Waste Heat Recovery from Marine Diesel Engine Using Organic Rankine Cycle

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Abstract— Many carbon dioxide (co_2) super critical cycles and associated systems were compared in this study. Several carbon dioxide (co_2) super critical systems were proposed, and many elements had an impact on the cycles discussed in this paper. Various working settings for super critical carbon dioxide cycles have been simulated, and diverse working has been examined. Also, a thermodynamics analysis of this system was conducted to determine the ideal operating settings for maximizing operation while increasing overall efficiency. For the analysis, the characteristic equations were solved using the MATLAB program. In addition, a performance analysis was carried out to evaluate each system's applicability and power output under various engine loads. The suggested designs also included an evaluation of their economic and environmental implications.

Index Terms—carbon dioxide, energy management, waste heat recovery, organic rankine cycle

I. INTRODUCTION

One of the biggest challenges affecting the weather from all angles is global warming [1]. There are numerous ways to switch from traditional energy sources to green energy sources, but the proposals for new resources are urgently needed in light of the energy and environmental issues. Modern organic rankine cycles use natural working fluid, as carbon dioxide co_2 which has significant demands for developing new technology to support its use based on using in safe and secure methods in order to reduce global warming and the greenhouse impact [2, 3].

This study aimed at improving the design efficiency of marine diesel engines as a mean of decreasing fuel consumption and reducing carbon dioxide emission. The scope of reducing carbon dioxide emission by means of controlling the generation of emissions inside the engine cylinder, after-treatment of the exhaust by removing the emission and higher maximum combustion pressure. A noticeable improvement reached that increasing engine efficiency, as current engines can achieve a thermal efficiency of around 50%, is a significant strategy to reduce emissions. Hence, using the approximately 50% of waste heat produced by the waste heat for additional power generation is the best method currently available for lowering emissions [4].

The waste heat energy available from a diesel engine mainly resides in the exhaust gas which is primarily used in the

Turbocharger for increasing the pressure of the scavenge air in order to achieve proper scavenging and improve the Volumetric efficiency of the engine. Yet, there is still roughly 25.4% of the fuel's energy that can be used. As a result, waste heat is primarily generated by exhaust gas [4].

Energy usage is rising worldwide as a result of the world's rapid development. The International Energy Agency (IEA) conducted research in 2002 that indicated that the world's energy requirements would rise by 11%. (IEA report, 2004).

Many environmental issues, including air pollution, global warming, and ozone depletion, have been brought on by the massive usage of fossil fuels.

In addition to, coupled with consistently expanding energy and, Blackouts and electricity shortages have increased, and more continually all over the world. These reasons have led to an increase in interest in generating electricity utilizing low-grade heat sources to increase energy usage efficiency.

Most low-grade heat sources don't maintain a steady temperature throughout the process of utilizing their energy; instead, they exhibit a declining temperature.

This occurred for two reasons: either the heat source's temperature was insufficiently high or its mass flow was constrained. As a result, the temperature of the heat source flow began to fall as soon as the energy in it was released.

Several studies have adopted the aim of improving the power generation efficiency of marine diesel engines owing to its high impact on the environment considering the significant role of maritime transport. Marine transport contributes significantly to global trade; it conveys nearly as 90% of traded goods by volume which verifies its impact on the global economic development. The demand for maritime transport has risen in recent years, with the amount of payload transported rising from 2.6 billion tones in 1970 to 9.5 billion tones in 2013[5, 6].

Consequently, the need to enhance the power generation efficiency of diesel engines resulted from the recognition of the current climate change problem and the procedures taken

to decrease the greenhouse gas emissions that accompany the use of carbon dioxide as a primary source of energy. During the period of 2007-2012 the annual average fuel consumption from marine diesel engines ranged approximately from 247 million ton to 325 million ton. [5].

The International Maritime Organization (IMO) reported that the total emission of carbon dioxide from maritime

Shipping in 2012 was estimated at 938 million tons and it is expected to rise up to 250% by 2050. On the other hand, design efficiency of new ships has dropped by 10% since 1990, due to the less hydrodynamic hull design as a need to

Nomenclature

ρair	Density for air, kg/m3
cp_{gas}	The specific heat constants for gas, kJ/kg $^{\circ}c$
T_{air}	Air temperature, $^{\circ}c$
D	Diameter, m
m_{gas}	Mass flow rate for gas, Kg/s
m_{fuel}	Mass flow rate for fuel, Kg/s
m_{air}	Mass flow rate for air, Kg/s
m_{c}	Mass flow rate for carbon dioxide, Kg/s
h	Specific enthalpy, KJ/kg
HV	Heating value, KJ/kg
p	Pressure, bar
Q	Heat transfer rate, KW
R	Specific gas constant, kJ/kg $^{\circ}c$.
r	Radios, mm
t_{gas1}	Inlet Gas temperature, $^{\circ}c$
t_{gas2}	Outlet Gas temperature, $^{\circ}c$
t_{1}	Inlet carbon dioxide temperature to pump, $^{\circ}c$
t_{2}	Inlet carbon dioxide temperature to evaporator, $^{\circ}c$
t_{3}	Inlet carbon dioxide temperature to turbine, $^{\circ}c$
t_{4}	Inlet carbon dioxide temperature to condenser, $^{\circ}c$
t_{sw1}	Outlet Sea water temperature, $^{\circ}c$
t_{sw2}	Inlet Sea water temperature, $^{\circ}c$
cp_{sw}	Specific heat capacity for sea water, kJ/kg $^{\circ}c$.
s	Gas entropy, $kJ \cdot c^{-1}$
μis	Isentropic effiency
w_{t}	Turbine work, Kw

Acronyms

WHR	Waste heat recovery
ICE	internal combustion engine
IMO	the International Maritime Organization
ORC	organic Rankine cycle
SRC	Steam Rankine cycle
HFO	Heavy Fuel Oils
co_{2}	carbon dioxide
IEA	the International Energy Agency

maximize cargo capacity [7].

As renewable energy has been advocated for decades and longer, a number of professionals have been trying to develop and improve a new combined power/heat thermo-fluid cycle.

The Supercritical Rankine cycle system, which uses heat energy as the energy source and as the working fluid, was initially conceived and built in 2004 due to the analytical finding that the system efficiency would be higher than that of a regular steam cycle. Benefits from the system analysis also include those relating to the economy and the environment [8].

Therefore, has a low critical point, with a critical temperature and pressure of 31.1 °C and 7.38 MPa, respectively [9].

Because to its low critical point, the thermo-physical characteristics show that it changed its phase easily to a supercritical phase at moderate working conditions. When utilized as a working fluid in the organic rankine cycle at temperatures between 30 and 200 °C, it offers a tremendous potential for high efficiency.

Literally, it is simple to deliver to the user and is utilized as the working fluid for the organic rankine cycle to produce energy as output power and/or to supply heat energy for domestic consumption. The use of renewable energies for thermal energy supplies and electric power generation is the most effective way to preserve the environment and promote sustainable development.

Hence, because of its poor thermal efficiency and high volume flows, the conventional the performance of the steam Rankine cycle is unsatisfactory.

Additionally the exhaust gases will have high moisture contents after expanding if the low-grade heat source is insufficient to appropriately superheat the working fluid to a specific high temperature. The turbine blades will erode as a result, and they might possibly be destroyed. [10] The so-called organic Rankine Cycle (ORC) was created as a result of their capacity to utilize the energy in low-grade waste heat.

I. ORGANIC RANKINE CYCLE

The organic Rankine cycle (ORC), a non-superheating thermodynamic cycle, uses an organic working fluid like carbon dioxides to generate energy (electricity).

The expanding vapor from the saturated vaporization of the working fluid is employed to run an extension device [11].

Because the working fluid does not need to be superheated, ORC has the potential to increase cycle efficiency. for many organic compounds in order to prevent moisture erosion at the turbine output [11].

The temperature profile of carbon dioxide in the supercritical zone can match the temperature glide of the heat source better than the other.

With the counter flow heat exchanger, the so-called "pinching" that usually happens for other working fluids will not be present. The cycle comprises of four primary components namely, a steam generator ‘boiler’, condenser, expansion device ‘turbine’, and feed pump. The layout of these components is shown in **Fig.1**. [12]

The working fluid is first pumped to the evaporator, where it is evaporated before entering the turbine using the waste heat from the heating fluid. Heat energy is converted to kinetic energy, which drives the turbine's rotation and produces work output that is then sent to an output shaft or an electric generator.

Recently, the system functions on a closed cycle where the working fluid is condensed in the condenser before pumping to the boiler [12].

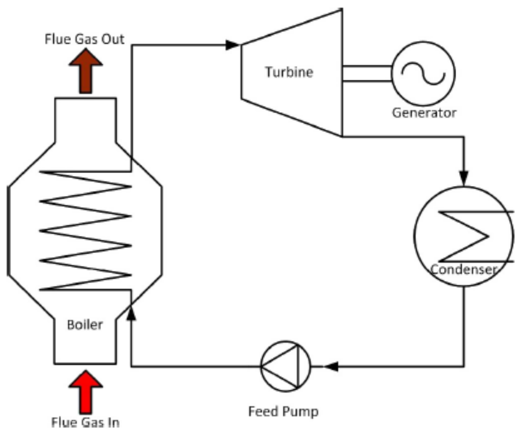

Fig. 1 simple rankine cycle

The ORC is a modified cycle based on the SRC where instead of water, the working fluid used is an organic fluid for example, carbon dioxide, and refrigerants. The simple cycle consists of the main components of the SRC, where different configurations can be made to achieve optimal efficiency and heat utilization according to the type of working fluid used [13].

A large variety of working fluids are available, and as a result, the system's performance can vary.

They can be classed as wet, dry, and isentropic fluids according to the slope of their saturated vapor line on the temperature-entropy diagram as depicted in **Fig. 2**.

The choice of working fluid is mostly determined on the temperature range of waste heat, where wet fluids are not advised for low and medium temperatures due to the lack of superheating capabilities and the potential for turbine erosion. The performance of the cycle depends mostly on the working fluid selected which lead to the vast study of working fluids selection for diverse systems in prior literature.

The ORC is considered as a promising system for recovering waste heat and improving the ship efficiency, while considering the chemical stability, safe handling, Global Warming Potential, and storage of the working fluid [13].

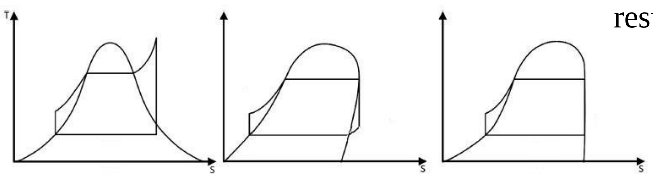

Fig. 2 Types of working fluids: a) wet fluid b) dry fluid c) isentropic fluid.

The energy conversion process of an ORC system is quite similar to a typical steam cycle, where the thermal energy is transferred to kinetic energy that can subsequently be used to create electricity. The term optimization can be defined from a practical standpoint as to get the “best solution” to a particular process within limitations [14].

This is a significant part of chemical engineering and one of the major quantitative techniques in decision making. The competitive advantage of being able to run process systems at their best efficiency is crucial to the financial viability of many businesses across a range of industries, which is what motivates the development and implementation of optimization strategies.

Process optimization requires the work of developing a quantitative scalar performance index, frequently referred to as the objective function, in order to quantify the "optimal solution." The operational costs, the production yield, or the system efficiency are simply a few instances of what this can be. The objective function is increased or decreased by varying the values of the decision variables, which may reflect the physical size of the equipment, plant parameters, or the thermodynamic properties of the working fluid, such as temperature, pressure, and enthalpy. These decision variables frequently need to be changed in order to be compliant with system and process constraints, which are frequently expressed as the system's physical limitations, environmental laws, financial restrictions, and other restrictions.

II. DIESEL ENGINE SPECIFICATIONS

The 6RT-flex58T-E diesel engine of HYUNDAI-WARSTILA, a power plant with a maximum output of 13.94 MW was used for the duration of the experiment. a constant pressure two-stroke marine diesel engine turbo charging powers the vehicle. The manufacturer conducted an experimental official shop test on the engine as a method for quality control before installation onboard ship, and these results were used to determine the engine's operating specifications.

The continuous service rating of the engine, which is shown in **Table I**, will serve as the foundation for the systems given in this study's design. The engine manufacturer's maximum output guarantee for continuous daily operation is known as the continuous service rating [15].

The temperature and mass flow rate of the charge air and exhaust gas at various operating loads are provided by the shop test (sea trial), as shown in **Table II.**

Table. I Operational parameters of diesel engines at the maximum service rating.

Output power		12456	KW
Charge air (exit of the compressor)	Mass flow rate	28.2069	kg/s
	Temperature	213	$^\circ c$
Exhaust gas (exit of the turbine)	Mass flow rate	28.8179	kg/s
	Temperature	251	$^\circ c$
Fuel	Mass flow rate	0.611	kg/s
	Heating value [40]	38759	kJ/kg

Table. II a distribution of the mass flow rates of the charge air and exhaust gases under different operating loads

Load (%)	25	50	75	90
mass flow rate for air (kg/s)	6.8829	14.7927	23.546	28.2096
mass flow rate for fuel (kg/s)	.179838	.34431	.506116	.61107
mass flow rate for gas (kg/s)	7.0627	15.137	24.052	28.817

III. OPERATING CONDITIONS

The steady state physical and thermodynamic equations, which are based on the first law of thermodynamics, were applied in this investigation.

When the ship is at sea, the steady state equations are appropriate, but they were disregarded when there was rough weather and sea [16].

Throughout the investigation, the following presumptions are made:

Assumed stable state for the system.

- No pressure decrease on the heating fluid side, which will imply that the WHR unit has no impact on the engine's performance as It is assumed that the system is in a steady condition.
- The specific heat capacity of air is assumed to be cp=1.005 kJ/kg$^\circ c$.
- The specific heat capacity of sea water is assumed to be cp=3.85 kJ/kg$^\circ c$.
- The specific heat capacity of the exhaust gas produced by burning

HFO in the engine is assumed to be cp=1.1 kJ/kg $^{\circ}c$.

- The specific heat capacity carbon dioxide is assumed to be cp=.846 kJ/kg $^{\circ}c$.
- The isentropic efficiency of the turbine is assumed to be 95%.

IV. SYSTEM ILLUSTRATION AND CORRESPONDING CYCLE DESCRIPTION

1-The simple rankine cycle using carbon dioxide

A. *The System description*

The first model analyses a standard WHRS working on the simple Rankine cycle, depending mostly on the exhaust gases of the diesel engine for creating superheated steam.

Cycles are made up of four processes: compression (1-2), isobaric heat supply (2-3), expansion (3-4), and isobaric heat rejection (4-1).

The carbon dioxide simple rankine system schematic is shown in Figure 3.

The subsequent design is based on the engine's continuous serving rating at 90% load.

The heat available in the exhaust depends mostly on the entrance temperature of the exhaust gases, and the output temperature of the exhaust gases. **Table III** displays the specified carbon dioxide parameter

Table. III Carbon dioxide simple rankine cycle operating conditions

item		value	unit
The inlet exhaust temperature	t_{gas1}	251	$^{\circ}c$
The outlet exhaust temperature	t_{gas2}	150	$^{\circ}c$
Exhaust gas mass flow rate	m_{gas}	28.81797	kg/s
Specific heat capacity of the exhaust gas	cp_{gas}	1.1	kJ/kg $^{\circ}c$
Inlet temperature under supercritical	t_1	29	$^{\circ}c$
Inlet temperature above supercritical	t_2	40	$^{\circ}c$

The t_{gas2} is set at a minimum value of 150 $^{\circ}c$, to avoid the dew point of the exhaust gas that produced sulfuric acid. This will prevent the condensation of sulfur dioxide and the formation sulfuric acid which can damage the boiler and shorten its lifetime [16].

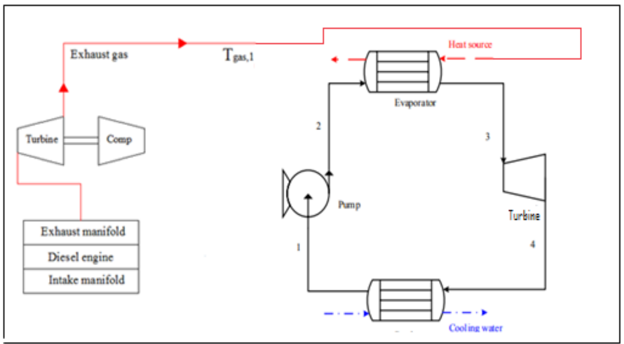

Fig. 3 Schematic diagram for the carbon dioxide simple rankine system

The temperature of evaporator inlet t_2 and enthalpy h_2 is equal to the saturated liquid temperature and enthalpy at the condenser pressure, where the work of the feed pump is neglected.

The turbine efficiency is isentropic which is assumed to be 95%.

B. Equations

1. Energy equations

The equations used for developing the first thermodynamic law are the foundation of the WHR systems, which are energy balance equations. for energy conversion. The energy balance equation for the simple rankine cycle using carbon dioxide can be calculated from equation (1).

$$Q_{overall} = m_{gas}cp_{gas}(t_{gas1} - t_{gas2}) = m_c(h_3 - h_2)$$

(1)

The mass flow rate of carbon dioxide m_c can be calculated from equation (2) which is derived from the overall energy balance of the Evaporator equation (1).

$$m_c = \frac{m_{gas}cp_{gas}(t_{gas1} - t_{gas2})}{(h_3 - h_2)}$$

(2)

The temperature of the carbon dioxide at the exit of the turbine T_4 is calculated from equation (3).the isentropic efficiency for the turbine assumed to be 95%.

$$t_4 = \mu_{is} * (t_3 - t_4^{/}) + t_3$$

(3)

Since t_4 calculated from equation (3) the turbine work may be calculated from equation (4)

$$w_t = m_c(h_3 - h_4)$$

(4)

The energy from condenser can be calculated from equation (5).

$$Q_{condenser} = m_{sw}cp_{sw}(t_{sw2} - t_{sw1}) = m_c(h_4 - h_1)$$

(5)

The outlet sea water temperature calculated from equation (6) the inlet sea water temperature will be assumed to be 20℃

$$t_{sw2} = \frac{m_c(h_4 - h_1)}{m_{sw}cp_{sw}} + t_{sw1}$$

(6)

The total efficiency for the organic rankine cycle will be calculated from equation (7).

$$\mu_{ov} = \frac{w_{out}}{w_{in}} = \frac{Q_{condenser}}{Q_{overall}} * 100\%$$

(7)

From dividing the output work and the input work.

2. Exergy equations

The equations used in the systems are exergy equations.

The exergy of carbon dioxide for the pump can be calculated from equation (8)

$$I_{pump} = t_0(s_2 - s_1)$$

(8)

The exergy of carbon dioxide for the evaporator can be calculated from equation (9)

$$I_{evap} = t_0(s_3 - s_2) + (h_3 - h_2) * \frac{t_0}{t_r}$$

(9) The exergy of carbon dioxide for the turbine can be calculated from equation (10)

$$I_{turbine} = t_0(s_3 - s_4)$$

(10)

The exergy of carbon dioxide for the condenser can be calculated from equation (11)

$$I_{cond} = h_4 - h_1 - t_0(s_4 - s_1)$$

(11)

The total exergy of carbon dioxide for the turbine can be calculated from equation (12) by adding equation (8), (9), (10), and (11)

$$I_{total} = I_{pump} + I_{evap} + I_{turbine} + I_{cond}$$

(12)

The chart design T-S digaram for simple rankine cycle presented in **fig.4**.

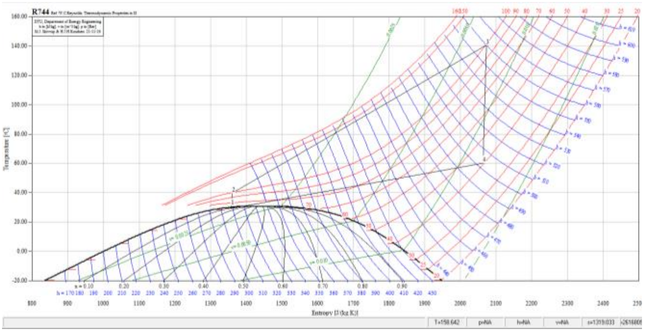

Fig. 4 T-S diagram for simple rankine cycle using carbon dioxide chart design

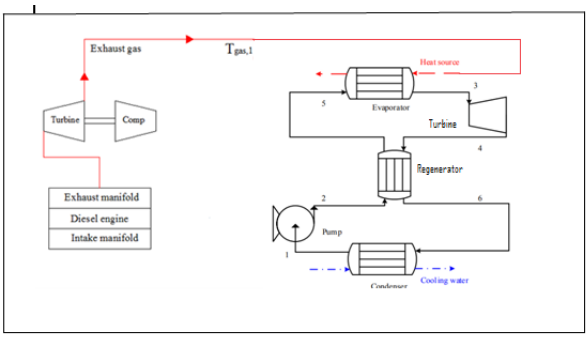

Fig. 5 Schematic diagram for the Regenerate rankine cycle using carbon dioxide

2-Regenerate rankine cycle using carbon dioxide

A. *System description*

Whereas earlier literature only focused on using the exhaust gas as a source of waste heat, this model will attempt to incorporate the waste heat present in the exhaust gas into a single pressure Rankine cycle.

The design includes a recover and an evaporator.

Five basic parts make up the regenerate rankine cycle system: a condenser (gas cooler), pump, regenerator, turbine, and gas heater. With this technique, the carbon dioxide is pushed to a supercritical pressure before being heated in the petrol heater. The heated supercritical carbon dioxide will expand inside the turbine.

Vapor condenses in a condenser. released from the turbine outlet will then be cooled (gas cooler).

As shown in fig. 5, to improve system performance, the internal heat exchanger (regenerator) is added to the basic system.

B. *Equations*

1. Energy equations

The temperature distribution and the output power of the system will be computed at the selected range of pressure.

The optimum working pressure will be set based on the highest output power, a sufficient dryness fraction, and a temperature distribution which satisfies the set assumptions.

Energy balance equations based on the first thermodynamic law for energy conversion are used to construct the WHR systems. The energy balance equation for the simple rankine cycle using carbon dioxide can be calculated from equation (13).

$$Q_{overall} = m_{gas} cp_{gas} (t_{gas1} - t_{gas2}) = m_c (h_3 - h_5)$$

(13)

The mass flow rate of carbon dioxide mc can be calculated from equation (13) which is derived from the overall energy balance of the Evaporator equation (14).

$$m_c = \frac{m_{gas} cp_{gas} (t_{gas1} - t_{gas2})}{(h_3 - h_5)}$$

(14)

The temperature of the carbon dioxide at the exit of the turbine T_4 is calculated from equation (15).the isentropic efficiency for the turbine assumed to be 95%.

$$t_4 = \mu_{is} * (t_3 - t_4^{/}) + t_3$$

(15)

Due to t_4 calculated from equation (15) the turbine work can be calculated from equation (16)

$$w_t = m_c(h_3 - h_4)$$

(16)

The energy from condenser can be calculated from equation (17)

$$Q_{condenser} = m_{sw}cp_{sw}(t_{sw2} - t_{sw1}) = m_c(h_6 - h_1)$$

(17)

The outlet sea water temperature calculated from equation (18) the inlet sea water temperature will be assumed to be 20℃

$$t_{sw2} = \frac{m_c(h_6 - h_1)}{m_{sw}cp_{sw}} + t_{sw1}$$

(18)

The total efficiency for the organic rankine cycle will be calculated from equation (19).

$$\mu_{ov} = \frac{w_{out}}{w_{in}} = \frac{Q_{condenser}}{Q_{overall}} * 100\%$$

(19)

From dividing the output work and the input work.

The heat from regenerator can be calculated from equation (20)

$$Q_r = m_c(h_5 - h_2) = m_c(h_4 - h_6)$$

(20)

2. Exergy equations

The equations used in the systems are exergy equations [17].

The exergy of carbon dioxide for the pump can be calculated from equation (21)

$$I_{pump} = t_0(s_2 - s_1)$$

(21)

The exergy of carbon dioxide for the evaporator can be calculated from equation (22)

$$I_{evap} = t_0(s_3 - s_5) + (h_3 - h_5) * \frac{t_0}{t_r}$$

(22) The exergy of carbon dioxide for the turbine can be calculated from equation (23)

$$I_{turbine} = t_0(s_3 - s_4)$$

(23)

The exergy of carbon dioxide for the condenser can be calculated from equation (24)

$$I_{cond} = h_6 - h_1 - t_0(s_6 - s_1)$$

(24)

The exergy of carbon dioxide for the regenerator can be calculated from equation (25)

$$I_r = t_0(s_5 - s_2) - t_0(s_4 - s_1)$$

(25)

The total exergy of carbon dioxide for the turbine can be calculated from equation (26) by adding equation (21), (22), (23), (24), and (25)

$$I_{total} = I_{pump} + I_{evap} + I_{turbine} + I_{cond}$$

(26)

The chart design T-S diagram for Regenerate rankine cycle using carbon dioxid chart design presented in **fig.6**.

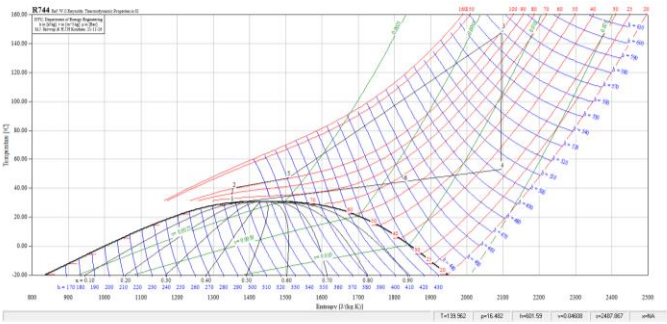

Fig. 6 T-S diagram for Regenerate rankine cycle using carbon dioxide chart design

V. RESULTS

In this chapter, the results of the MATLAB program developed to solve the thermodynamic characteristic equations representing the proposed systems are displayed. The program was built on the base of iterative procedure to exhibit the effect of varying operating pressure for the proposed systems on performance parameters including: the output power, temperature distribution. The optimum operating pressure will be selected for each system based on the maximum power output and a suitable dryness fraction for the exhaust steam.

A. The simple rankine cycle using carbon dioxide

The system proposed in this case offered a conventional waste heat recovery steam Generator relying on the exhaust gas as a heat source. The purpose of this system is to compare the method used in previous literature but with the preferences of the diesel engine used in this study, to further asses the improvement developed in this study by comparing its performance to that of the configurations proposed.

In **fig.7** due to increase the load the overall heat is going to increase.

In **fig. 8** due to increasing the load the overall heat increase which is affect on the temperature for the carbon dioxide so the mass flow rate for the carbon dioxide is going to increase.

In **fig. 9** the turbine work increase because of the assumption for the efficiency is 95%.

In **fig. 10** by increasing the load the heat from condenser is going to increase to cool the sea water which is used to cool the carbon dioxide.

In **fig. 11** the heat of the pump is going to increase by increasing the load because of the fresh of carbon dioxide which is entered the pump.

In **fig. 12** the total efficiency is high because of the heat from the condenser and the evaporator

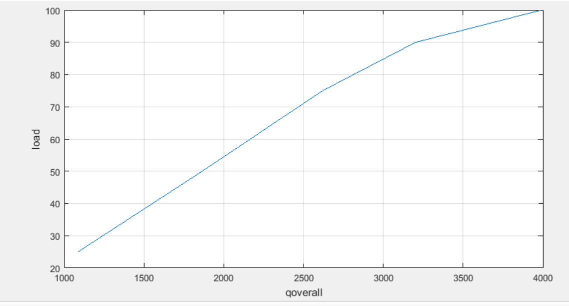

Fig. 7 refer to relation between load and overall heat

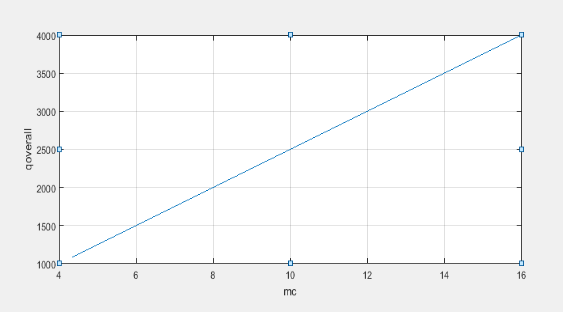

Fig. 8 refer to relation between mass flow rate for organic carbon dioxide and load.

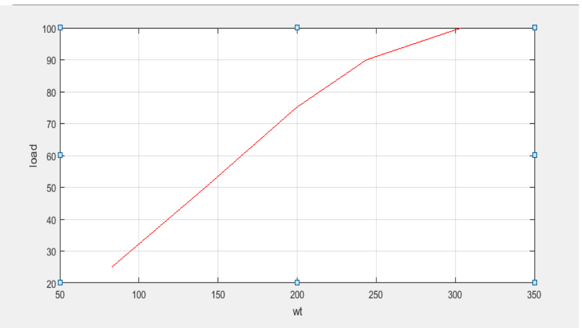

Fig. 9 refer to relation between work for turbine and the load

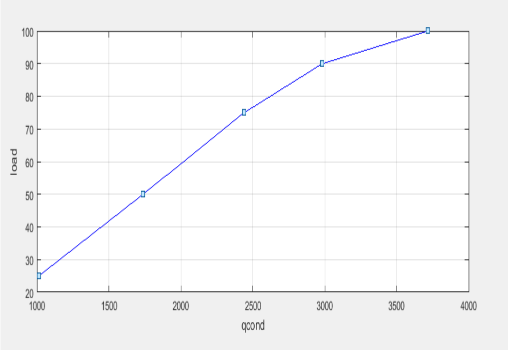

Fig. 10 refer to relation between heat of condenser for the cycle and the load.

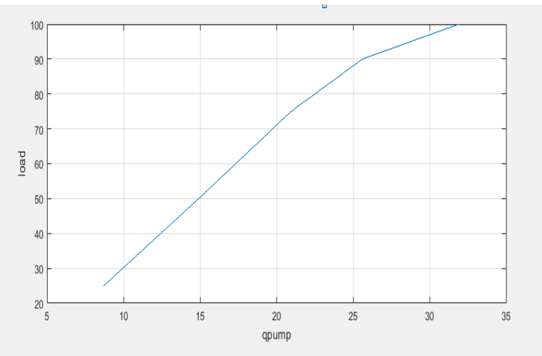

Fig. 11refer to relation between heat of pump for the cycle and the load.

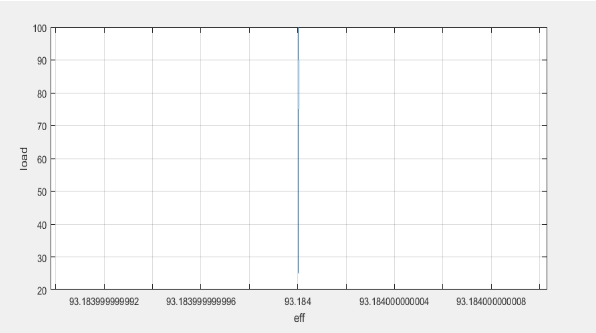

Fig. 12 refers to relation between efficiency for the cycle and the load

Eff
93.18%

B. Regenerate rankine cycle using carbon dioxide

The system proposed in this case offered a conventional waste heat recovery steam regenerator relying on the exhaust gas as a heat source. The purpose of this system is to compare the method used in previous literature but with the preferences of the diesel engine used in this study, to further asses the improvement developed in this study by comparing its performance to that of the configurations.

In **Fig.13** due to increasing the load the overall heat increase which is affect on the temperature for the carbon dioxide so the mass flow rate for the carbon dioxide is going to increase.

In **Fig. 14** the heat of the pump is going to increase by increasing the load because of the fresh of carbon dioxide which is entered the pump.

In **Fig.15** the turbine work increase because of the assumption for the efficiency is 95%.

In **Fig.16** by increasing the load the heat from condenser is going to increase to cool the sea water which is used to cool the carbon dioxide.

In **Fig. 17** due to increase the load the overall heat is going to increase.

In **Fig. 18** the total efficiency is high because of the heat from the condenser and the evaporator.

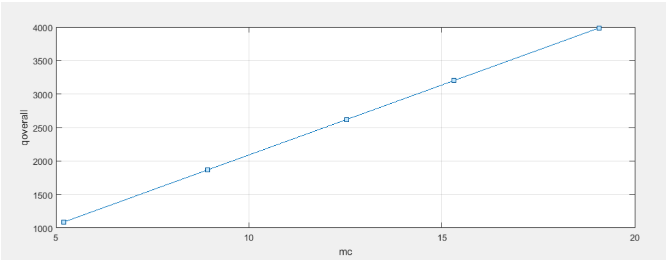

Fig. 13 refer to relation between load and mass flow rate.

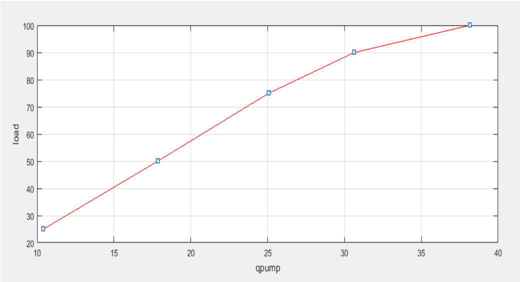

Fig. 14 refer to elation between pump heat and load.

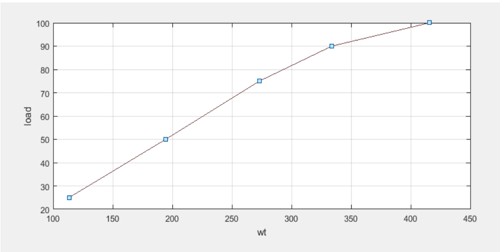

Fig. 15 refer to relation between work from turbine and the load.

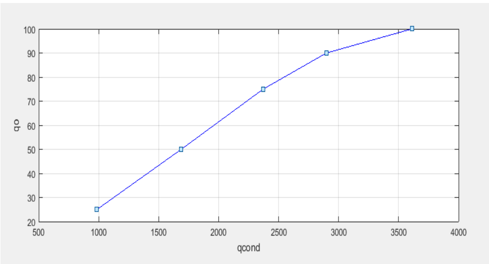

Fig. 16 refer to relation between heat of condenser for the cycle and the load. .

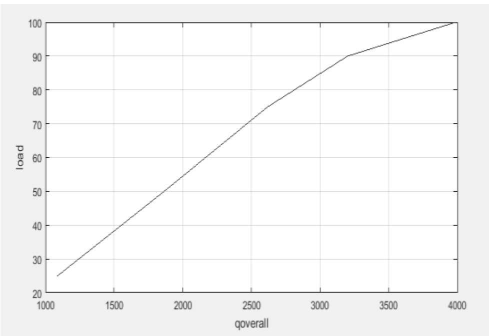

Fig. 17 refer to relation between overall heat for the cycle and the load.

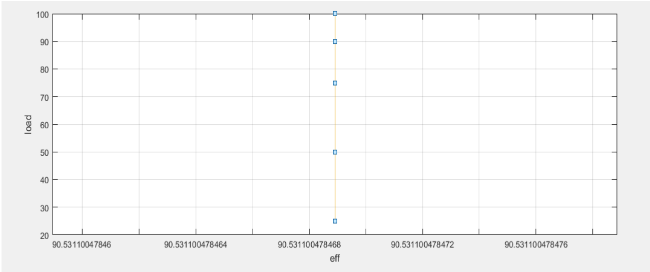

Fig. 18 refer to relation between efficiency for the cycle and the load

Eff
90.5%

Conclusion

This final report outlines the key outcomes and accomplishments from the ORC's several cycles and represents the key activities that took place over those cycles.

The project's objective was to assess the economic and technological viability of using organic Rankine cycle units to recover waste heat from diesel engines. The study involved both experimental testing and the creation of simulation models. The execution of this project led to the following key conclusions:

The unit has successfully passed its testing, which included the installation and logging of additional measurement devices and the examination of part-load operation.

The test results were used to successfully certify the numerical models of ORC units that were subsequently developed. used to calculate the likelihood of future ORC unit deployments.

Based on the operational experience gathered during the testing, it was advised that potential future large-scale ORC unit installations take into account automatic HT management, dynamic instabilities during low power operation, evaporator insulation, and back-using for condenser cleaning.

The size of the heat exchanger, particularly the condenser, poses a challenge in designing ORC units for low temperature heat recovery. Two other options were considered in addition to dividing the condenser into two smaller, parallel-operating condensers.

The size of the condenser might have been greatly reduced by the second strategy, which required increasing the condensing pressure but it was not advised because it would have decreased the net power output of the ORC unit.

Considering the first model's high efficiency, it is recommended.

Recommendations for future work

a) Implementing the scavenge air as a waste heat source for operating an organic Rankine cycle, while considering different cycle configurations.
b) Assessing the possibility of incorporating an absorption cycle which uses the scavenge air as a heat source for achieving tri-generation

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