Waste Heat Recovery from A Large Marine Diesel Engine Through a CO_2-organic Cascading Cycle

The thesis is submitted in partial fulfilment of the requirements for the degree of

Master in Mechanical Engineering

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May 2019

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Acknowledgement

I would like to offer my sincere thanks to all who provided me this opportunity and granted me the capability to proceed successfully with my thesis. This thesis appears in its current form due to the assistance and guidance of several people. I am grateful to my guide Prof Sudipta De and Mr. Subho Mondal for the help and guidance to complete my Masters thesis. I want to convey special thanks to Dr. Subho Mondal who was instrumental in helping me to design the problem statement. I am deeply indebted to the esteemed faculty of Mechanical Engineering Department of Jadavpur University who have acquainted me with the various theoretical concepts that were extensively used while working on the aforementioned problem statement. I am grateful to all the professors and present HOD Dr Goutam Mojumdar of Mechanical engineering Department for their words of appreciation and faith they shown in me

Date 29th May **Soumitra Dutta**

Abstract

With increasing oil price and emission from marine diesel engine, it is necessary to improve utilization efficiency of a marine diesel engine. In recent time waste heat recovery is emerging as one of the promising technology for improving energy efficiency. In present study a cascading between a CO_2 power cycle and an ORC is considered for the utilization of waste heat released by the marine diesel engine. R290, R600 and R1233zd (e) are considered as the working fluids of the bottoming cycle for their lower GWP. The analysis revealed that cascading cycle with all three selected bottoming cycle working fluid can deliver significantly higher power compared to that of a regenerative T-CO_2 power cycle if operating pressure in the flue gas-CO_2 heat recovery unit (FGCHRU) is less than 14MPa. Corresponding bare module costs per unit power output of cascading cycle are also significantly small. The cascading cycle delivers highest power output and lowest BMC per kW by using R1233zd (e) as the working fluid of the bottoming cycle.

Key words

Marine diesel engine, waste heat, CO_2 power cycle; CO_2-organic cascading; bare module cost;

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Nomenclature

Symbols

B_1, B_2	bare module factor of equipment
C_1, C_2, C_3	pressure factor of equipment
C_P^0	purchased equipment cost, $
c_p	specific heat, kJ kg^{-1} K^{-1}
C_{BM}	bare module cost, $
F_P	pressure factor
F_M	material factor
K_1, K_2, K_3	coefficients of equipment cost, $
g	acceleration due to gravity, m s^{-2}
h	heat-transfer coefficient, kWm^{-2} K^{-1}, enthalpy, kJ kg^{-1}
k	thermal conductivity, kW m^{-1} K^{-1}
M	molecular weight of working fluid, g $mole^{-1}$
m	mass flow rate, kg s^{-1}
Nu	Nusselt number
Pr	Prandtl number
P	pressure, MPa
Q	heat transfer rate, kW
Re	Reynolds number
s	entropy, kJ $^{-1}$ K^{-1}
T	temperature, ^{o}C
$T_{g,I}$	exhaust gas inlet temperature, ^{o}C
$T_{g,o}$	exhaust gas outlet temperature, ^{o}C
ΔT	mean logarithmic mean temperature difference, ^{o}C
U	overall heat-transfer coefficient of the heat exchanger, kW m^{-2} K^{-1}
$W_{t,tur}$	power output of the turbine of topping cycle, kW
$W_{b,tur}$	power output of the turbine of bottoming cycle, kW
$W_{t,pump}$	power consumed by the pump of topping cycle, kW
$W_{b,pump}$	power consumed by the pump of topping cycle, kW
$W_{t,NET}$	power output from topping cycle, MW

$W_{b,NET}$ power output from bottoming cycle, MW

$W_{CASCADE}$ total power output from the CO_2-Organic Fluid Cascade system, MW

X equipment type

Y the capacity or size parameter of equipment, kW or m^2

Greek symbols

μ dynamic viscosity, Pa-s

ρ density

Subscripts

b Bottoming cycle

con condensation, condenser

cw cooling water

cyl cylinder cooling water

exh exhaust gas

exp expander

i inside, inlet

j section

o outside

r organic working fluid for bottoming cycle

sca scavenging air cooling water

t Topping cycle

tur turbine

Acronyms

BMC Bare Module Cost

COFHRU CO_2-organic fluid heat recovery unit

CEPCI chemical engineering plant cost index

FGCHRU Flue gas CO_2 Heat Recovery Unit

GWP Global warming potential

ODP Ozone depletion potential

ORC Organic Rankine cycle

TRC Transcritical Rankine cycle

1. Introduction

It is necessary to reduce fossil fuel consumption and corresponding CO_2 emission to ensure future sustainable development. As appreciable amount of fossil fuel is consumed by marine transport system, incorporation of waste heat recovery in marine diesel engine would lead to reduced fossil fuel consumption.

1.1 Motivation behind Waste Heat Recovery in Marine Diesel Engine

The majority of goods transported world-wide are carried by sea. Although shipping is environmentally friendly and cost-effective compared to other means of transport, the shipping industry is responsible for large amounts of emissions of CO_2, SO_x (sulfur oxides) and NOx (nitrogen oxides). Such emissions result from the combustion of fossil fuels like heavy fuel oil (HFO) in the machinery system of the ship. The machinery systems on marine vessels need to fulfil demands for propulsion power, electrical power and heating. For large vessels, the propeller shaft is coupled directly to a slow speed two-stroke diesel engine, which delivers the required propulsion power. Electricity demands for pumps, fans, lighting, cooling, etc., are typically supplied by four stroke auxiliary engines or alternatively by a shaft generator mounted on the propeller shaft. Heating is required for space heating, HFO preheating and the generation of fresh water. Heating demands can be satisfied by auxiliary oil boilers or from waste heat sources on the ship. For example, heat from the jacket cooling water is typically used in the fresh water generator, while service steam can be generated in an exhaust gas boiler for satisfying space heating and HFO preheating demands. In large ships, the heating demands are lower than the available waste heat from the main engine. This enables utilization of the remaining waste heat energy for electricity production by means of suitable waste heat recovery (WHR) technologies. The electricity produced from WHR technologies can either be used for auxiliary power consumption and also for propulsion in case the addition energy produced from waste exceeds the auxiliary power requirements. In this way, the emissions from the machinery system and the fuel consumption can be reduced.

Over a few years, fuel consumption and emissions from international shipping have been significantly increased [1]. Researchers and environmental policy makers notify that ship emissions have been recognised as a growing problem and give an impact to an environmental pollution [2]. Production of exhaust gases and particles into the marine boundary layer from ocean-going ships contribute drastically to the full emissions from the transportation area [3]. Modern ships emits a lot of compounds such as carbon dioxide (CO_2), 2-3%, nitrogen oxide

(NO_x),10-15%, sulphur dioxide (SO_2), 4-9% and other gases such as carbon monoxide (CO), volatile compounds (VOC), sulphur dioxide(SO_2), black carbon (BC) and particulate organic matter (POM) [4-7] that have catastrophic environmental impacts . Latest studies reveals that ocean-going ships consume approximately about 289 million metric tons $(Mt)^2$ per year of fuel [4-7]. NOx emissions from shipping are fairly high due to most marine engines usually operate at high temperatures and pressures without effective reduction technologies. However, most of ocean-going ships also produce high amount of SO_2 emissions due to high average sulphur content of marine fuels (at around 2.4 to 2.7%)[5].The consumption of fossil fuels and discharge of pollutants of marine diesel engines is increasing rapidly with the progression of the navigation shipping industry. Due to increasing fuel prices and stricter upcoming regulations, there is high motivation in the marine sector to increase the propulsion system energy efficiency [8]. More attention on energy conservation and emission reduction has been paid in terms of waste heat recovery.

The waste heat sources are generally can classified by temperature range into low grade, medium grade and high grade. The temperature range of low grade are below 230 ^{0}C, medium grade around 230-650 ^{0}C and high grade are above 650 ^{0}C [11]. The energy recovery potential obtained from the waste heat of a marine diesel engine is obviously significant. Diesel engines have an efficiency of about 35% and the rest 60-70% of the energy released mostly in form of heat from the fuel by an engine is lost to surrounding [13]. Despite recent improvements of diesel engine efficiency, a considerable amount of energy is still expelled to the ambient with the exhaust gas [9].Without a suitable WHR system being integrated to the marine diesel engine, only half of the fuel energy converted into useful power output, while the remaining energy is expelled to the surrounding in the form of thermal energy through the exhaust gas, the jacket cooling water and others, such as the air cooling system and lubrication system [12]. As waste heat recovery is known as one of the promising energy saving methods and effective way to ensure fuel saving, waste heat recovery may be implemented in a large marine diesel engine to make a more efficient usage of fuels energy. In other words, implementation of waste heat recovery in marine diesel engine would address issues related to environmental improvement as well as energy crisis simultaneously by producing more power on the basis of the same emission quality [10].

1.2 Literature Review

The work done on Waste heat recovery by various researchers and its relevance in the present study is discussed at length in the following paragraphs. Many have identified that Waste Heat Recovery have a great application in marine diesel engine.

Recently it was reported by Mondal and De [14] that conversion of readily available low and medium grade heat into power and other energy utilities would reduce fossil fuel consumption and greenhouse gas emission simultaneously. Recent studies indicated that Organic Rankine cycle [15-18], Organic flash cycle [19-21], Transcritical CO_2 power cycle [22-24] are a few of the developing technologies to generate power from low and medium grade waste heat. In recent time, substantial research is going on marine diesel engine waste heat recovery. Mondejar et al. [25] evaluated the potential of using ORC systems for waste heat recovery of ships and the results show that ORC units recovering heat from the exhaust gases of engines using low-sulphur fuels could yield fuel savings between 10% and 15%. Mohd et al. [26] stated that in the highly efficient modern engines will have only around 25-50% of thermal efficiency and the balance 50-85% of low heating values of the fuel are wasting into the environment as heat and exhaust gas enthalpy, if the exhaust gas is transferred to surrounding directly. Jianqin et al .[27] reported that a diesel engine has the highest of thermal efficiency due to its exceptionally high compression ratio (spanning from 30 to 40%).The engine of ship particularly that of huge tonnage ship runs at a consistent pace for a long term compared to that of the car working conditions. Thus, it is simpler to make use of waste heat on ships compared to the waste heat utilization in other vehicles. In a marine diesel engine waste heat recovery system is the waste heat acts as the heat source and sea water acts as the heat sink. Shu et al. [10] state that by applying waste heat recovery, the system would produces other useful forms of energy on-board than direct heating. Fuel consumption and the sailing cost could be reduced appreciably by the application of waste heat recovery of a Marine Diesel engine [28]. Song et al. [29] showed that an ORC can be driven economically by the waste heat released by a marine diesel engine. Yang and Yeh [30] conducted thermodynamic and economic performances optimization for an ORC system, recovering the waste heat of exhaust gas from a large marine diesel engine of the merchant ship using R1234ze, R245fa, R600 and R600a. The results showed that R245fa performed the most satisfactorily in terms of the optimal economic performance, while R1234ze had the largest thermal efficiency. The use of ORC system could reduce 76% CO_2 emission per kWh. Yang [31] conducted the optimization of a marine diesel engine waste heat recovery system to select the best possible working fluids for the transcritical ORC. According to this analysis, R236fa appeared to be the best performing working fluids out of six considered working fluids. Yang [32] also evaluated the payback period for ORC based marine engine waste heat recovery system that utilized mixtures of different working fluids instead of a pure working fluid. Yang and Yeh [33] proposed a new parameter, namely "net power output index" to evaluate the economic performance of the marine diesel engine exhaust driven Transcritical Organic Rankine Cycle (TORC)

In many of the recent studies CO_2 based power cycles were considered for engine waste heat recovery mainly due to non-flammable, non toxic and environment friendly nature of CO_2 [34-35].CO_2 is also readily available at a lower cost. It is also preferred for engine waste heat recovery as it is chemically stable even at higher temperature [36]. However, operating pressure in the heat recovery unit of CO_2 based power cycle is appreciably high.

In present thesis, instead of considering a single working fluid, cascading between a CO_2 power cycle and an ORC is considered for the utilization of waste heat released by the marine diesel engine. The cycle is designated as CO_2-Organic Fluid cascading Cycle, with CO_2 cycle as the topping one. R600, R290 and R1233zd (e) are considered as the different alternative working fluid of the bottoming cycle of the cascading due to their lower GWP. The proposed system is analysed thermodynamically and economically. Results are presented by considering a Regenerative T-CO_2 power cycle as the baseline system.

2. System description and selection of working fluids

In our present work, exhaust gas, cylinder cooling water and scavenge air cooling water of a large marine diesel engine is considered as the three waste heat sources. Heat content along with the inlet and outlet temperature of the above mentioned streams are summarised in table-1 [33].

Table 1: Waste heat from a large marine diesel engine [33]

Waste heat source	Mass flow rate (kg/s)	Inlet Temperature (^{0}C)	Minimal exit temperature (^{0}C)
Exhaust gas	148.51	290	138
Cylinder cooling water	158	90	73
Scavenge air cooling water	162.5	76	36

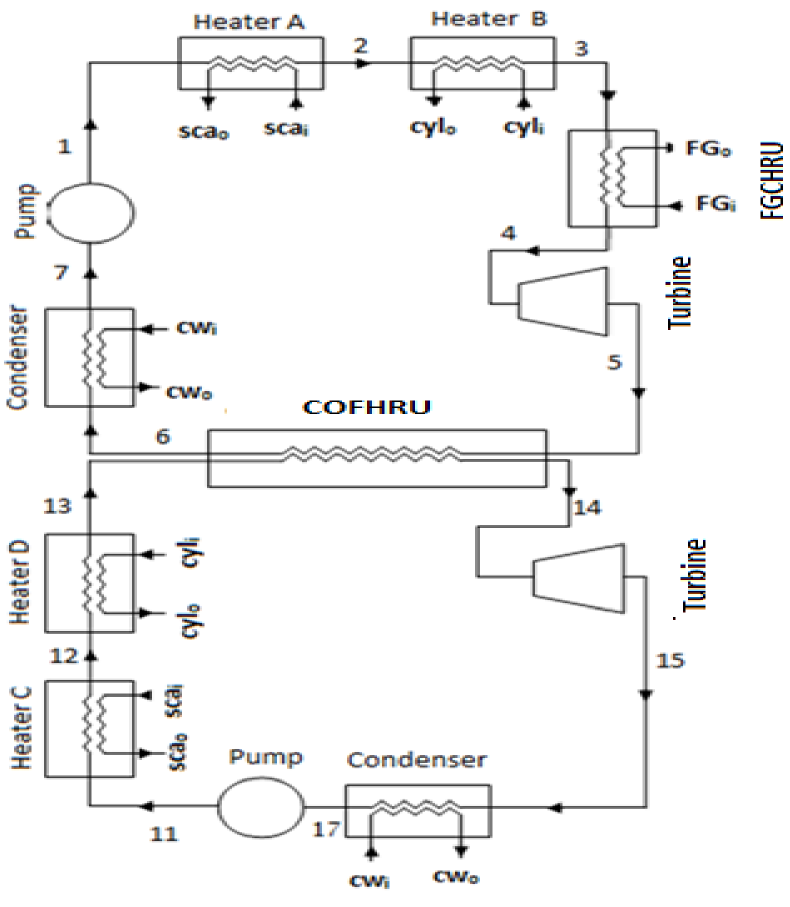

Fig 1: CO_2-Organic cascading Cycle

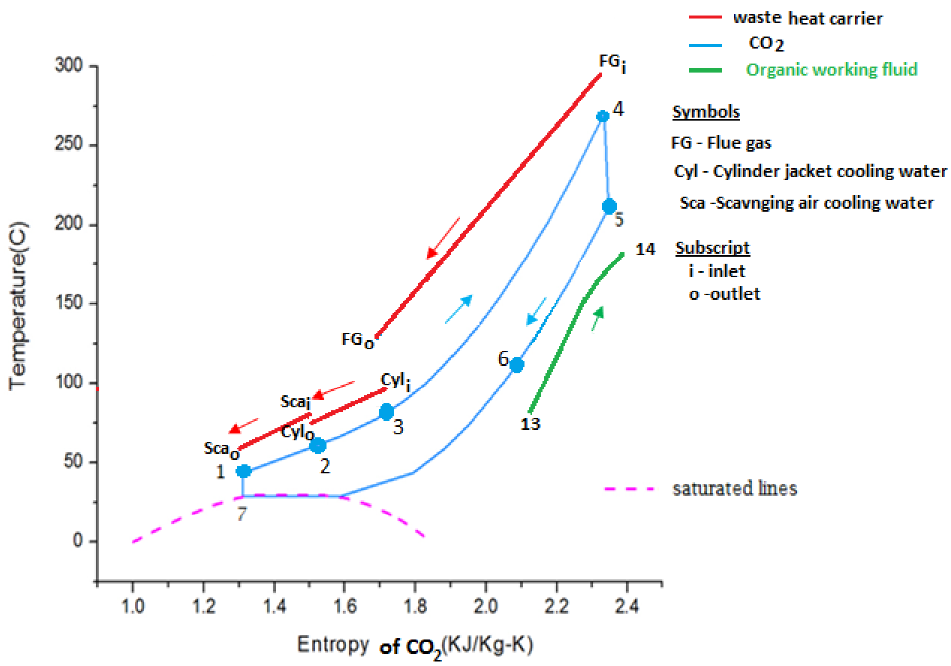

Fig. 2(a): T-S diagram for topping CO_2cycle of the cascading

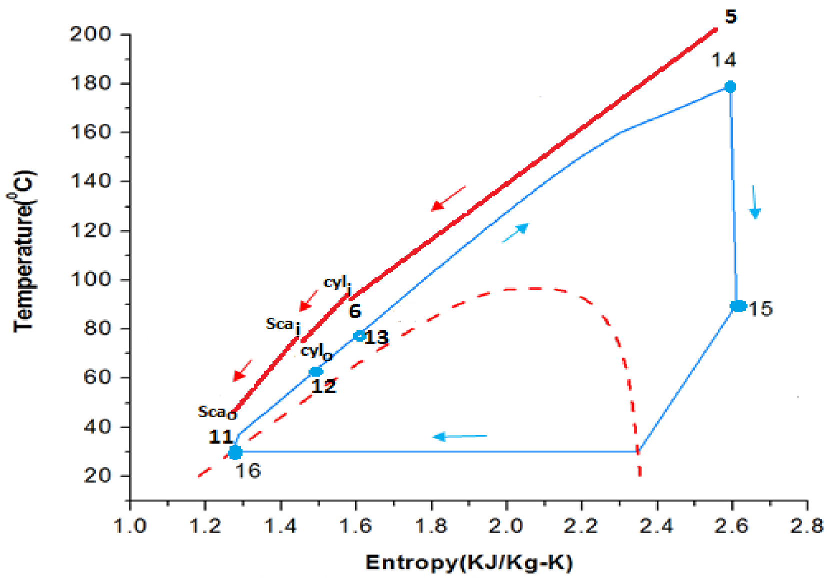

Fig. 2(b): T-S diagram for bottoming transcritical ORC of the cascading

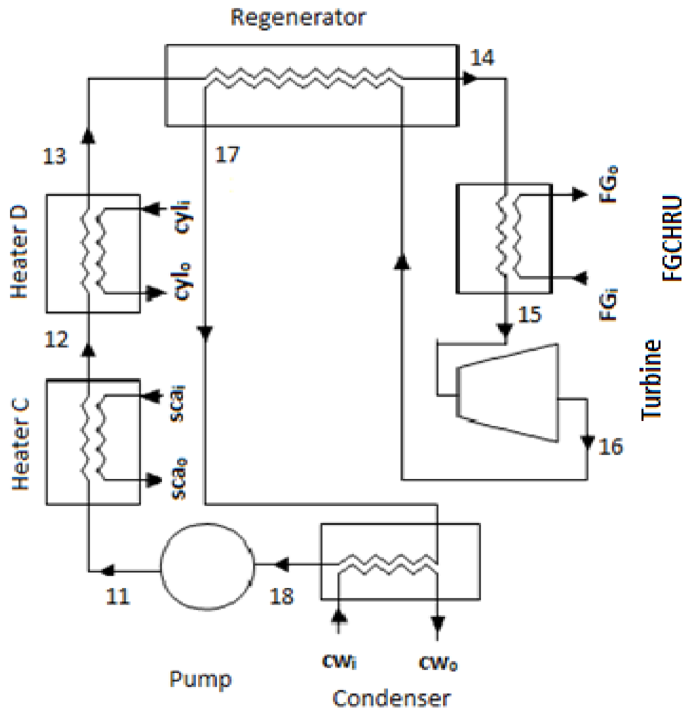

Fig. 3: Layout Regenerative T-CO_2 power cycle.

Layout of proposed CO_2-Organic Fluid cascading cycle, utilizing waste heat from a Marine engine is presented in Fig. 1. Fig.2 (a) and 2(b) are T-S diagrams for the topping transcritical CO_2 cycle and the bottoming transcritical ORC respectively.

The sequence of waste heat recovery by the CO_2 is according to the available temperature. CO_2 stream, exiting the pump at state-1 recovers heat from scavenging air cooling water, jacket cooling water and exhaust flue gas of a large marine diesel engine in heater-A (i.e process 1-2), heater-B (i.e process 2-3) and flue gas-CO_2 heat recovery unit (FGCHRU) (i.e process 3-4) respectively. The intention of the present study is to heat the CO_2stream close to the flue gas inlet temperature to ensure higher thermal efficiency. Thus the CO_2 stream is heated to 270^0C by the heat of exhaust flue gas stream. Mass flow rate of CO_2 stream is estimated from the energy balance of the FGCHRU.

The CO_2 mass at state-4 enters the expander (i.e. the turbine) to produce the power output. The temperature of CO_2 stream at the exit of the turbine (i.e. state-5) is appreciably high. Thus, this CO_2 is cooled (process 4-5) in a CO_2-organic fluid heat recovery unit (COFHRU) by exchanging heat to any one of the three selected organic fluids. The organic fluid exiting the COFHRU (at state-14) also expands (i.e. process 14-15) in a turbine to produce some power output. The organic fluid also recovers heat from the scavenging air cooling water and the jacket cooling water in heaters C (i.e. process 11-12) and D (i.e. process 12-13)respectively before entering the COFHRU. Organic fluid mass flow rate is estimated from the energy balance of the COFHRU. Energy balance of different heaters reveal that it is not possible to cool entire mass of scavenging air cooling water and jacket cooling water by the process of waste heat recovery.

Layout of the engine waste heat driven baseline T- CO_2 power cycle is presented in Fig. 3. In the baseline T-CO_2 power cycle, CO_2 stream exiting the jacket cooling water is heated in the regenerator by the heat of the CO_2 stream exiting the turbine. CO_2 mass flow rate for the baseline T-CO_2 power cycle is also estimated from the energy balance of the FGCHRU. For better representation of operating conditions, terminal temperature differences indifferent heat recovery units are presented in Table-2.

Table 2: Terminal temperature differences in HRUs

Cycle	Heat recovery unit	Δt_{LTE} (°C)	Δt_{HTE} (°C)
CO_2-organic cascading	FGCHRU	48	20
	Heater-A	10	10
	Heater-B	8	10
	Heater-D	10	10
	Heater-E	8	10
	COHRU	10	20
Baseline T-CO_2	FGCHRU	variable	20
	Heater-A	10	10
	Heater-B	8	10
	Regenerator	10	variable

Selection of suitable working fluid is critical as use of the Chlorofluorocarbon (CFCS) and most of the Hydro-chloro-fluorocarbons (HCFCs) are restricted due to either ozone depleting nature or higher values of GWP. HFCs are to be phased out soon according to Kigali amendment to the Montreal protocol. In present study two hydrocarbons (R290, R600) and one HFO (R1233zd (E)) refrigerant are considered as the working fluid of the bottoming cycle as listed in table-3 due to lower values of GWP. As turbine exit temperature of topping CO_2cycle varies between 190o to 235°C, the bottoming organic cycle can be operated in transcritical mode.

Table 3: Properties of selected refrigerants

Properties	R290	R1233zd(e)	R600
Critical temperature (°C)	96.74	166.45	151.98
Critical pressure (MPa)	4.2471	3.6237	3.7960
ODP	0	0	0
GWP(100 years)	3.3	1	3

3. Mathematical Modelling

In initial part of the mathematical modelling, equations are developed to represent the energetic performances of the waste heat recovery scheme. Thermodynamic and transport properties of various working fluids are evaluated using REFPROP-9. During the modelling following assumptions are considered to simplify the analysis:

I. All equipment runs at steady state steady flow conditions.

II. Turbine isentropic efficiency is 90%

III. Isentropic efficiencies of the pump as well as the compressor are assumed to be 85% each.

IV. Ambient condition is specified by 100 kPa and 20^0C.

V. Maximum permissible flue gas velocity is 15 m/s.

VI. All heat exchangers are assumed to have shell and tube configuration with multi pass arrangement.

VII. During heat exchanger design flue gas thermo-physical properties are assumed to be same as air.

3.1 Thermodynamic modelling:

Mass flow rate CO_2 through the topping cycle can be evaluated from the energy balance of the FGCHRU as follows:

$$\mathrm{m_{CO2}} = m_g c_{Pg} \frac{(\mathrm{T_{g,i}} - \mathrm{T_{g,o}})}{(\mathrm{h_4} - \mathrm{h_3})} \qquad (1)$$

Mass flow rate organic working fluid through the bottoming cycle can be evaluated from the energy balance of the COHRU as presented in eqn-2:

$$\mathrm{m_r} = \frac{\mathrm{m_{CO2}}(\mathrm{h_5} - \mathrm{h_6})}{(\mathrm{h_{14}} - \mathrm{h_{13}})} \qquad (2)$$

Power outputs from topping cycle turbine and bottoming cycle turbine are estimated in eqn-3 and eqn-4 respectively.

$$\mathrm{W_{t,tur}} = \mathrm{m_{CO2}}(\mathrm{h_4} - \mathrm{h_5}) \tag{3}$$

$$\mathrm{W_{b,tur}} = \mathrm{m_r}(\mathrm{h_{14}} - \mathrm{h_{15}}) \tag{4}$$

Equations 5 and 6 are representing pump power inputs of the topping cycle and the bottoming cycle respectively.

$$\mathrm{W_{t,pump}} = \mathrm{m_{CO2}}(\mathrm{h_1} - \mathrm{h_8}) \tag{5}$$

$$\mathrm{W_{b,pump}} = \mathrm{m_r}(\mathrm{h_{11}} - \mathrm{h_{17}}) \tag{6}$$

Now, net power outputs of the topping cycle as well as the bottoming cycle are evaluated as follows:

$$W_{t,NET} = W_{t,tur} - W_{\mathrm{t,pump}} \tag{7}$$

$$W_{b,NET} = W_{b,tur} - W_{\mathrm{b,pump}} \tag{8}$$

Now, power output of the cascading cycle

$$W_{Cascade} = W_{t,NET} + W_{b,NET} \tag{9}$$

3.2 Heat exchanger area estimation:

Heat exchangers are divided in "N" number of subsections for taking care of varying transport property of working fluids with varying temperature. Enthalpy drops across each of the subsections are assumed to be equal. Now area of any one of the subsections can be evaluated as follows:

$$\mathrm{A_{exh,j}} = \frac{\mathrm{Q_{exh,j}}}{\mathrm{U_{exh,j}} F \, \Delta \mathrm{T_{mean,exh,j}}} \qquad (10)$$

In eqn-10,$\Delta \mathrm{T_{mean,exh,j}}$ is LMTD for the counter flow arrangement and F is correction factor to take care of multi passes. Elemental heat duty of the eqn-10 can be estimated by eqn-11.

$$\mathrm{Q_{exh,j}} = m_{WF} \frac{|(h_i - h_o)|}{N} \qquad (11)$$

In eqn-11,h_i and h_o are enthalpies of working fluid in inlet and exit of the heat exchanger respectively.

Over all heat transfer coefficient of each of the heat exchanger element can be expressed as

$$U_{exh,j} = \frac{1}{{}^{1}/_{\alpha_{tube}} + {}^{1}/_{\alpha_{shell}}} \qquad (12)$$

α_{tube} and α_{shell} are tube side and shell side convective heat transfer coefficients respectively. Various correlations considered for convective heat transfer coefficients are summarised in Table 4.

Table 4: Correlations for estimation of convective heat transfer coefficient [37-40]

Equation of heat-transfer coefficient	Fluid	Phase	Heat Exchanger
$Nu = \left[\frac{\left(\frac{f_b}{8}\right) Re_r\, Pr_r}{(f_b/8)^{0.5}\left(Pr_r^{\frac{2}{3}}-1\right)+1.07}\right]\left(\frac{C_{p_{av}}}{C_{p_v}}\right)\left(\frac{k_b}{k_{wall}}\right)\left(\frac{\mu_b}{\mu_{wall}}\right)$ $0.5 \le Pr \le 2000$ $3 \times 10^3 \le Re \le 5 \times 10^6$	Working fluid	Supercritical	FGCHRU COFHRU Heaters A-D
$h_r = 55 Pr_r^{0.12-0.4343ln(R_p)}\left(-0.4343 ln(Pr_r)\right)^{-0.55} M^{-0.5}(q)^{0.67}$		2 phase Organic Fluid	COFHRU
$Nu = \left[\frac{\left(\frac{f_b}{8}\right) Re_r\, Pr_r}{(f_b/8)^{0.5}\left(Pr_r^{\frac{2}{3}}-1\right)+1.07}\right]$		CO_2 vapour	COFHRU
$Nu = 0.0131 Re^{0.883} Pr^{0.36}$ $4.5 \times 10^5 \le Re \le 7 \times 10^6$		Organic fluid Vapour	Condenser
$Nu = 0.729\left(\frac{g\, \rho_{f(\rho_f-\rho_g) D_o^3 i'_{fg}}}{\mu_f K_r\, (T_{sat} - T_{wall})}\right)^{1/4}$		2 phase organic fluid	Condenser
$Nu = 0.05 Re_{eq}^{0.8} Pr_{sat.liq}^{0.33}$ $Re_{eq} = Re_{vap} \frac{\mu_{sat\ vap}}{\mu_{sat\ liq}}\left(\frac{\rho_{sat\ liq}}{\rho_{sat\ vap}}\right)^{0.5} + Re_{liq}$ $Re_{liq} = \frac{m_f}{A_f} . (1-x) . \left(\frac{D_w}{\mu_{sat\ liq}}\right)$ $Re_{vap} = \frac{m_f}{A_f} . x . \frac{D_w}{\mu_{sat\ vap}}$		2-phase CO_2	Condenser
$Nu = \left[\frac{\left(\frac{f_b}{8}\right) Re_r\, Pr_r}{(f_b/8)^{0.5}\left(Pr_r^{\frac{2}{3}}-1\right)+1.07}\right]$ $0.5 \le Pr \le 2000$		Superheated CO_2	Condenser
$Nu = 0.71 Re^{0.5} Pr^{0.36}\left(\frac{Pr}{Pr_{wall}}\right)^{0.25}$ $1000 \le Re \le 2 \times 10^5$	Exhaust Gas	gas	FGCHRU
$Nu = 0.023 Re^{0.8} Pr^{0.3}$ $Re > 10^4$ $0.7 \le Pr < 160$	Jacket cooling water, scavenging air cooling water, cooling water	Liquid	Heaters A,B,C,D, Condenser

3.3 Bare module cost estimation:

In order to estimate the cost of the equipment for preliminary design, the cost equations were proposed by Turton et al. [41] are employed. Equation used for the purchased cost of individual equipment (C_p^0) at ambient operating pressure and using carbon steel (CS) construction is as follows:

$$log_{10}C_p^0 = K_1 + K_2 log_{10}Z + K_3(log_{10}Z)^2 \quad (13)$$

where, Z is the parameter for capacity and size of the equipment as provided in the Table 5.K_1 K_2 K_3 are the constants, as shown in Table 5. Since the equipment rarely operate at ambient pressure, pressure factor F_p is used to take care of elevated operating pressure. The bare module cost for shell-and-tube heat exchangers and pump are given by

$$C_{BM} = C_P^0(B_1 + B_2 F_P F_M) = C_P^0 F_{BM} \quad (14)$$

Bare module cost of turbine is expressed as

$$C_{BM} = C_P^0 F_P F_{BM} \quad (15)$$

In these equations F_P; F_M and F_{BM} are pressure factor, material factor and bare module factor respectively constants. B_1 and B_2 are constants as presented Table 5. [25]. F_p can be estimate from following equation

$$log_{10}F_p = C_1 + C_2 log_{10}(10P - 1) + C_3(log_{10}(10P - 1)^2 \quad (16)$$

C_1, C_2, and C_3 are constants whose values are also provided in Table 5. In eqn-16, P is the operating pressure in MPa Subsequently the total cost of the equipment is obtained by adding the cost of individual equipment used in the system as shown below

$$C_{Tot} = (\sum C_{BM,eq}) * CEPCI_{currentyear}/CEPCI_{2001} \quad (17)$$

In eqn-17, CEPCI is the chemical engineering plant cost index, taking the effect of time on purchased equipment cost into account.

Table 5: Equipment cost parameters [44]

<table>
<tr><th>Equipment</th><th>Performance parameters (Z)</th><th>K_1</th><th>K_2</th><th>K_3</th><th>B_1</th><th>B_2</th><th>F_M</th><th>C_1</th><th>C_2</th><th>C_3</th></tr>
<tr><td>FGCHRU</td><td>A_{exh} (m^2)</td><td rowspan="3">4.3247</td><td rowspan="3">-0.303</td><td rowspan="3">0.1634</td><td rowspan="3">1.63</td><td rowspan="3">1.66</td><td rowspan="3">1.4</td><td rowspan="3">0.0388</td><td rowspan="3">-0.11272</td><td rowspan="3">0.08183</td></tr>
<tr><td>Heaters A,B,C,D</td><td>A_{sca},A_{cyl} (m^2)</td></tr>
<tr><td>Condenser</td><td>A_{con} (m^2)</td></tr>
<tr><td>COHRU</td><td>A_{reg}(m^2)</td><td>4.3247</td><td>-0.303</td><td>0.1634</td><td>1.63</td><td>1.66</td><td>1.4</td><td>-0.395</td><td>0.3957</td><td>-0.00226</td></tr>
<tr><td>Pump</td><td>W_{pump}(kW)</td><td>3.3892</td><td>0.0536</td><td>0.1538</td><td>1.89</td><td>1.35</td><td>-0.3935</td><td>-0.395</td><td>0.3957</td><td>-0.00226</td></tr>
<tr><td>Turbine</td><td>W_{Tur} (kW)</td><td>2.7051</td><td>1.4398</td><td>-0.1776</td><td>0</td><td>1</td><td>3.4</td><td>0</td><td>0</td><td>0</td></tr>
</table>

4. Result and discussion

In the present study, a cascading between T-CO_2 power cycle and Organic Rankine cycle is considered for the recovery of waste heat rejected by a large marine diesel engine. Results are presented by considering a regenerative T-CO_2 power cycle as the baseline one.

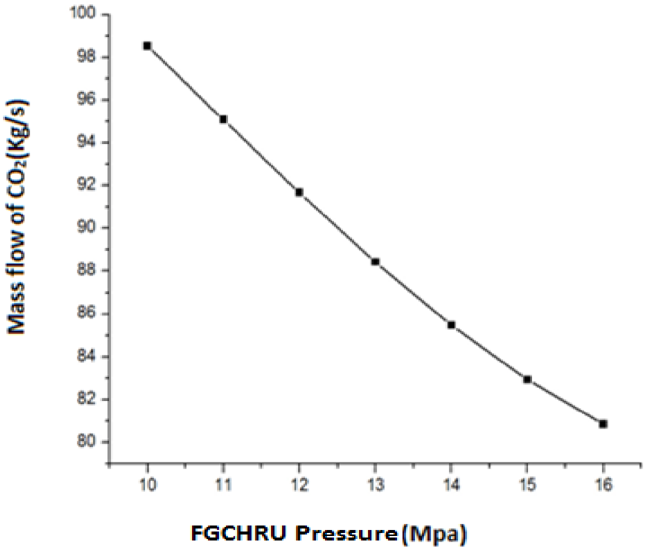

Fig. 4(a): CO_2 mass flow rate of the topping CO_2 power cycle vs. FGCHRU pressure

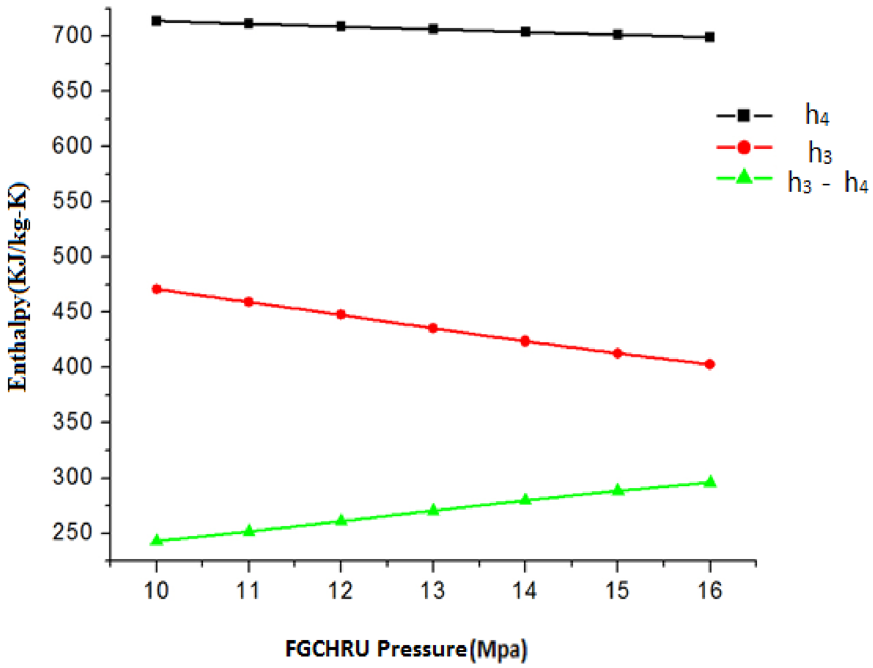

Fig. 4(b): State point Enthalpy variation with varying pressure in FGCHRU

The mass flow of CO_2 for the cascading cycle is determined from the energy balance between CO_2 and exhaust gas in the flue gas-CO_2 HRU (FGCHRU). It can be seen from Fig.4 (a) that the mass flow rate of CO_2 decreases with an increase in FGCHRU Pressure. This can be easily explained from Fig.4 (b). It is observed in the Fig.4 (b) that with an increase in pressure of the FGCHRU, both h_3 and h_4 decreases. However, difference between h_3 and h_4 increases with an increase in pressure of the FGCHRU. Thus, heated mass of CO_2 reduces as heat released by flue gas is constant.

Power output of the topping cycle of the cascading increases with an increase in FGCHRU pressure as shown in Fig.5. The total power is directly proportional to product of mass and the specific work output. Although the mass flow rate decreases, it is overcompensated by specific work output and thereby increasing the power.

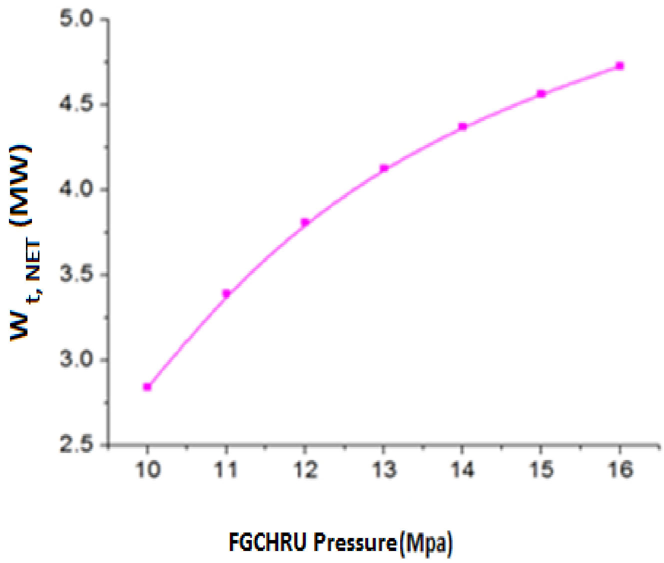

Fig 5: Power output of the topping CO_2 power cycle vs. FGCHRU pressure

With an increase in topping cycle turbine inlet pressure, CO_2 mass flow rate decreases as already presented in Fig. 4(a). Temperature of CO_2 exiting the turbine of the topping cycle also reduces with an increase in FGCHRU pressure as shown in Fig. 6. Lower turbine exit temperature of the topping cycle also results in lower turbine inlet temperature for the bottoming ORC. Thus, total heat available for heating the organic fluid of bottoming cycle as well as efficiency of the bottoming cycle reduces. Due to reduction in heat input as well as thermal efficiency, power output of the bottoming cycle decreases with an increase in FGCHRU pressure as shown in Fig. 7.

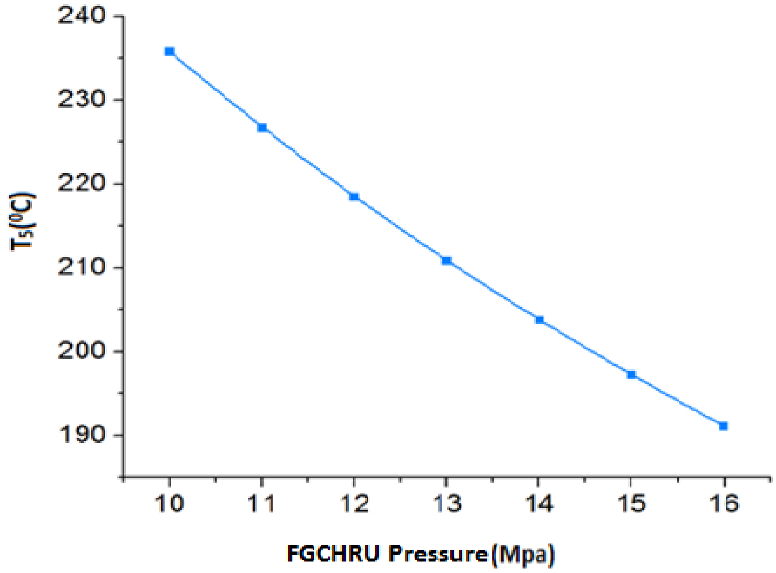

Fig. 6: Turbine exit temperature (T_5) vs. FGCHRU pressure

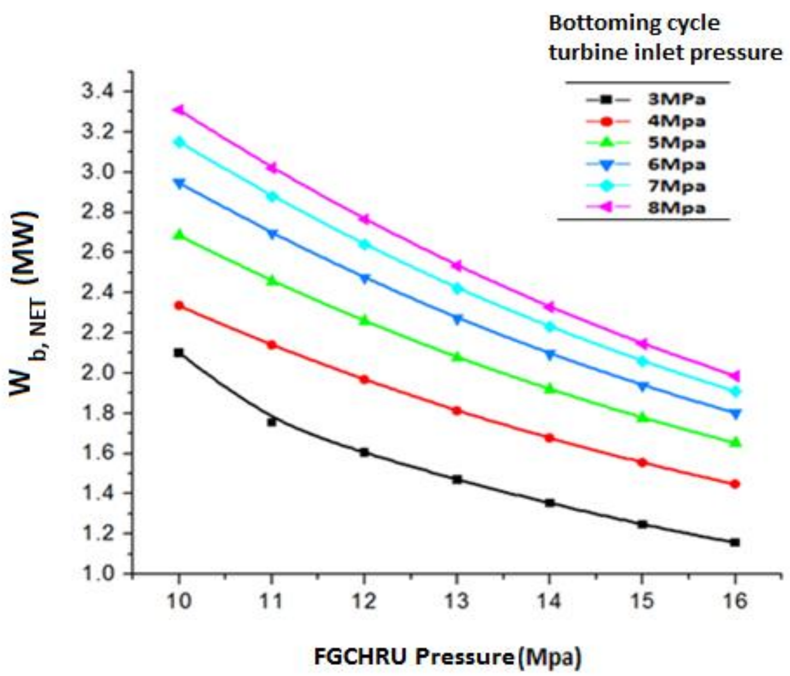

Fig 7: Power output of the bottoming ORC with R290 vs. FGCHRU pressure.

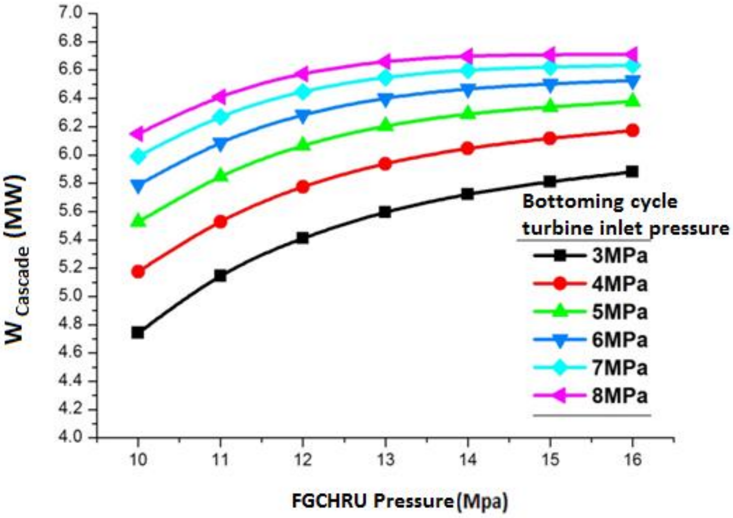

Fig 8: Power output of the CO_2-R290 cascading cycle vs. FGCHRU pressure

The total power output of the cascading cycle ultimately increases with an increase in FGCHRU pressure as shown in Fig.8. It is important to note that improvement achieved in total power output of CO_2-organic cascading cycle becomes negligible above a certain value of FGCHRU pressure. It is also observed in Fig.8 that for a specified working fluid and FGCHRU pressure, total power output of the cascading cycle increases with an increase in bottoming cycle turbine inlet pressure. However, above a certain value of bottoming cycle turbine inlet pressure, this improvement is almost negligible. Thus for a specified turbine inlet pressure of FGCHRU, there exists a turbine inlet pressure of the bottoming cycle above which no appreciable improvement in power output of the cascading cycle occurs.

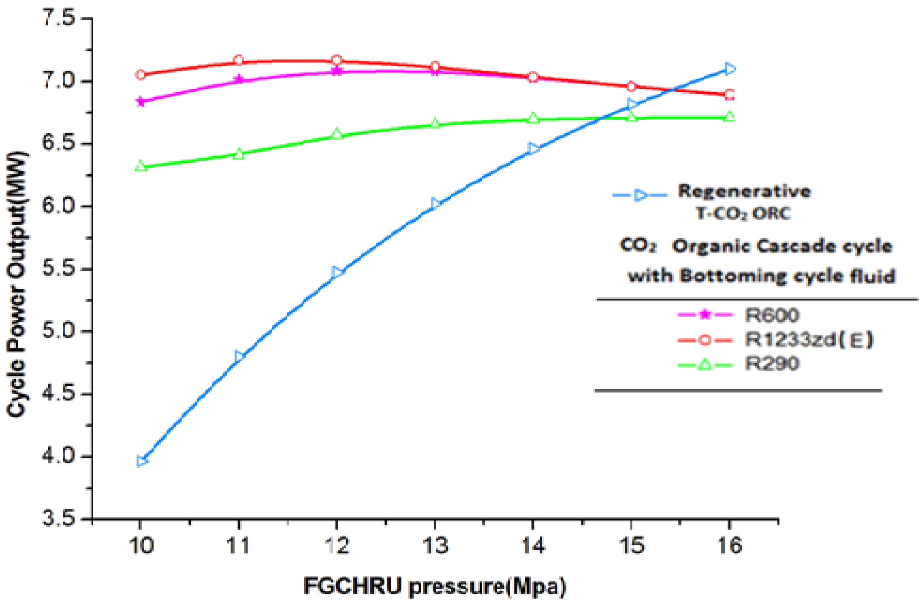

Fig. 9: Comparison of power output of cascading cycle with that of the baseline T-CO_2 cycle

In Fig.9 power outputs of the cascading cycle are compared with the power outputs of the baseline cycle (i.e. the regenerative T-CO_2 power cycle) for varying FGCHRU pressure. R290, R600 and R1233zd (E) are working fluids considered for the bottoming cycle of the CO2-Organic cascading cycle. For all cascading systems, power outputs are considered at the bottoming cycle turbine inlet pressure above which power output of a cascading cycle becomes almost insensitive to the varying bottoming cycle turbine inlet pressure. The total power output of the cascading cycle increases with an increase in pressure of the FGCHRU and reaches to a peak if R1233zd (e) or R600 is used as the working fluid of the bottoming cycle. For R290, above a certain value of the FGCHRU pressure, the variation of cascading cycle power output is negligible. It is also observed that for lower values of FGCHRU pressure (<15MPA), the cascading cycle can yield appreciably higher power output compared to that of the baseline regenerative T-CO2 power cycle with all three selected working fluids of the bottoming cycle. The cascading cycle yields the highest power output if R1233zd (e) is used as the working fluid of the bottoming cycle of cascading. As pressure in the FGCHRU is increased, the power output of the baseline cycle increases sharply and becomes comparable to that of the cascading cycles as the pressure in the FGCHRU reaches close to 15MPa. The baseline cycle would deliver higher power if the pressure in the FGCHRU is increased beyond 16 MPa.

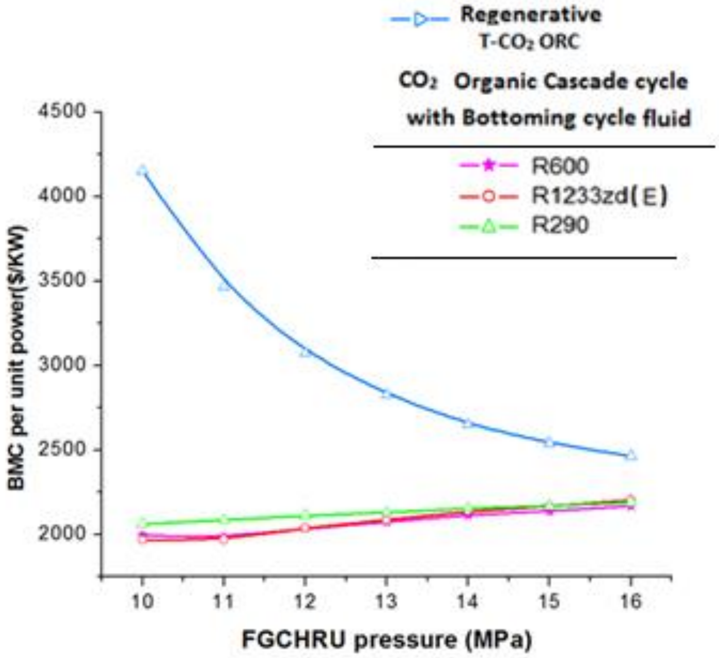

Fig. 10: Effects of varying FGCHRU pressure on BMC per unit power

Effects of varying FGCHRU pressure on bare module cost per unit power output is presented in Fig.10. It is observed in Fig.10 that at a lower FGCHRU pressure BMC per unit power is appreciably higher for the baseline T-CO_2 power cycle. However, BMC per unit power output reduces sharply as elevated pressure is ensured in the FGCHRU pressure. It should be noted that though the total BMC of the base line cycle increases with an increasing FGCHRU pressure, BMC per unit power sharply reduces with the increasing pressure of FGCHRU due to steady rise in cycle power output. Beyond a certain value of the FGCHRU pressure, reduction achieved in BMC per unit power output of the baseline cycle is negligible.

It is further observed in Fig. 10, that BMC per unit power output of all the cascading cycles are appreciably smaller compared to the baseline cycle, especially at lower pressures of the FGCHRU. This is due to higher power outputs of cascading cycles at lower operating pressures of FGCHRUs. However, BMC per unit power of the cascading cycle increases slowly (for R290 and R600) with increasing pressure of the FGCHRU.

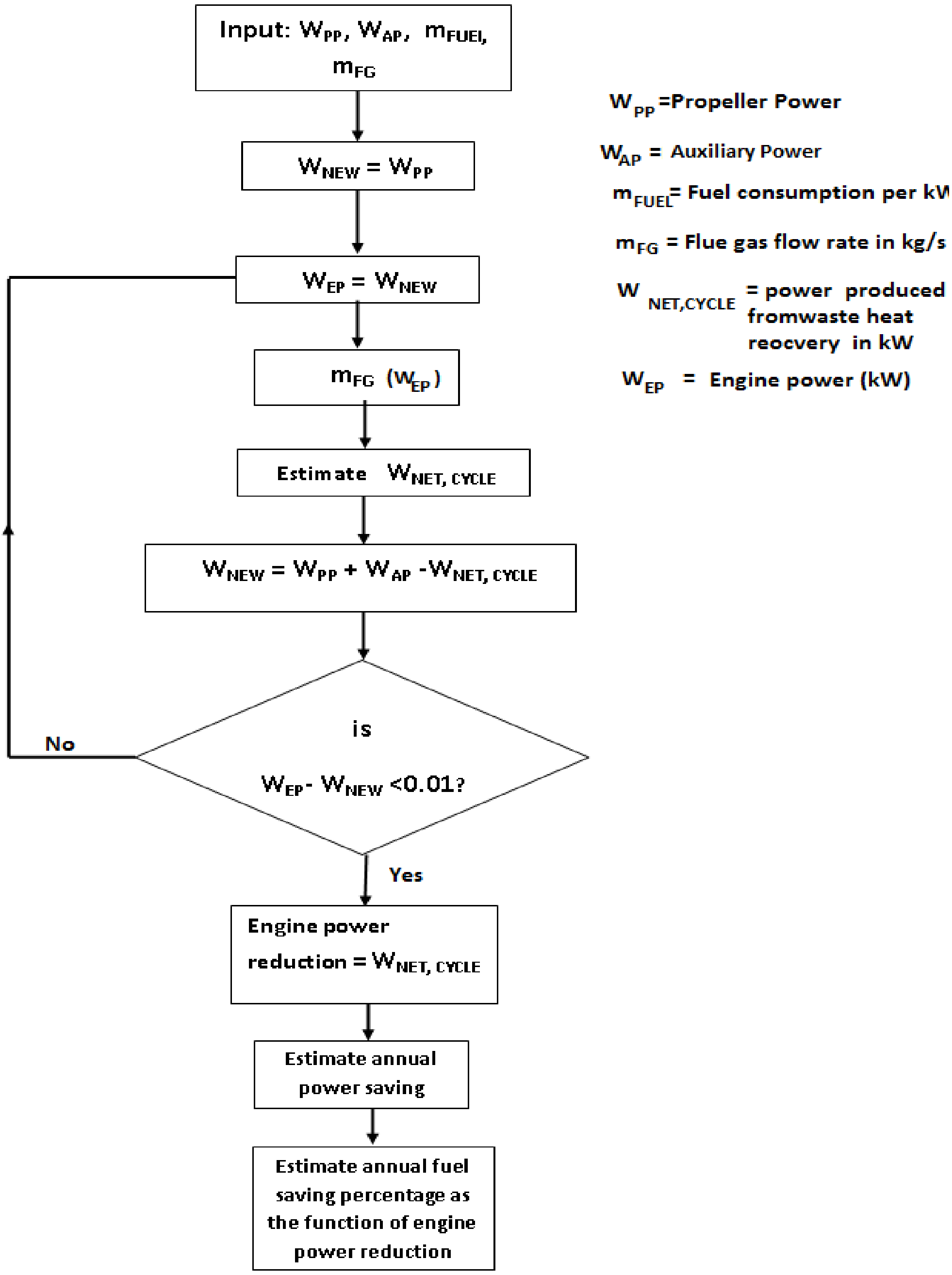

Fig.11: Methodology for the estimation of fuel saving

It should be noted that the auxiliary power requirement of a ship may be assumed to be 5% of the total power output of a Marine diesel engine [26]. Thus the additional power produced from waste heat recovery can be utilized to supply the auxiliary power. The power produced by the waste heat recovery scheme may be higher to some extent compared to the auxiliary power requirement of the ship. In this situation, after catering the auxiliary power, the remaining power of the waste heat recovery system may be utilized to reduce propeller engine power requirement. However, this would affect output of the waste heat recovery unit as available waste heat would also reduce. Thus, if power output of the waste heat recovery unit is higher compared to the power requirement of the auxiliary unit, fuel savings due to the incorporation of waste heat recovery scheme can be estimated through an iterative calculation as shown in Fig.11.

During this calculation, fuel consumption is assumed to be 0.167kg/kW-h and annual operation hour is assumed to be 7200 hours. Waste heat released by the engine supplying the propeller power is only considered for the waste heat recovery. The annual fuel savings is closely related with the additional power produced by the proposed Waste Heat Recovery Scheme. The more the power produced, the greater is the fuel saving.

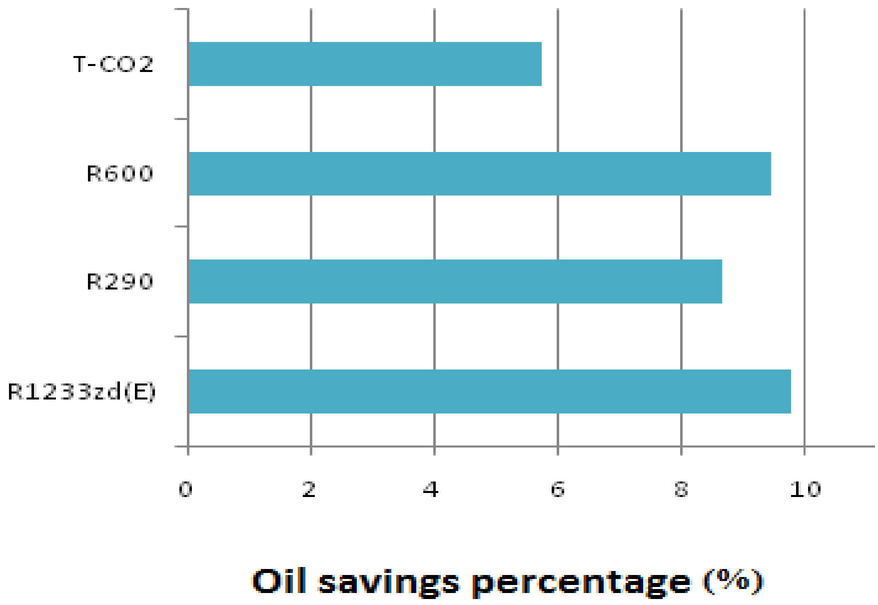

Fig.12 (a): Annual fuel savings due to waste heat recovery at FGCHRU Pressure 10MPa

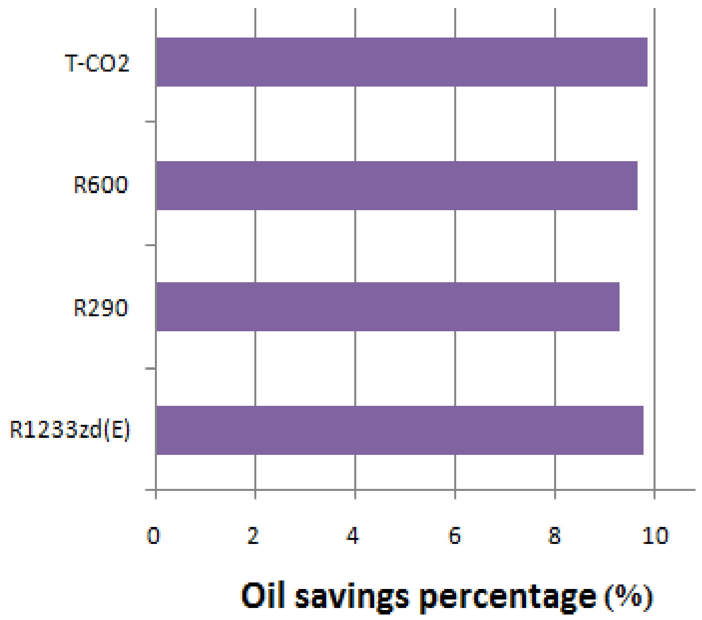

Fig.12 (b): Annual fuel savings due to waste heat recovery at FGCHRU Pressure 16MPa

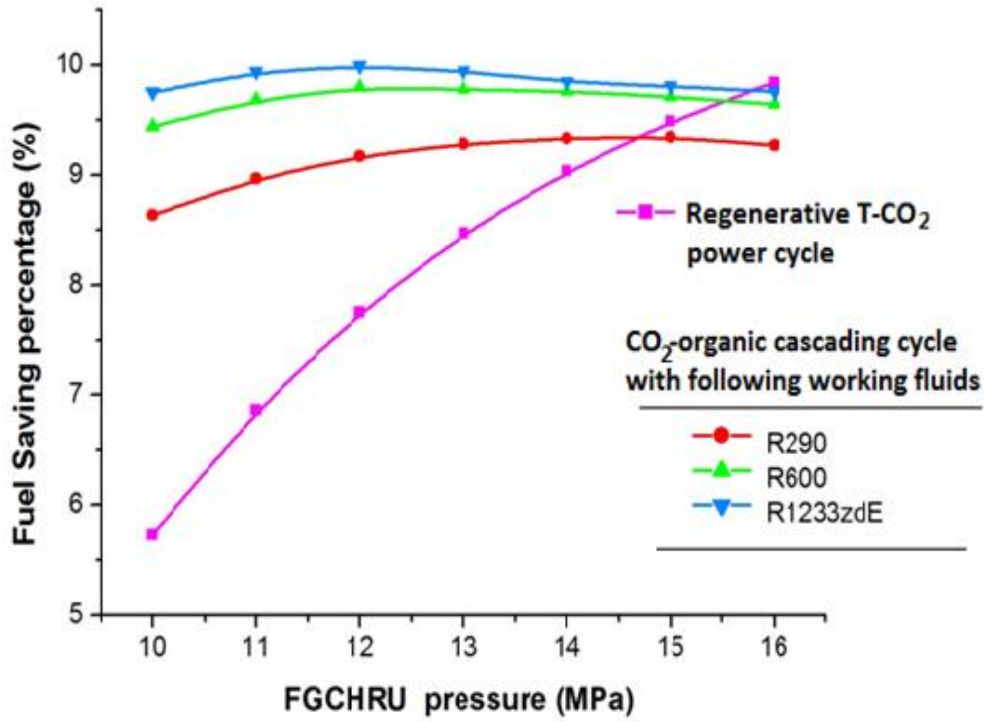

Fig.13: Annual fuel savings FGCHRU Pressure

The percentage of oil saved is plotted for CO_2-Organic Fluid Cascade System and Regenerative T-CO_2 cycle operating at FGCHRU pressures of 10 MPa (Fig. 12(a)) and 16MPa (Fig. 12(b)) respectively. It is observed that at lower operating pressure of 10 MPa, theCO_2-R1233zd (E) cascading cycle can save 9.752% of fuel annually while the same is significantly lower in Regenerative T-CO_2 at 5.73%. The annual fuel saving percentage improves significantly for Regenerative T-CO_2 cycle when the FGCHRU pressure is increased. The annual fuel savings for Regenerative T-CO_2 cycle at 16MPa is 9.84% which is marginally higher than the CO_2 Organic Cascade Cycle. It is interesting to note from Fig. 13 that the annual fuel savings of CO_2 Organic Cascade Cycle does not vary significantly with FGCHRU pressures but that of the Regenerative T-CO_2 cycle improves significantly, owing to higher power output at higher pressure. However, operating at lower pressure is always better from capital investment as well as operational simplicity. Hence, cascaded cycle will provide better overall performance than T-CO_2 cycle with better oil saving operating at lower pressure.

5 Conclusions and Future Scopes

5.1 Conclusions

In the present study, a cascading between T-CO_2 power cycle and Organic Rankine cycle is considered for the recovery of waste heat rejected by a large marine diesel engine. R290, R600 and R1233zd (e) are selected as the working fluids of the bottoming cycle considering their lower GWP. The cascading cycle can produce significantly higher power by using the available waste heat compared to that of the regenerative T-CO_2 power cycle if pressure in the fluegas-CO_2 heat recovery (FGCHRU) unit is kept below 14 MPa. However, a regenerative T-CO_2 power cycle would deliver higher power for some higher value of the FGCHRU pressure (i.e. More than 16 MPa). Bare module cost per kW is significantly small for the cascading cycle – though difference of BMC per unit power output of the cascading cycle and that of the baseline regenerative T-CO_2 power cycle sharply reduces as operating pressure of the FGCHRU is increased. R1233zd (e) appears as the best performing working fluid, as the cascading cycle that is using R1233zd (e) as bottoming cycle fluid yields highest power and lowest BMC per kW. R1233zd (e) is also less flammable compared to remaining two selected working fluids. Use of R1233zd (e) as the working fluid also ensures annual oil savings which is comparable to oil savings can be achieved by operating the T-CO_2 power cycle with a heat recovery unit pressure close to 16 MPa.

5.2 Future Scope of work

The entire Waste Heat Recovery system has been designed and analysed considering steady supply of waste heat sources at a particular operating condition. However, a ship might not run at the desired operating conditions all the time. Hence, there is scope for analysing off design operations. The variation of ambient temperature can be a cause of off design operation. In the given problem we can also integrate an ejector driven refrigeration cycle into the Waste Heat Recovery System as the turbine outlet temperature of the Bottoming Cycle is in appreciably high. Significant amount of waste heat of the cylinder jacket cooling water and scavenging air cooling water was unused. Thus the waste heat recovery system may be optimized further to minimize the unused waste heat.

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