) Effects of pump inlet temperature on recovery cycle output power



c) Effects of pump inlet temperature on recovery cycle energy efficiency



d) Effects of pump inlet temperature on recovery cycle exergy efficiency

Figure 9. An examination of the effects of increasing turbine inlet temperatures on recovery cycle output power and energy efficiency

According to Figure 9-a, the mass flow rate of the recovery cycles changes as the turbine inlet temperature changes. When the inlet temperature of the turbine increases, the mass flow of all cycles decreases. In different cycles, this decreasing trend does not remain constant. Figure 9-a shows that two-stage pressure increases have a higher rate of mass flow rate decrease than other recovery cycles. Furthermore, the cycle with only the exhaust converter has the lowest mass flow rate. According to this paper, a regenerative converter's cycle is least affected by increased turbine inlet temperature. A system with two preheating converters has the highest mass flow reduction rate with increasing turbine inlet temperature.

Increased turbine inlet temperature changes the system's output power, as shown in Figure 9-b. In all cycles, increasing the turbine's inlet temperature raises output power. The cycle with regenerative converters also boosts power most rapidly. Compared with other cycles, the cycle with a regenerative converter reaches its highest power at 250°C. Next, two-stage cycles with heat extraction converters from the engine body and two-stage cycles with preheating converters lead in terms of output power. The cycle with the exhaust tons converter has the lowest power at 250°C.

The energy efficiency and exergy of the recovery systems investigated in this paper (Figure 9-c and d) increase with the turbine inlet temperature. As a result of an increase in inlet temperature to the turbine, the highest system efficiency rate is achieved with the regenerative converter, the two-stage cycle with two preheating converters, and the two-stage recovery cycle with body cooling fluid converters.

Table 3 illustrates the efficiency of energy and exergy of the current system based on the compressor inlet temperature of 25°C and turbine inlet temperature of 35°C, and the pressure ratio of 4.74.

According to the data in Table 3, although the cycle with two heat exchangers and the cycle with the body cooling medium has the highest production power by the turbine, the high power consumption of compressor-2 causes the network of the cycle to decrease in general, resulting in reduced thermal efficiency and exergy. In contrast, despite having a lower production power than the two cycles listed above, the cycle with the regenerative converter consumes less power in the compressor section because the pressure of one stage increases, increasing the system's usable power. It is thus the most efficient thermal and exergy recovery system in this research. Afterward according to energy and exergy analysis of energy and exergy, the cycle with the body cooling converter and the cycle with two preheating converters have the highest efficiencies. Cycles with only exhaust converters have the lowest production and consumption power. It also has the lowest thermal efficiency and exergy among the cycles studied.

TABLE 3. Comparison of the production power, thermal efficiency, and exergy of the recovery systems investigated in this study

Recovery cycle	Compressor1- power consumption (kW)	Compressor-2 power consumption (kW)	Turbine production power (kW)	Output power (kW)	Energy efficiency (%)	Exergy efficiency (%)
Exhaust only	109.9	-	283.6	173.7	13.48	9.86
Exhaust with regenerative converter	207.1	-	435.4	228.3	17.72	12.85
Exhaust with body cooling converter	68.32	227.5	494.7	201.9	15.93	11.36
Exhaust with two preheating converters	51.48	379.3	611.5	180.7	14.03	10.17

6. CONCLUSION

This study aimed to design and model four critical carbon dioxide recovery cycles in EES software to recover wasted thermal energy from marine diesel engines (MAN B&W L35MC6-TII). The following are the major Findings of this research

- All recovery cycles' output power and exergy efficiency decrease with the increasing fluid temperature at the compressor inlet.
- In all recovery cycles, increasing the fluid temperature at the turbine's inlet reduces the compressor's power consumption, increases output power, and increases energy efficiency.
- It is important to note that increasing fluid pressure at the compressor output causes an increase in the compressor's power consumption, but its effect is more significant in increasing the turbine's output power; as a result, when the pressure of the compressor is increased, the output power of the system increases.
- The recovery cycle incorporating the exhaust-regenerator converter has the highest output power and efficiency among the four examined cycles.

- Compared to other recovery cycles, the exhaust converter recovery cycle has the lowest production power, output power, consumption power, energy efficiency, and exergy.
- It has the highest production power with the exhaust converter-2 preheating converter. Still, because the two-stage pressure increases and the compressor-2 consumes more power, energy and exergy efficiency is lower than that of the one-stage exhaust-regenerative converter cycle.
- Increasing the output pressure of the compressor leads to a 5.22% increase in the output power of the system.
- Increasing the fluid inlet temperature to the compressor results in less than 1% increase in the system's output power.
- Increasing the fluid inlet temperature to the turbine from 180 to 250 °C causes a 41% increase in the system's output power.
- By examining the changes in the system's exergy efficiency, it was shown that the highest efficiency is achieved at a temperature of 30 °C and a pressure of 35 MPa (10.67%), while the lowest efficiency belongs to a compressor pressure of 27 MPa and a temperature of 25 °C.
- Increasing the turbine temperature from 180 degrees to 250 °C at pressures of 35 and 27 MPa results in an increase in the system's output power by 54 and 33.7 kW, respectively.

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Persian Abstract

چکیده

از آنجایی که جمعیت و فعالیت های اقتصادی تقاضای انرژی را افزایش داده است، محققان و دانشمندان به بازیابی انرژی های تلف شده در سیستم های مختلف روی آورده اند. این مطالعه با هدف به حداکثر رساندن بازیابی انرژی از موتورهای دیزل دریایی از طریق استفاده از تبادل حرارت در ٤ نوع مختلف از جریان‌های تلف شده (گاز خروجی، مایع خنک‌کننده موتور، مایع خنک‌کننده روغن موتور) انجام شده است. در این مطالعه، چهار چرخه بازیابی کربن دی‌اکسید بحرانی، طراحی و مدل‌سازی شده‌اند تا از انرژی گرمای تلف شده از موتورهای دیزل دریایی مدل (MAN B&W L35MC6-TII) استفاده شود. برای محاسبات ریاضی از نرم‌افزار مهندسی EES برای استفاده شده است. در هر سیکل، تاثیر تغییرات پارامترهای مختلفی همچون فشار خروجی کمپرسور، دمای ورودی کمپرسور و دمای ورودی توربین بر روی توان خروجی، بازده اگزرژی و توان مصرفی کمپرسور مورد تحقیق و بررسی قرار گرفته است. نتایج به دست آمده از این تحقیق نشان می‌دهد که استفاده از چرخه بازیابی حرارت در موتور دیزل، از هدر رفت مقدار زیادی از انرژی در موتور جلوگیری می‌کند. همچنین افزایش دمای سیال در ورودی کمپرسور باعث کاهش توان خروجی و بازده اگزرژی در کلیه سیکل‌های بازیاب می‌شود و افزایش دمای سیال در ورودی توربین باعث کاهش توان مصرفی کمپرسور، افزایش توان خروجی و افزایش بازده اگزرژی در کلیه سیکل‌های بازیاب می‌شود. هنگامی که روغن خنک‌کننده بدنه در چرخه بازیابی استفاده می‌شود، توان خروجی و کارآیی انرژی سیستم افزایش می‌یابد. همچنین چرخه دارای ژنراتور می باشد تا گرمای خروجی کربن دی‌اکسید از توربین بازیابی شود. بر اساس یافته‌های این مطالعه، با وجود بالاترین توان خروجی توربین (٦١١/٥ کیلووات) متعلق به سیستم بازیابی گاز خروجی با دو مبدل است، سیستم بازیابی گاز خروجی با مبدل بازیابی دارای بیشترین توان خروجی قابل استفاده (٢٢٨/٣ کیلووات) است. این سیستم بیشترین انرژی و اگزرژی (١٧/٧٢ و ١٢/٨٥ درصد) را دارد.