4E Performance Analyses of Supercritical Carbon Dioxide Bottoming Cycle for Shipboard Applications

O.P.Sharma[a*], S.K.Tyagi[b], K.Manjunath[c], and S.C.Kaushik[b]

[a]1MP Naval Unit NCC, Bhopal, Madhya Pradesh462002,India;

[b]Department of Energy Science and Engineering, Indian Institute of Technology, Delhi, Hauz Khas, New Delhi 110016, India;

[c]Department of Mechanical Engineering, Delhi Technological University (Formerly Delhi College of Engineering) (Govt. of Delhi) Bawana Road Delhi 110042, India.

[*]Corresponding author: opsharma.iitd@gmail.com

Synopsis

Typically, shipboard gas turbine exhibits higher exhaust temperatures (ranging from about 450^{0}C at around 25% load, up to around 575^{0}C at the rated load) as compared to their diesel engine equivalent, which implies that a higher amount of useful thermal energy vents out through its exhaust. The thermal exergy contained in a typical LM2500 exhaust can be tapped to generate additional power by thermodynamically inter-connecting a supercritical CO_2 based bottoming power cycle. This research article therefore presents investigations of Energy-Exergy-Economic & Environment (*4E)* performance analyses of supercritical carbon dioxide regenerative waste heat recovery (bottoming cycle) power cycle thermodynamically coupled with LM2500 gas turbines (topping cycle) onboard a typical frigate class platform to improve overall plant efficiency and produce additional power. A range of 100% to 10% load has been considered since onboard naval ships, gas turbines scarcely operate at 100% power (only around 1-2% of the entire life) while because of the parabolic nature of the propeller (load) curve, fleet speeds between 12 to 16 knots are achieved with GTs running around 40 % (8.8 MW) or lower of their rated power (22 MW). With the proposed integration, significant improvement (~ 11%) in both energy & exergy efficiency of the shipboard GT is accruable, besides an additional power increment of around 4.8 MW (~ 22% of the GT rated power) without any extra fuel and carbon emissions. With the novel energy recovery system, ship can achieve additional range (26490 nm) and additional endurance (almost 69 days-at-sea) per year. In addition, the fleets can save significant carbon emissions of 4100 (ton-[CO_2]/yr/ship) at 60% relative GT load, besides earning carbon-credits worth about USD 61501 at 60% relative GT load per ship annually.

Keywords: Energy-Exergy-Economic-Environment (*4E*); Supercritical CO_2; LM2500 gas turbines; Bottoming cycle; Frigate; Shipboard applications.

Author's Biography

Dr O. P. Sharma, Ph.D. from IIT Delhi for pioneer research on Supercritical CO_2 based Energy Systems for Shipboard Power Generation & Cooling Applications. He has more than 25 years' experience in operations, maintenance, training, design & research in marine engineering & propulsion systems. He is also a Life Member of Institute of Marine Engineers (India). He is currently serving as marine engineering officer in the Indian Navy.
Dr S. K. Tyagi, a doctorate from IIT Delhi has more than 27 years of research experience and expertise in the area of renewable energy both solar thermal and biomass energy and other energy related fields. He is presently working as Associate Professor at Department of Energy Science and Engineering, Indian Institute of Technology Delhi since April, 2017.
Dr. Manjunath K. obtained M.Tech., Ph.D. degrees from IIT Delhi and Fellow of Institution of Engineers India. He has more than 21 years of teaching and research experience in the field of thermal engineering. He is currently working as Assistant Professor, Department of Mechanical Engineering, Delhi Technological University, Delhi, India.
Prof. S.C. Kaushik., Ph.D. in Plasma Science, has over 45 years of pioneer research & teaching experience at IIT Delhi in the field of Thermal Science & Engineering, Energy Conservation & Heat Recovery, Solar Power Generation, Alternative Refrigeration & Air Conditioning, and Thermal Energy Storage & Solar Power Generation. He has been on the editorial advisory board of many National/ International Journals of repute including Applied Thermal Engineering, Elsevier. Currently, he is working on Phase Change Thermal Energy Storage, Solar Sorption, Thermoelectric Cooling and Energy Efficient Buildings.

1. Introduction

The gas turbines are known for their inherent advantages viz., higher power density, compactness, low weight, quick start-up, low noise, and modularity as compared to the diesel engine in the power generation field. However, the gas turbines are also associated with relatively lower rated as well as part-load efficiencies resulting in higher fuel consumption and operating costs. Advanced measures such as design modifications, advanced aerodynamics, novel materials, advanced blade cooling and fabrication techniques besides, cycle efficiency improvement processes such as re-heating, inter-cooling, regeneration, isothermal heat addition and/or their combinations (Kaushik *et al.* 2003, 2017, Tyagi *et al.* 2005, 2006, 2008, Frost *et al.*1997, Göktun and Yavuz 1999) have resulted in modern GTs that can withstand higher inlet turbine temperature (approaching 1800 ^{0}C) and rated-load energy efficiency of around 40 percent or above. Another approach for further improving the efficiency is through integrating an external exhaust-waste-heat-recovery system, which apart from significantly improving efficiency also retains inherent advantages associated with modern gas turbines (Pierobon*et al.* 2014, Baldi and Gabrielii 2015, Singh and Pedersen 2016).

Further, for a typical open-cycle gas turbines, the exhaust gas characteristics at both rated and higher part-loads are relatively more favourable for waste heat recovery owing to higher temperature of exhaust (>400 ^{0}C), quantity of heat (MW-scale) and quality of heat (Medium-to-high grade), as compared to conventional diesel engines (Harrington1992). The shipboard gas turbines exhaust gases having temperature range at ~ 430 to 570 ^{0}C or ~ 700 to 840 K can be considered as the medium grade energy source which if recycled, will not only generate additional power output, cooling or heating but would also serve as Infra-Red-Suppression-System (IRSS) to reduce the infra-red (IR) signatures of modern gas turbines. The steam cycles, though efficient and well proven, have extensive space and weight implications and are, therefore usually configured for the land based WHR applications, and particularly with plant capacity above 120 MW (Moroz *et al.*2015), whereas relatively newer options such as organic Rankine cycle (ORC) based systems, though compact, are however, effective only for medium to low grade energy sources (up to about 300 ^{0}C) (Sarkar 2018). Carbon Dioxide (CO_2) is a stable, non-toxic, non-flammable, nil-ODP, minimum-GWP and natural working fluid having low critical point (30.98^{0}C and 73.8 bar) and high critical density (468 kg/m^3) as compared to conventional steam (373.95^{0}C and 220.6 bar, critical density=322 kg/m^3) (Feher1968, Angelino1969). Some advantages of regenerative Brayton cycles (RBC) employing supercritical CO_2 (SC-CO_2) over conventional working fluids like steam or helium or air are reported in literature (Dostal *et al.*2004, Sarkar2009, Neises and Turchi2014, Ahn *et al.*2015, Zhao *et al.*2015, Crespi *et al.*2017, Kim*et al.*2017) as: (a) greater power density (b) higher average heat input temperature (c) compactness of heat exchangers and the turbo-machinery (almost 10 times more compact) (d) up to 30% higher efficiency for energy recovery (e) higher heat transfer (f) lower leakage rates (g) simpler designs (h) cost effectiveness and (i) reduced footprint & overall weight. Sharma *et al.* (2017) investigated thermodynamic aspects of energy recovery system using regenerative-recompression (RRCBC) variant; while Akbari and Akbari and Mahmoudi(2017) studied SC-CO_2 based co-generation system. Thermodynamic analysis and optimisation of performance of RBC and RRCBC variants of SC-CO_2 cycles for WHR from modern gas turbine were investigated (Hou *et al.*2017, Cheng *et al.*2017, Khadse *et al.*2018) and reported them to be about 28% more economical than conventional steam cycle, improved part-load performance at an optimum pressure ratio. A novel SC-CO_2 cycles based system is reported to also help ships significantly boost the power output (by ~18%) as well as their installed cooling capacity (by ~892 TR) Manjunath *et al.*(2018).

Thermodynamic performance analyses of a WHR based power generation system was presented by Butcher & Reddy (2007) while analyses of supercritical CO_2 power (Brayton) cycle (RBC) was presented by Persichilli *et al*.(2012) and Kacludis *et al.*(2012)whereas Wright *et al.*(2016) investigated its thermo-economic performance. Energy & exergy based thermo-economic analyses of Brayton closed-loop power system was presented in a systematic manner(Moran *et al.*2010, Cenjel *et al.*2011) with the application of simulation tools such as Klein (2008), which employs new equation of state for CO_2(Span and Wagner(1996).Various combined cycle systems based on CO_2 as the working fluid for the bottoming cycle for energy recovery from gas turbines in an offshore oil and gas installation have been studied by Walnum *et al.*(2013), Nord and Bolland(2013) and Zhang *et al.*(2016). Recently, a novel concept comprising of hybrid combined cycle power plant (CCPP) based on GE LM6000 GT (topping cycle) with energy storage was presented by Herrmann *et al.* (1992). Unlike energy analyses, the exergy analyses, in addition to the exergy balance, also accounts for entropy generation, and exergy destruction or irreversibility associated in a system (Kotas 2012, Bejan 2002, Dincer and Zamfirescua 2014). This paper, therefore investigates the *4E* performance of supercritical CO_2 based bottoming cycle for typical gas turbine in a frigate class ship using typical operating parameters and exhaust gas emission profile data of typical marine gas turbines available in the literature (Sharma *et al.*2017, GE website 2018).

2. System Description and Analysis

The proposed shipboard WHRS with Carbon Dioxide (SC-CO_2) Regenerative Brayton Cycle (RBC) contains five prime components namely, supercritical CO_2 turbine and compressor, regenerator, precooler and heat recovery heat exchanger (HRHE) as shown in Figure 1.

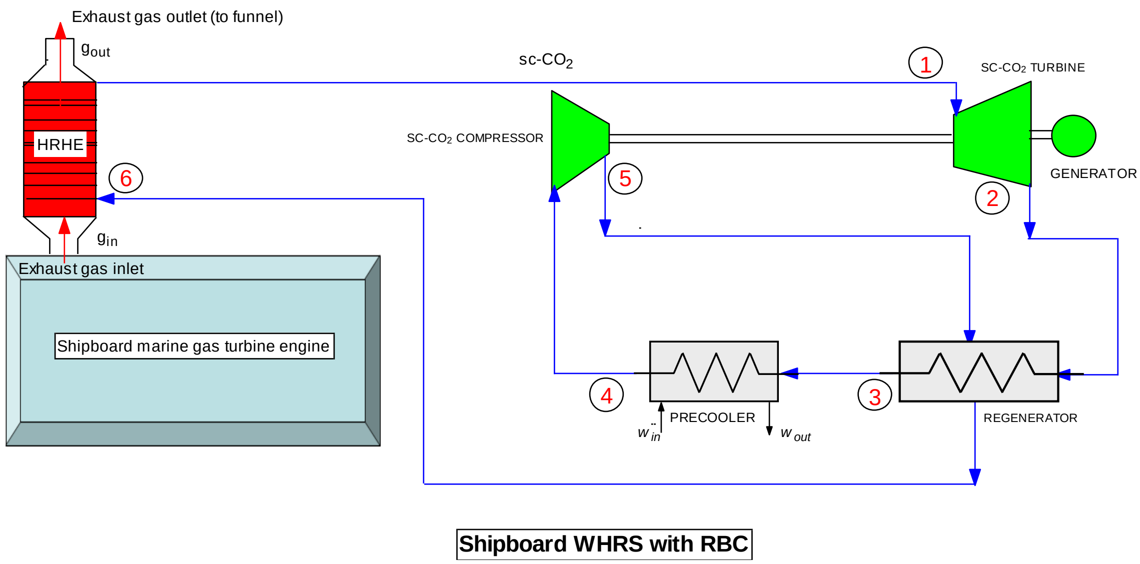

Figure 1: Schematic of proposed supercritical CO_2 RBC based WHRS for shipboard application.

The hot flue gases of the topping cycle (GE LM2500 aeroderivative gas turbine) enter the HRHE where the high enthalpy exhaust gases transfer heat to SC-CO_2 (working fluid) of bottoming cycle while the working fluid attains a superheated state (state point *1*). Thereafter, the superheated SC-CO_2 enters the turbine to deliver expansion work. The CO_2 at the turbine outlet enters the regenerator (hot stream) and transfers the heat to the cold stream coming from the compressor and, in turn cools down to state point *3*, as illustrated in Figure 1. In the heat sink (precooler), the hot stream from the regenerator is further cooled to state point *4* during process *3-4* before its inlet to the compressor. The compressor discharge (state point *5*) gets heated up to state point *6* in the regenerator and subsequently enters the heat source (HRHE) to again reach its superheated state, thereby completing the closed cycle. The specifications of shipboard GT considered for this study are taken from the published literature (Sharma *et al.*2017, GE website 2018).

2.1 Assumptions

A few assumptions for the thermodynamic modelling for *4E* analyses of the CO_2 based WHRS are as stated below:

- Steady-state steady-flow (SSSF) system.
- Adiabatic system with a single control volume for each component.
- Nil variation in the kinetic and the potential energies.
- Nil CO_2 related leakage in the system.
- Nil heat loss from system pipes to the environment or vice versa.
- Isentropic efficiency of the turbo machinery is constant.
- Source (exhaust gas in HRHE) and sink (heat rejection in precooler) are assumed to be non-isothermal.

- Exergy of the outgoing streams viz., flue gases from the HRHE and the cooling water from the precooler, are not accounted for since they are not utilised further.
- Temperature of flue gases exiting to the environment in HRHE is ensured to be above the dew point.
- The dead or reference temperature has been set as 35 ^{0}C instead of ISO conditions (15 ^{0}C), considering the “tropical/ extreme tropical” conditions.

2.2 Pressure Drop in Heat Exchangers

The pressure drop fraction, $\delta\bar{P}_{ij}$ ($\delta\bar{P}_{ij} = \Delta P / P_i$), where ΔP = ($P_i - P_j$), can be defined for working fluid stream ($i - j$), across various heat exchangers (Kotas 2012):

The isentropic efficiencies of the turbomachinery are defined as (Kotas 2012):

$$\eta_{isen,turbine} = \left(h_1 - h_2\right) / \left(h_1 - h_{2s}\right) \tag{1}$$

$$\eta_{isen,compressor} = \left(h_{5s} - h_4\right) / \left(h_5 - h_4\right) \tag{2}$$

Where h_{2s} and h_{5s} represent the enthalpy of CO_2 corresponding to the state point *2s* and *5s*, respectively while the suffix ‘*s*’ denotes the isentropic process.

The effectiveness of the regenerator and HRHE respectively can be expressed as (Dostal 2004):

$$\varepsilon_{regen} = \left(h_2 - h_3\right) / \left(h_2 - h_{5'}\right) \tag{3}$$

$$\varepsilon_{HRHE} = \left(h_{g,in} - h_{g,out}\right) / \left(h_{g,in} - h_{6'}\right) \tag{4}$$

where, $h_{5'}$ is the enthalpy at (T=T_5, P=P_3) while $h_{6'}$ is the enthalpy at (T=T_6, P=$P_{g,\,out}$). Further, the heat capacity of the hot streams (*2-3*) and (g_{in}-g_{out}), in Equations (3) and (4), respectively is assumed to be having the minimum value (C_{Min}).

2.3 First-Law Analysis

For the various heat-exchangers: regenerator, precooler and HRHE, respectively, following expressions based on energy balance can be written [3, 5, 41]:

$$h_2 - h_3 = h_6 - h_5 \tag{5}$$

$$\dot{m}_{CO_2}\left(h_3 - h_4\right) = \dot{m}_w\left(h_{w,out} - h_{w,in}\right) \tag{6}$$

$$\dot{m}_{CO_2}\left(h_1 - h_6\right) = \dot{m}_g\left(h_{g,in} - h_{g,out}\right) \tag{7}$$

where, $\dot{m}_{CO_2}$ represents CO_2 mass flow rate, while $\dot{m}_w$ represents the cooling water (seawater) mass flow rate in the Precooler. Further, the expressions based on heat transfer with source and sink (Kotas 2012) are:

$$\dot{Q}_{in,RBC} = \dot{m}_{CO_2}\left(h_1 - h_6\right) \tag{8}$$

$$\dot{Q}_{out,RBC} = \dot{m}_{CO_2}\left(h_3 - h_4\right) \tag{9}$$

where $\dot{Q}_{in,RBC}$ represents the heat transfer rate from exhaust gas (source)to SC-CO_2in HRHE during the heating process (*1-6*), and $\dot{Q}_{out,RBC}$ represents the heat transfer rate from SC-CO_2 to the cooling water (sink) in Precooler.

Defining maximum possible rate of energy that can be transferred from the flue gas stream (source) to working fluid of RBC (assuming the gas is cooled down till the ambient or dead state) as (Butcher and Reddy 2007):

$$\dot{Q}_{in,RBC,\max} = \dot{m}_g\left(h_{g,in} - h_{0,g}\right) \tag{10}$$

Where $h_{g,in}$ and $h_{0,g}$ represent enthalpies of flue gases at HRHE inlet and dead state respectively.

The expressions for work delivered by turbine and work consumed by compressor in the bottoming cycle can be respectively written as (Kotas 2012):

$$\dot{W}_{turbine} = \dot{m}_{CO_2}\left(h_1 - h_2\right) \tag{11}$$

$$\dot{W}_{compressor} = \dot{m}_{CO_2}\left(h_5 - h_4\right) \tag{12}$$

The net work produced by the bottoming cycle can be expressed as (Kotas 2012):

$$\dot{W}_{net,RBC} = \dot{W}_{turbine} - \dot{W}_{compressor} \tag{13}$$

The first-law efficiency of bottoming cycle ($\eta_{I,RBC}$), can be expressed as the ratio of useful output to input, in terms of the energy flow as (Cengel 2011 *et al.*)

$$\eta_{I,RBC} = \dot{W}_{net} / \dot{Q}_{in,RBC} \tag{14}$$

Similarly, for the combined cycle (TC+RBC), we can express the energy efficiency as (Kotas 2012)

$$\eta_{I,CC} = \left(\dot{W}_{net,TC} + \dot{W}_{net,RBC}\right) / \dot{Q}_{in,TC} \tag{15}$$

where, $\dot{Q}_{in,TC}$ is the rate of heat input (58.7 MW) (Kim *et al.* 2017) for the topping cycle.

Since the system involves heat recovery, an efficiency term called *heat recovery energetic efficiency* ($\eta_{I,HR}$) expressed as ratio of actual energy recovery to the maximum possible energy recovery from the gas turbine exhaust to the working fluid (Kim *et al.* 2017)

$$\eta_{I,HR} = \dot{Q}_{in,RBC} / \dot{Q}_{in,RBC,\max} \tag{16}$$

Further, to signify the energetic performance of WHRS, another term called as *system energetic efficiency* ($\eta_{I,sys}$) can be expressed as ratio of the net work produced and the maximum possible energy recovery rate from the gas turbine exhaust (Kim *et al.* 2017)

$$\eta_{I,sys} = \dot{W}_{net,RBC} / \dot{Q}_{in,RBC,\max} \tag{17}$$

Equations (16) and (17), when combined yield the following relationship:

$$\eta_{I,sys} = \eta_{I,RBC}\eta_{I,HR} \tag{18}$$

2.4 *Second-Law Analysis*

The general expression for exergy flow on specific mass basis is given as (Cengel 2011 *et al.*)

$$e_i = \left(h_i - h_0\right) - T_0\left(s_i - s_0\right) \tag{19}$$

The total exergy input rate from the hot exhaust gases, to the bottoming cycle can be expressed as:

$$\dot{E}_{in,RBC} = \dot{Q}_{in,RBC}\left(1 - T_0 / T_{g,avg}\right) \tag{20}$$

$T_{g,avg}$ represents the average heat source temperature evaluated in an entropic manner, as follows

$$T_{g,avg} = \left(h_{g,in} - h_{g,out}\right) \Big/ \left(s_{g,in} - s_{g,out}\right) \tag{21}$$

The *maximum possible* rate of exergy that can be transferred from the exhaust gas at state (g_{in}) into the WHRS can be defined as

$$\dot{E}_{in,RBC,\max} = \dot{m}_g e_{g,in} \tag{22}$$

Defining non-dimensional fraction for irreversibility rate (i_j) for each component of the cycle as ratio of exergy destruction rate across it and total incoming exergy rate, the expressions for various components of the bottoming cycle are given as (Cengel 2011 *et al.*)

$$i_{turbine} = \left\{\dot{m}_{CO_2}\left(e_1 - e_2\right) - \dot{W}_{turbine}\right\} / \dot{E}_{in,RBC} \tag{23}$$

$$i_{compressor} = \left\{\dot{m}_{CO_2}\left(e_4 - e_5\right) + \dot{W}_{compressor}\right\} / \dot{E}_{in,RBC} \tag{24}$$

$$i_{regen} = \left[\dot{m}_{CO_2}\left\{\left(e_2 - e_3\right) + \left(e_5 - e_6\right)\right\}\right] / \dot{E}_{in,RBC} \tag{25}$$

$$i_{precooler} = \left\{\dot{m}_{CO_2}\left(e_3 - e_4\right)\right\} / \dot{E}_{in,RBC} \tag{26}$$

$$i_{HRHE} = \left\{ \dot{E}_{in,RBC} + \dot{m}_{CO_2} \left(e_6 - e_1 \right) \right\} / \dot{E}_{in,RBC} \quad (27)$$

Thus, the exergy efficiency ($\eta_{II,RBC}$), which is the key indicator of the exergetic performance can be expressed, for the proposed RBC and the combined cycle (CC) as (Kotas 2012, Cengel 2011 *et al.*)

$$\eta_{II,RBC} = \dot{W}_{net} / \dot{E}_{in,RBC} = 1 - \sum_{j=1}^{6} i_{j,RBC} \quad (28)$$

$$\eta_{II,CC} = \left(\dot{E}_{out,TC} + \dot{E}_{out,RBC} \right) \Big/ \left[\dot{Q}_{in,TC} \left(1 - T_0 / T_H \right) \right] \quad (29)$$

where $\dot{E}_{out,TC}$ and $\dot{E}_{out,RBC}$ represent the rate of exergy output associated with the TC and the RBC respectively, while $\dot{Q}_{in,TC}$ and T_H represent the rate of energy input and the source temperature for the TC respectively.

Whereas, exergy-based performance indicators such as heat-recovery-exergetic efficiency ($\eta_{II,HR}$) and system-exergetic efficiency ($\eta_{II,sys}$), are expressed as (Kim *et al.* 2017)

$$\eta_{II,HR} = \dot{E}_{in,BC} / \dot{E}_{in,RBC,\max} \quad (30)$$

$$\eta_{II,sys} = \dot{W}_{net,RBC} / \dot{E}_{in,RBC,\max} \quad (31)$$

Equations (30) and (31) when combined yield the following relationship:

$$\eta_{II,sys} = \eta_{II,RBC} \eta_{II,HR} \quad (32)$$

2.5 Economic Analysis

The key economic performance parameters are the annual fuel savings and the simple payback period. The associated assumptions simplifying the model are as follows (Moran and Shapiro 2010)

- Ship's average operation-at-sea of 20 days per month.
- One out of the two marine gas turbines (LM2500) are always connected to meet ship's propulsion power.
- Two SC-CO_2WHRS (one for each gas turbine) are employed to augment the ship's propulsion power.
- Each SC-CO_2WHRS has the designed capacity of 5000 [kW].
- Average specific fuel consumption of the shipboard gas turbine is approx. 250 [g/kWh] of the GT fuel.
- Average ship speed is approximately 16 Knots (nautical miles per hour).
- The cost and the density of the marine grade fuel is around USD 0.95/litre and 830 kg/m^3 respectively.
- The installed cost for the typical SC-CO_2 WHRS is around USD 2200 per kW of the power output (Kacludis *et al.* 2012).

The annual fuel savings, on mass basis ($\dot{m}_{f,s\_RBC}$) can be estimated as the total mass of the fuel saved by the WHRS, [ton/yr] and can be expressed as follows:

$$\dot{m}_{f,s\_RBC} = (SFC) \dot{W}_{net\_RBC} t_{op} \Big/ 10^6 \quad (33)$$

The ship's fuel consumption in [ton/nm], at the average ship speed of V_s [Knots] is given as:

$$\dot{m}_{f,nm\_RBC} = \left((SFC) \dot{W}_{net\_TC} \right) \Big/ \left(10^6 V_s \right) \quad (34)$$

where, SFC represents the average specific consumption of fuel (g/kWh) of the topping cycle plant, $\dot{W}_{net,RBC}$ represents the network output (kW) from the RBC, while t_{op} is the annual running hours clocked on each gas turbine.

The net increase in the range (distance covered in nm) and the endurance (days-at-sea) per year accruable by the ship with the SC-CO_2 WHRS are respectively given as:

$$E_{d\_nm} = \dot{m}_{f,s\_RBC} \Big/ \dot{m}_{f,nm\_RBC} \quad (35)$$

$$E_{d\_x} = \left(\dot{m}_{f,s\_RBC} 10^6 \right) \Big/ \left((SFC) \dot{W}_{net\_TC} 24 \right) \quad (36)$$

where, E_{d_nm} and E_{d_x} represent the net increase in the annual range i.e., the distance (in nm) covered by the ship and the endurance (no. of days-at-sea), respectively.

The annual fuel savings ($\dot{C}_{f,s}$) in [USD/year] can be defined as the cost of fuel savings accrued by the ship per year and is given by

$$\dot{C}_{f,s\_RBC} = \left(\dot{m}_{f,s\_RBC} C_f / \rho_f\right) \times 1000 \tag{37}$$

where, C_f and ρ_f are the total cost and the density of fuel consumed by both GTs. The simple payback period (SPP_y) without incentives, in no. of years for the SC-CO_2 WHRS can be estimated as:

$$SPP_{y\_RBC} = C_{I\_RBC} \Big/ \dot{C}_{f.s\_RBC} \tag{38}$$

where, C_{I_RBC} is the total installed cost of the SC-CO_2WHRS in USD.

2.6 Environmental Performance

The amount of CO_2 emissions avoided and the potential carbon credits earned by the ship owing to installation of the SC-CO_2 WHRS is estimated. The associated assumptions are (Kacludis *et al.* 2012)

- GT fuel (LSHSD) approximated as pure cetane (*n-dodecane*), chemical formula: $C_{16}H_{34}$
- Emission factor: the mass of [CO_2] emitted per unit power produced by the shipboard GT using $C_{16}H_{34}$ as fuel, is calculated as 0.246 kg [CO_2]/kWh
- Potential Carbon Credits earnings per ton of [CO_2] emission avoided is around \$15(Wright*et al.* 2016)

The annual [CO_2] emissions avoided ($\dot{m}_{[CO_2],RBC}$) on mass basis in [ton/yr] can be expressed as

$$\dot{m}_{[CO_2],RBC} = \left(EF\right)_f \dot{W}_{net,RBC} t_{op} \Big/ 1000 \tag{39}$$

Where $\left(EF\right)_f$ is emission factor of fuel (kg [CO_2]/kWh), calculated as 0.246 [kg [CO_2]/kWh] for a typical GT marine diesel fuel. Accordingly, the annual Carbon Credits earned (\$) annually by the ship with SC-CO_2 WHRS is estimated.

Table 1 Input parameters for supercritical carbon dioxide RBC

Parameter	**Set value**
Load(% of rated power) on topping gas turbine	100% to 40%
Flue gas composition	Ideal gas air
Flue gas mass flow rate	63.6 kgs^{-1}
Flue gas inlet temperature	572 ^oC
Flue gas inlet pressure	0.1084 MPa
RBC turbine inlet temperature	550 ^oC
RBC compressor inlet temperature	40 ^oC
RBC compressor discharge pressure	20 MPa
RBC pressure ratio (r_p) range	2.0 to 4.0
PPTD in HRHE (hot side)	22
PPTD in precooler (hot side)	15
Cooling water (seawater) inlet temperature	30 ^0C
Pressure drop in heat exchangers (%)	Nil
RBC turbine isentropic efficiency (%)	90
RBC compressor isentropic efficiency (%)	85
Regenerator effectiveness (%)	86
HRHE effectiveness (%)	90

2.7 *Simulation and Input Parameters*

The thermodynamic analyses are undertaken by theoretically simulating the in-house developed mathematical models using Engineering Equation Solver (EES) software (Klein 2008) which uses Span and Wagner's equation of state (Span and Wagner1996) to determine thermodynamic properties of CO_2 at various state points. The models search for various state points across the supercritical Carbon Dioxide RBC (bottoming cycle) using a set of input parameters defined for proposed supercritical Carbon Dioxide based WHRS for marine gas turbines, as shown in Table 1. The assumed set values are based on earlier studies on SC-CO_2 Brayton cycles in the published literature (Sharma *et al.* 2017). Topping cycle loads of 100% to 40% load have been considered since onboard naval ships, gas turbines operate at 100% power for only around 1-2% of the entire life, while because of the parabolic nature of the propeller (load) curve, fleet speeds between 12 to 16 knots are achieved with GTs running around 40 % (8.8 MW) or lesser of their rated power.

2.8 *Model Validation*

The thermodynamic model for proposed supercritical CO2 based bottoming cycle is validated with the published literature as shown in Table 2, which shows reasonable agreement.

Table 2 Validation of Thermodynamic Model

Design parameter	**(Dostal *et al.* 2004)**	**(Neises and Turchi 2014)**	**(Zhao *et al.* 2016)**
Turbine inlet temperature	550 ^{0}C	650 ^{0}C	550 ^{0}C
Compressor inlet temperature	32 ^{0}C	50 ^{0}C	32 ^{0}C
Cooling water inlet temperature	27 ^{0}C	35 ^{0}C	15 ^{0}C
Compressor discharge pressure	20 MPa	25 MPa	20 MPa
Compressor inlet pressure	7.4 MPa	7.4 MPa	7.7 MPa
RBC Pressure ratio	2.7	3.4	2.6
Compressor efficiency	89%	89%	89%
Turbine efficiency	90%	93%	90%
Regenerator effectiveness	97.5%	97%	90%
Pressure drop in heat exchangers	-	-	-
Published literature results	η_I = 39.5%*	η_I = 44.6%	η_{II} = 55.0%
Present model results	η_I = 39.5%	η_I = 43.1%	η_{II} = 56.0%
Deviation (if any)	-	(-)3%	(+)2%

**As per conservative design of the turbomachinery (Dostal et al. 2004) and 100% Load*

2.9 *Optimisation of RBC*

The optimisation of the cycle is undertaken based on the optimum value of its pressure ratio (r_p) corresponding to the highest exergetic performance. Variation of r_p was done in increments of 0.01 bar having upper and lower bounds of 40 and 20 bars respectively keeping the maximum pressure (P_{max}) constant at 200 bar, since high operating pressures would add to the operating cost and design complexity of the system.

3. Results and Discussion

After validation, the RBC based WHRS model was simulated by varying key design variables considering the input parameters as defined in Table 1 and various assumptions mentioned in section 2, for detailed *4E* performance analyses. The salient results obtained along with their physical significance are discussed below.

3.1 *Exergy Balance*

In order to account for the exergy, and the exergy destruction or the irreversibility associated within the cycle, the SC-CO_2 bottoming cycle has been plotted on the *T-s* and *T-e* planes as shown below in Figures 2 and 3 respectively, while the exergy balance sheet indicating various specific exergy flows across the cycle and its components is presented below in Table 3. It is seen from Figures. 2 and 3 and Table 3, that the total irreversibility rate across the heat-exchangers is substantially higher than that across the turbomachinery. It is found that the precooler, the HRHE and the regenerator altogether contribute more than 80% of the total irreversibility rate in the cycle, due to greater heat transfer temperature difference. It is also inferred from these results that the most critical system components for design of a modular WHRS for shipboard applications, from the exergy destruction perspective are the precooler, the HRHE and the regenerator.

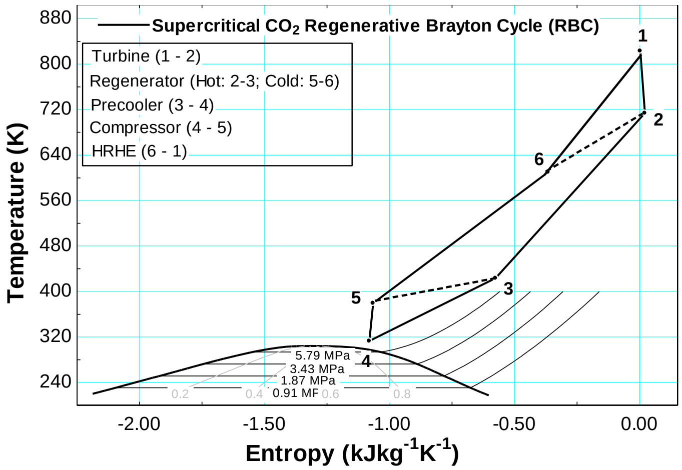

Figure 2: SC-CO_2 RBC on temperature-entropy diagram

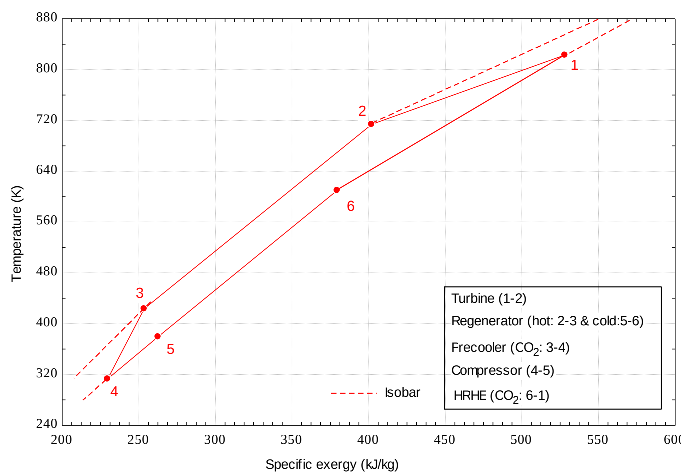

Figure 3: SC-CO_2 RBC on temperature-specific exergy diagram

Table 3 Specific exergy balance for SC-CO_2 based WHRS*

Exergy	**$kJkg^{-1}$**	**% (of exergy inlet)**	**Exergy destruction**	**$kJkg^{-1}$**	**% (of exergy inlet)**	**% (of total exergy destruction**
Exergy inlet to the cycle (RBC)	168.9	100.0	HRHE	18.3	10.9	22.3
			Precooler	31.9	19.0	39.0
			Regenerator	16.5	9.8	20.2
Exergy recovered through Turbine	152.7		Turbine	7.7	4.6	9.5
Exergy input to Compressor	66.6		Compressor	7.4	4.4	9.0
Net exergy output	**86.1**	**51.3**	**Total**	**81.8**	**48.7**	**100.0**

*$r_p = 3.3$

3.2 Influence of Critical Point on Thermo-Physical Properties of CO2

Figure 4 illustrates the behaviour of the CO_2 specific heat, in proximity to its critical point and at the key temperature and pressure conditions of interest for the proposed WHRS. It is seen that CO_2 exhibits sharp fluctuations or variations of the specific heat in the critical point region while these variations rapidly diminish away when the temperature and pressure conditions have moved sufficiently far away from this region. This unique trend of CO_2 bears great impact on its thermo-physical and transport properties, for example, CO_2 has a very high density (around 10.62 mol/l or 467 kg/m^3) near the critical point. This feature of the SC-CO_2 systems could be advantageous in realising highly power dense and efficiently controlled energy recovery and power generation systems as compared to the conventional Brayton/ Rankine systems (Dostal *et al.* 2004), making them very suitable for the shipboard applications.

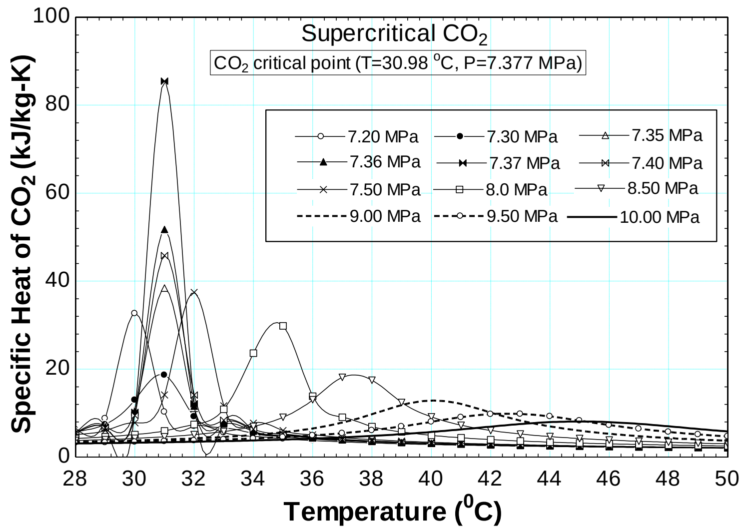

Figure 4: Variation of specific heat of CO_2 near critical point and at key (T, P) conditions

3.3 Optimization of Cycle Pressure Ratio (rp)

The influence of pressure ratio (r_p) on efficiency and power or work output of the proposed RBC assuming the constant turbine entry conditions is shown in Figure 5. As evident from Figure 5, both efficiency and power output vary significantly with r_p. At the low pressure ratios, both cycle efficiency and work output are found to be increasing albeit at different rates, with the rise in the pressure ratio, up to their respective optimum pressure ratio value. The cycle efficiency reaches its maxima at a lower value of the optimum pressure ratio than the work output. After reaching the respective maxima, both the cycle efficiency and the power output decrease albeit at different rates, with further increase in the cycle pressure ratio beyond its respective optimum pressure ratio. Further, it is also seen that, prior to attaining the respective maxima, the rate of increase in the efficiency is higher than the net work output, while, after attaining the maxima, the rate of decrease in the cycle efficiency is significantly higher than the net work output.

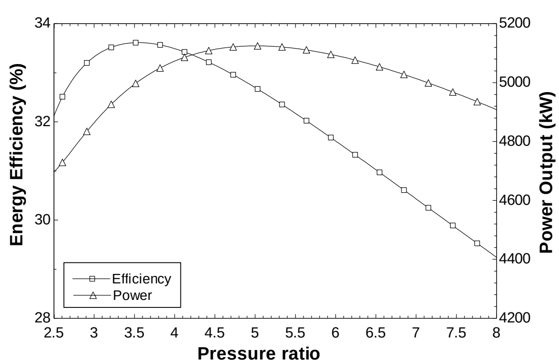

Figure5: Effect of pressure ratio on efficiency and power output

3.4 Influence of Minimum Cycle Temperature

Figure 6 presents the influence that the minimum cycle temperature exerts on the cycle efficiency and the optimum pressure ratio for RBC. Figure 6 indicates that the cycle efficiency as well as the optimal pressure ratio exhibit a drop (almost linear) with increasing compressor inlet temperature values from 32^0C to 50^0C. The efficiency shows a linear drop with the minimum cycle temperature because of its influence on the temperature difference or the temperature lift of RBC. The lift decreases with rise in the minimum cycle temperature, and thus causes a linear decrease in the efficiency. It is found that an increase in the compressor inlet temperature from 32-50^0C results in about 7% decrease in the cycle efficiency.

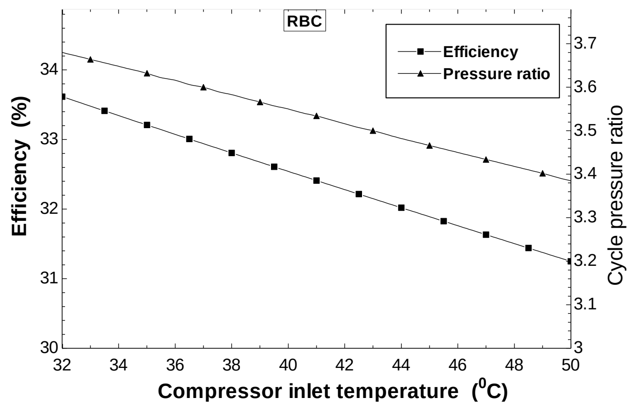

Figure 6: Influence of min. cycle temperature on efficiency and optimum-pressure-ratio

Figure 7 shows the influence of the minimum cycle temperature on exergy destruction rate across cycle components, at the corresponding optimum-pressure-ratios. It is seen that the effect of minimum cycle temperature on the exergy destruction rate is most pronounced (about 30% increase) for the precooler, while it is found to be least pronounced for the turbine (nil change) and the compressor (11% increase). Further, it is seen that, with the increase in the compressor inlet temperature, the exergy destruction rate in the precooler increases significantly.

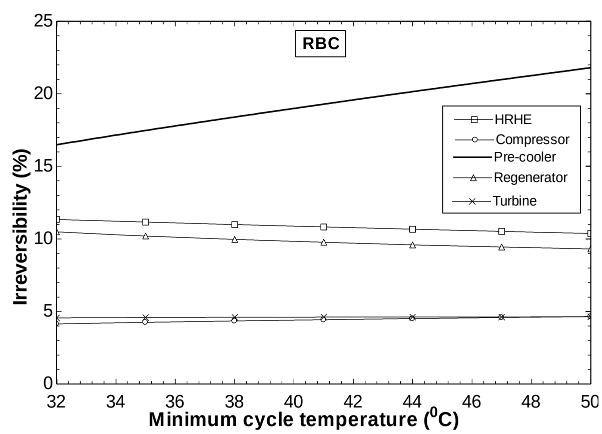

Figure7: Influence of min. cycle temperature on exergy destruction in components

3.5 Effects of Maximum Cycle Temperature

Figures 8 and 9 show the effects of the maximum cycle temperature (also, TIT) on the critical performance indicators of the SC-CO_2 regenerative Brayton cycle (RBC) system, considering input parameters stated earlier. Figure 8 indicates $r_{p,\,opt}$ shows almost a direct relationship with TIT. This trend contrasts with the trend observed earlier (section 3.4). The physical significance of this trend is attributable to the fact that change in the optimum value of pressure ratio depends on specific heat variation of CO_2 (Dostal *et al.* 2004). The inlet conditions across the turbine unlike the compressor are located sufficiently far with respect to the critical point and hence involve minimal or negligible specific heat variation of CO_2 leading to the lower variation in the optimum cycle pressure ratio, due to variation in the TIT. It is further seen from Figure. 8 that both efficiencies improve with the surge in the TIT. This particular trend is expected since, with the increase in the TIT, the temperature-lift of the cycle increases, leading to increase in the underlying thermodynamic efficiency of a Brayton cycle. In other words, with the increase in TIT, the energy quality and hence, the availability per unit time per cycle increases, resulting in an increase in the exergy efficiency. It is seen that an increase in the TIT from 450-550 ^{0}C, resulted in efficiency improvement of about 13 %.

Figure 9 presents the effects due to TIT on irreversibility of different components of the RBC. It is seen from Figure. 9 that the influence of TIT on the irreversibility is most pronounced for the HRHE (about 26% decrease) followed by the precooler (about 19% increase) and the regenerator (about 19% increase), while it is found to be the least pronounced for the turbine (about 4% decrease) followed by the compressor (about 10% increase).

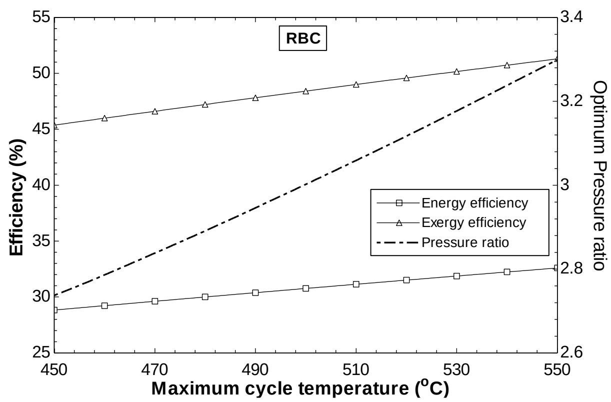

Figure 8: Influence of max. cycle temperature on efficiency and optimum-pressure-ratio

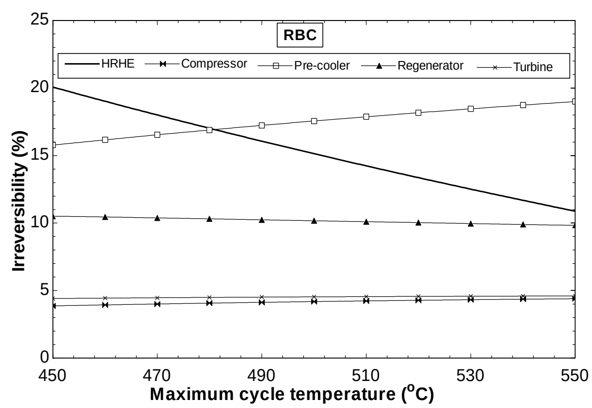

Figure 9: Influence of max. cycle temperature on irreversibility across components

3.6 Influence of Maximum Cycle Temperature (T1) on System and WHR Efficiency

Figure 10 illustrates the influence on RBC, system and WHR efficiencies ($\eta_{I,RBC}, \eta_{I,sys}$ and $\eta_{I,WHR}$) for a maximum cycle temperature (T_1) range (372-552 ^{0}C), by varying the PPTD across the HRHE. Figure 10 indicates that, cycle efficiency shows almost direct relationship with T_1. This particular trend is expected since increases in the turbine inlet temperature (*TIT*) increases both the availability per unit time per cycle and the temperature-lift, resulting in improved cycle efficiency.

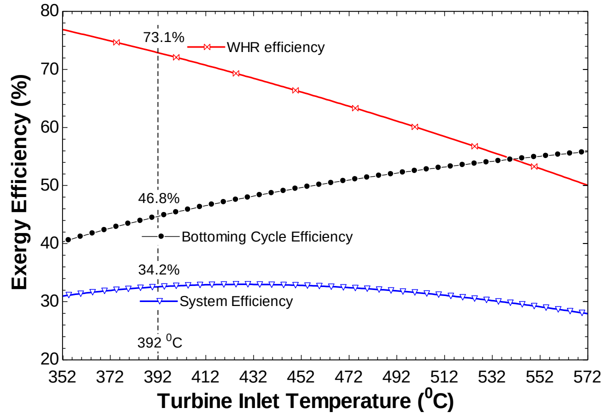

Figure 10: Influence of max. cycle temperature (T_1) on system and WHR efficiency

Whereas, as seen from Figure 10, the WHR efficiency shows a reverse trend i.e., it significantly decreases with the rise in the TIT. This particular trend is attributable to the fact that, with an increase in *TIT* the preheating of CO_2 across the regenerator improves, which in turn, adversely affects the energy transfer rate across the HRHE. It is also found from the simulation results that the system efficiency reaches maxima at a *TIT* of around 392^{0}C (665 K), where $\eta_{II,BC} = 46.8\%, \eta_{II,sys} = 34.2\%$, and $\eta_{I,WHR} = 58\%$.

3.7 Energy-Exergy-Economic-Environment (4E) Performance

The energy, exergy, economic & environment (*4E*) performance analyses results for SC-CO_2 RBC under various operating load conditions onboard a typical frigate is presented in Table 4.

4. Conclusions

In this article, the authors have attempted to integrate a compact supercritical CO_2 regenerative Brayton cycle with a shipboard GT (LM2500: topping cycle) for efficient and effective energy recovery from its exhaust to improve the overall plant efficiency and produce additional power for the ship. The *4E* performance of the novel energy recovery system has been investigated for a shipboard platform. The following are the salient conclusions: -

- The most critical components for design of modular SC-CO_2 based WHRS for shipboard application, from the exergy destruction perspective are the heat exchangers namely, precooler, HRHE and regenerator.
- The cycle performance is less sensitive to maximum cycle temperature than minimum cycle temperature.
- Substantial improvement (about 11 %) in overall efficiency of the shipboard GT plant is feasible through the integration of shipboard GT with proposed WHRS.
- With the proposed WHRS, augmentation of shipboard shaft power of around 4.8 MW, equivalent to 23.3 % of the GT rated power, without any additional fuel, is found to be feasible at 100% load condition.
- It is seen that, at maximum relative GT load operation, the ship can accrue extra range of around 22565 nm per year and endurance of around 60 days-at-sea per year.
- The annual carbon emissions savings varies from 5821 (ton-[CO_2]/year) at 100% relative GT load to 4100 (ton-[CO_2]/year) at 60% relative GT load.
- The annual carbon credits earnings vary from USD 87313 at 100% relative GT load to about USD 61501 at 60% relative gas turbine load.
- The simple payback period for the system is found to vary from 3.1 years at 100% relative GT load to 3.8 years at 60% relative GT load.

The future-ready warship designs support the multi-mission roles and the intense high power electric weapon systems which warrant effective & dependable propulsion technologies with high power density such as Gas Turbines and Fully/ Integrated Electric propulsion & power generation systems. Integrating these with such compact novel WHR systems ensures significant improvement in ship's overall energy efficiency design index including gas turbine's part-load performance resulting in an additional endurance, range, and proportionate reduction in the fleet operating cost. In addition, superior environmental performance also ensures sustainability and ship's compliance to the stringent IMO norms of the future.

Table 4 4E performance results

Relative Load / Performance Parameter	**(%)**	**100 (*)**	**87**	**73**	**60**
$r_{p,opt}$	(-)	3.12	3.03	2.94	2.83
UA	(kJ/K)	161.1	155.0	147.7	139.3
$\eta_{I,RBC}$	(%)	31.6	30.9	30.3	29.5
$\eta_{II,RBC}$	(%)	54.4	54.2	53.9	53.6
$\eta_{I,TC}$	(%)	33.6	32.5	31.1	29.5
$\eta_{II,TC}$	(%)	45.8	44.2	42.3	40.1
$\eta_{I,CC}$	(%)	41.9	40.7	39.4	37.9
$\eta_{II,CC}$	(%)	56.9	55.4	53.6	51.6
Net Work Output	(kW)	4108	3707	3300	2894
Fuel Savings	(USD, Million/year)	7.15	6.69	6.22	5.75
Extra Range	(nm/yr)	22565	23494	24720	26490
Extra Endurance	(days/yr)	58.8	61.2	64.4	69.0
Simple Payback Period	Yr	3.1	3.3	3.5	3.8
Carbon Footprints (Savings)	(ton-[CO_2]/year)	5821	5252	4676	4100
Carbon Credits (Earnings)	(USD/year)	87313	78786	70145	61501

**Assuming design point as relative GT load, corresponding to about 80% shipboard GT rated power (16.8 MW).*

5. *Acknowledgements*

This paper is a result of author's research at IIT Delhi for the award of PhD degree and follow-on of the paper which was presented at INEC 2017 in Singapore. The authors are thankful to the valuable inputs from GE and Department of Energy Science and Engineering, IIT Delhi during the course of research work.

Nomenclature

A	area (m^2)	*Greek letters*	
BC	bottoming cycle	ε	effectiveness (%)
C	cost ($)	η	efficiency (%)
$\dot{C}$	cost per year	ΔP	pressure drop (kPa)
C_p	isobaric specific heat ($kJ.kg^{-1}.K^{-1}$)	ΔT_{lm}	LMTD (K)
CO_2	Carbon Dioxide	$\delta\bar{P}_{ij}$	relative pressure drop fraction
e	specific exergy($kJ.kg^{-1}$)	ρ	density
E	endurance of ship (nm or days-at-sea)	*Subscripts and superscripts*	
$\dot{E}$	exergy rate (kW)	avg	average
$(EF)_f$	emission factor of fuel (kg- [CO_2]$kW^{-1}h^{-1}$)	CC	combined cycle (TC+RBC)
FTT	finite-time thermodynamics	CO_2	carbon dioxide (working fluid)
GT	gas turbine	d	design condition
h	specific enthalpy ($kJ.kg^{-1}$)	f	fuel
HRHE	heat recovery heat exchanger	fs	fuel savings

i_j	non-dimensional irreversibility fraction for the j^{th} component	g	exhaust gas
IRSS	Infra-Red-Suppression-System	GT	gas turbine
$\dot{I}$	rate of irreversibility or exergy destruction (kW)	H	heat source temperature
IMO	International Maritime Organisation		
load	operational load (% of rated power)	HR	heat recovery efficiency
LSHSD	low-sulphur-high-speed diesel	$HRHE$	heat-recovery-heat-exchanger
$\dot{m}$	flow rate (kg.s^{-1})	I	energy efficiency; installed (cost)
P	pressure (MPa)	II	exergy efficiency
PPTD	pinch point temperature difference (^{0}C or K)	$isen$	isentropic
$\dot{Q}$	rate of energy exchange (kW)	i	state points (*1 - 6*)
r_p	pressure ratio ($p_{\max}/p_{\min}$)	in	inlet
s	specific entropy (kJ.kg^{-1}.K^{-1})	j	component
sc	super-critical	nm	range or distance in nautical mile
SFC	specific-fuel-consumption (g.kW^{-1}h^{-1})	$regen$	regenerator
SPP	simple payback period	RBC	regenerative Brayton (bottoming) cycle
t	time (h)	RH	relative humidity (%)
T	temperature (K)	RL	relative load (%)
TC	topping cycle	net	net or resultant value
TIT	turbine inlet temperature (^{0}C)	op	operating or running
U	heat transfer coefficient (Jm^{-2}K^{-1})	opt	optimum value
$\dot{W}$	shaft or mechanical work or power, kW	out	outlet
WG	water-gauge (pressure) (inch)	s	ideal or isentropic
WHR	waste-heat-recovery	sys	system
WHRS	waste-heat-recovery-system	TC	topping cycle
4E	energy, exergy, economic and environmental (analyses)	w	water
		$WHRS$	waste heat recovery system
		x	no. of days-at-sea
		y	no. of years
		$1,2,...6$	state points
		o	dead or reference state

References

Ahn, Y., Bae, S.J., Kim, M., Cho, S.K., Baik, S., Lee, J.I. and Cha, J.E., 2015. Review of supercritical CO2 power cycle technology and current status of research and development. *Nuclear engineering and technology*, *47*(6), pp.647-661.

Akbari, A.D. and Mahmoudi, S.M.S., 2017. Thermo-economic performance and optimization of a novel cogeneration system using carbon dioxide as working fluid. *Energy conversion and management*, *145*, pp.265-277.

Angelino, G., 1969. *Real gas effects in carbon dioxide cycles* (Vol. 79832, p. V001T01A071). American Society of Mechanical Engineers.

Baldi, F. and Gabrielii, C., 2015. A feasibility analysis of waste heat recovery systems for marine applications. *Energy*, *80*, pp.654-665.

Bejan, A., 2002. Fundamentals of exergy analysis, entropy generation minimization, and the generation of flow architecture. *International journal of energy research*, *26*(7).

Butcher, C.J. and Reddy, B.V., 2007. Second law analysis of a waste heat recovery based power generation system. *International Journal of heat and mass transfer*, *50*(11-12), pp.2355-2363.

Cengel, Y.A., Boles, M.A. and Kanoğlu, M., 2011. *Thermodynamics: an engineering approach* (Vol. 5, p. 445). New York: McGraw-hill.

Cheng, W.L., Huang, W.X. and Nian, Y.L., 2017. Global parameter optimization and criterion formula of supercritical carbon dioxide Brayton cycle with recompression. *Energy Conversion and Management*, *150*, pp.669-677.

Crespi, F., Gavagnin, G., Sánchez, D. and Martínez, G.S., 2017. Supercritical carbon dioxide cycles for power generation: A review. *Applied energy*, *195*, pp.152-183.

Dinçer, İ. and Zamfirescu, C., 2014. *Advanced power generation systems*. Academic Press.

Dostal, V., Driscoll, M.J. and Hejzlar, P., 2004. A supercritical carbon dioxide cycle for next generation nuclear reactors.

Feher, E.G., 1968. The supercritical thermodynamic power cycle. *Energy conversion*, *8*(2), pp.85-90.

Frost, T.H., Anderson, A. and Agnew, B., 1997. A hybrid gas turbine cycle (Brayton/Ericsson): an alternative to conventional combined gas and steam turbine power plant. *Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy*, *211*(2), pp.121-131.

GE marine gas turbines, 2018. http://www.geaviation.com/marine/engines/ military/lm2500 [accessed on 07 Mar, 2018].

Göktun, S. and Yavuz, H., 1999. Thermal efficiency of a regenerative Brayton cycle with isothermal heat addition. *Energy Conversion and Management*, *40*(12), pp.1259-1266.

Guoqiang, Z., Jiongzhi, Z. and Angjun, X., 2016. Thermodynamic analysis of combined cycle under design/off-design conditions for its efficient design and operation. *Energy Conversion and Management*, *126*, pp.76-88.

Harrington, R.L. ed., 1992. *Marine engineering*. Society of Naval Architects &Marine Engineers.

Herrmann, S., Kahlert, S., Wuerth, M. and Spliethoff, H., 2017. Thermo-Economic Evaluation of Novel Flexible CAES/CCPP Concept. *Journal of Energy Resources Technology*, *139*(1).

Hou, S., Wu, Y., Zhou, Y. and Yu, L., 2017. Performance analysis of the combined supercritical CO2 recompression and regenerative cycle used in waste heat recovery of marine gas turbine. *Energy Conversion and Management*, *151*, pp.73-85.

Kacludis, A., Lyons, S., Nadav, D. and Zdankiewicz, E., 2012. Waste heat to power (WH2P) applications using a supercritical CO2-based power cycle. *Power-Gen International*, *2012*, pp.11-13.

Kaushik, S.C., Tyagi, S.K. and Singhal, M.K., 2003. Parametric study of an irreversible regenerative Brayton cycle with isothermal heat addition. *Energy Conversion and Management*, *44*(12), pp.2013-2025.

Kaushik, S.C., Tyagi, S.K. and Kumar, P., 2017. *Finite time thermodynamics of power and refrigeration cycles*. Springer International Publishing.

Khadse, A., Blanchette, L., Kapat, J., Vasu, S., Hossain, J. and Donazzolo, A., 2018. Optimization of supercritical CO2 Brayton cycle for simple cycle gas turbines exhaust heat recovery using genetic algorithm. *Journal of energy resources technology*, *140*(7).

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

KleinS.A., 2008. EES, *Engineering Equation Solver*, version 8.158, F Chart Software, Middleton, WI.

Kotas, T.J., 2012. *The exergy method of thermal plant analysis*. Paragon Publishing.

Manjunath, K., Sharma, O.P., Tyagi, S.K. and Kaushik, S.C., 2018. Thermodynamic analysis of a supercritical/transcritical CO2 based waste heat recovery cycle for shipboard power and cooling applications. *Energy conversion and Management*, *155*, pp.262-275.

Moran, M.J., Shapiro, H.N., Boettner, D.D. and Bailey, M.B., 2010. *Fundamentals of engineering thermodynamics*. John Wiley & Sons.

Moroz, L., Burlaka, M., Rudenko, O. and Joly, C., 2015, November. Evaluation of gas turbine exhaust heat recovery utilizing composite supercritical CO2 cycle. In *Proceedings of the International Gas Turbine Congress, Tokyo, Japan* (pp. 15-20).

Neises, T. and Turchi, C., 2014. A comparison of supercritical carbon dioxide power cycle configurations with an emphasis on CSP applications. *Energy Procedia*, *49*, pp.1187-1196.

Nord, L.O. and Bolland, O., 2013. Design and off-design simulations of combined cycles for offshore oil and gas installations. *Applied Thermal Engineering*, *54*(1), pp.85-91.

Persichilli, M., Kacludis, A., Zdankiewicz, E. and Held, T., 2012. Supercritical CO2 power cycle developments and commercialization: why sCO2 can displace steam? *Power-Gen India & Central Asia*, *2012*, pp.19-21.

Pierobon, L., Benato, A., Scolari, E., Haglind, F. and Stoppato, A., 2014. Waste heat recovery technologies for offshore platforms. *Applied Energy*, *136*, pp.228-241.

Sarkar, J., 2009. Second law analysis of supercritical CO2 recompression Brayton cycle. *Energy*, *34*(9), pp.1172-1178.

Sarkar, J., 2018. A novel pinch point design methodology-based energy and economic analyses of organic Rankine cycle. *Journal of Energy Resources Technology*, *140*(5).

Sharma, O.P., Kaushik, S.C. and Manjunath, K., 2017. Thermodynamic analysis and optimization of a supercritical CO2 regenerative recompression Brayton cycle coupled with a marine gas turbine for shipboard waste heat recovery. *Thermal Science and Engineering Progress*, *3*, pp.62-74.

Singh, D.V. and Pedersen, E., 2016. A review of waste heat recovery technologies for maritime applications. *Energy conversion and management*, *111*, pp.315-328.

Span, R. and Wagner, W., 1996. A new equation of state for carbon dioxide covering the fluid region from the triple-point temperature to 1100 K at pressures up to 800 MPa. *Journal of physical and chemical reference data*, *25*(6), pp.1509-1596.

Tyagi, S.K., Chen, J. and Kaushik, S.C., 2005. Optimal criteria based on the ecological function of an irreversible intercooled regenerative modified Brayton cycle. *International Journal of Exergy*, *2*(1), pp.90-107.

Tyagi, S.K., Chen, G.M., Wang, Q. and Kaushik, S.C., 2006. Thermodynamic analysis and parametric study of an irreversible regenerative-intercooled-reheat Brayton cycle. *International Journal of Thermal Sciences*, *45*(8), pp.829-840.

Tyagi, S.K., Wang, S. and Park, S.R., 2008. Performance criteria on different pressure ratios of an irreversible modified complex Brayton cycle.

Walnum, H.T., Nekså, P., Nord, L.O. and Andresen, T., 2013. Modelling and simulation of CO2 (carbon dioxide) bottoming cycles for offshore oil and gas installations at design and off-design conditions. *Energy*, *59*, pp.513-520.

Wright, S.A., Davidson, C.S. and Scammell, W.O., 2016, March. Thermo-economic analysis of four sCO2 waste heat recovery power systems. In *Fifth International SCO2 Symposium, San Antonio, TX, Mar* (pp. 28-31).

Zhao, H., Deng, Q., Huang, W., Wang, D. and Feng, Z., 2016. Thermodynamic and economic analysis and multi-objective optimization of supercritical CO2 Brayton cycles. *Journal of Engineering for Gas Turbines and Power*, *138*(8).