# Application of Supercritical $CO_2$ Cycle for Waste Heat Recovery in LNG Carriers

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**Abstract.** Nowadays, decarbonization of the shipping industry has become the top priority of the maritime community. In an effort to reduce emissions from shipping, numerous technological and design solutions are being investigated; Waste Heat Recovery (WHR) from marine engines is one of the most important and widespread ones. This paper investigates the utilization of a carbon dioxide Supercritical Brayton Cycle (SBC) for WHR of a Liquefied Natural Gas (LNG) carrier. SBC is an innovative, promising technology for power generation with unprecedented performance and a small form factor, due to the properties of the working fluid. A thermodynamic model is developed and programmed in MATLAB using the CoolProp free library. By means of this model, the performance of simple and recuperated SBC (RSBC) for WHR of a specific marine engine at its full load operation is assessed and the optimum compressor pressure ratio for power maximization of the RSBC is selected. The combined system exhibits an increase of about 2.9% in thermal efficiency and a similar reduction in specific fuel oil consumption, compared to the sole power production by the main engine, at its full load operation. Significant performance benefits are also demonstrated at part-load operation of the main engine. To assess how the benefits scale with the main engine power, seven similar marine engines of different power are considered, revealing a possible relationship between the optimal pressure ratio and SBC efficiency with the engine's exhaust gas temperature.

**Keywords:** supercritical $CO_2$ cycle, waste heat recovery, recuperator, combined cycle, LNG carrier

## 1 Introduction

Climate change has made industrial decarbonization an essential and urgent task. Although maritime transport is highly efficient, it accounts for about 2.9% of global emissions, which could rise 90–130% above 2008 levels by 2050 [1]. In response, the International Maritime Organization (IMO) set targets in 2018 to cut shipping Green House Gases (GHG) emissions by at least 50% by 2050 and to achieve a 40% carbon intensity reduction by 2030 and a corresponding 70% by 2050, compared to 2008 levels

[2]. Shipping is a multi-trillion-dollar industry facing major economic and logistical challenges in meeting IMO decarbonization targets [3], as current technologies are insufficient. Improving ship energy efficiency is therefore urgent, with propulsion and power systems offering the most direct opportunities. Since modern vessels rely heavily on diesel engines, alternative fuels appear promising for long-term decarbonization. However, the various alternative fuel options proposed present significant technical, safety, availability, and cost challenges, and require thorough life-cycle assessment to account for indirect emissions. Given the uncertainty surrounding new fuels and the risk associated to the related investments, it becomes evident that any potential solution for improving ship energy efficiency should be considered to support decarbonization in the medium and long term.

Steam and gas turbines have failed to dominate ship propulsion, largely due to the superior efficiency of diesel engines. Although steam turbines were widely used in early steamships and more recently in Liquefied Natural Gas (LNG) carriers to utilize boil-off gas, they were eventually replaced by dual-fuel diesel engines. Diesel engines offer higher efficiency, particularly at partial loads, making them a more attractive option for ship propulsion and power generation [4]. The supercritical $CO_2$ Brayton cycle (SBC) is an advanced technology that improves the efficiency of the conventional Brayton cycle by using $CO_2$ in its supercritical state as the working fluid, allowing for the design of very compact systems. Since SBC is not yet commercially mature, retrofitting older ships is impractical. Modern LNG carriers, running on natural gas, are being built in large numbers, and may soon require efficiency upgrades to meet future IMO rules. A Waste Heat Recovery system could generate mechanical power without extra fuel, improving overall operational efficiency. Among the advantages of using SBC are the requirement for lower compression work near the critical point of $CO_2$, the use of compact equipment of smaller dimensions, fewer compressor and turbine stages, and single-phase operation that avoids heat exchanger pinch point issues making it well-suited for marine applications.

Kim et al. [5] compared nine SBC layouts for gas turbine bottoming cycles, finding that, although the recompression cycle has the highest theoretical efficiency, it is unsuitable for bottoming applications. A dual-heated Brayton cycle with flow split offers the highest net work but is highly complex. Held et al. [6] analyzed SBC models and favored the simpler recuperated Brayton cycle for bottoming applications. Overall, recompression is rarely used for waste heat recovery, while recuperated cycles are popular due to their simplicity, compactness, and better off-design performance. SBC has shown strong potential for improving efficiency in onshore power plants and the last decade a lot of SBC configurations have been proposed, investigated and assessed; only in the last few years similar research has been extended to marine applications.

Hou et al. [7] studied Waste Heat Recovery (WHR) from marine diesel engines using SBC to generate electricity and improve thermal efficiency, concluding that system optimization is key for onboard applications. Sakalis [8] analyzed six configurations of SBC systems recovering heat from exhaust gas, scavenge air, and jacket cooling water of a marine engine; 6.6–7.25% efficiency gains were shown while accounting for heat exchanger size constraints. Yakkeshi and Jahanian [9] modeled four SBCs, finding that heat recovery reduces energy losses, with turbine inlet temperature improving performance and compressor inlet temperature reducing it; a single heat exchanger configuration achieved the highest usable power and efficiency (17.72% energy, 12.85% exergy). Hu et al. [10] showed that ship rolling destabilizes heat transfer in heat exchangers due to additional forces on the fluid, affecting the SBC system efficiency.

Reale et al. [11] studied WHR from gas turbine propulsion systems using SBC bottoming cycles and analyzed six compact layouts, including cascade ORC configurations; energetic and exergetic analyses showed efficiency gains up to 29%, with higher seawater temperatures reducing performance. Reale and Massoli [12] assessed off-design performance of a gas turbine coupled with a partially preheated, recuperated SBC, finding seasonal efficiency variations of 42–49% and WHR efficiency of 40–47%. Alzuwayer et al. [13] examined a cascade SBC system for marine gas turbines, showing that recompression cycle optimization can increase overall efficiency from 54% to 59%, offering a pathway to more energy-efficient marine propulsion.

One of the most promising applications is that of using SBC in nuclear-powered ships. Lee et al. [14] found that a small modular MSR coupled with SBC reaches 47.78% efficiency, about 12% higher than a PWR-based Rankine cycle. Ma et al. [15] analyzed simple and reheated SBCs for PWR-powered ships on the Northern Sea Route, reporting that reheated cycle achieves 30.1% efficiency with smaller heat exchangers and stating that SBC systems are ideal for space-limited marine applications, offering over 25 times higher volumetric power density than steam Rankine cycles. Since real gases deviate from ideal behaviour at high pressures due to molecular interactions, their accurate state modelling is essential for high-pressure power cycles like the SBC. Real gas models account for compressibility, variable heat capacities, van der Waals forces, and other effects, and are especially important near the critical or condensation points. Common such models include state equations of Van der Waals, Redlich-Kwong, Peng-Robinson, etc. Alternatively, thermodynamic look-up tables, generated from these models, can provide fluid properties for given states and are widely available.

In the light of the above, the present work examines the potential use of a SBC for WHR onboard system from a marine dual fuel engine, focusing on the Recuperated Supercritical Brayton Cycle (RSBC). Aiming to perform research on the use of SBC in marine applications in the long-term, the main objective of this paper is to perform a preliminarily thermodynamic design a closed-loop, recuperated SBC, indirectly fired by waste heat from a marine engine on an LNG carrier, and evaluate its performance. To this end, a thermodynamic model for the performance of the SBC and the combined cycle is developed in Matlab. Carbon dioxide state properties are taken into account by implementing the free CoolProp library [16]. A case study serves to conduct parametric studies of the combined cycle and assess its performance at full and part-load operation. A comparative study of seven engines with different powers is used to evaluate the scalability of the system benefits with engine size. Conclusions are drawn and directions for future research are proposed.

## 2 Methodology

### 2.1 Description of the recuperated SBC

Figure 1 depicts the configuration of a recuperated single-shaft closed SBC. The working fluid ($CO_2$), after being compressed by the compressor (C) and from state 1 (low cycle pressure) to state 2 (high cycle pressure), passes sequentially through the cold side of the recuperator and the heater, both being at high pressure. At the heater exit, except of high pressure, the working fluid has also high temperature and, as a consequence, high specific enthalpy. The expansion that follows in the turbine (T) produces work, part of which is consumed to move the compressor via the common C-T shaft, while the rest is the net work of the plant and is made available for the engine load at the free end of the shaft. The working fluid, still having high temperature, passes sequentially through the hot side of the recuperator and the cooler to obtain its initial state 1 and restart the cycle. In case of simple cycle the recuperator does not exist (or if it exists it is bypassed); in that case states 2 and 3 coincide and the same happens for states 5 and 6. The prerequisite in order to utilize a recuperator is that the relation $T_5>T_2$ has to be valid. This cycle corresponds to the operation of a thermal engine; the heater serves for the provision of heat to the cycle, the cooler serves for the rejection of heat, while he use of the recuperator makes the cycle regenerative by providing the capability of internal heat exchange. Figure 2 presents the ideal thermodynamic cycle corresponding to the aforementioned engine.

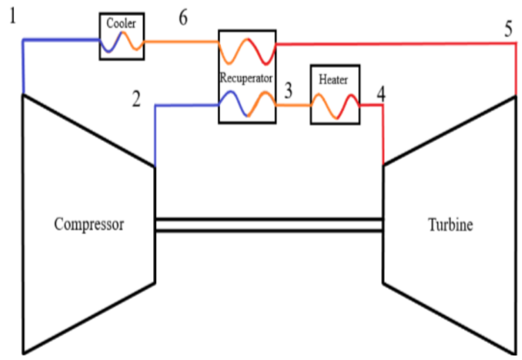



**Fig. 1.** Layout of recuperated Brayton cycle

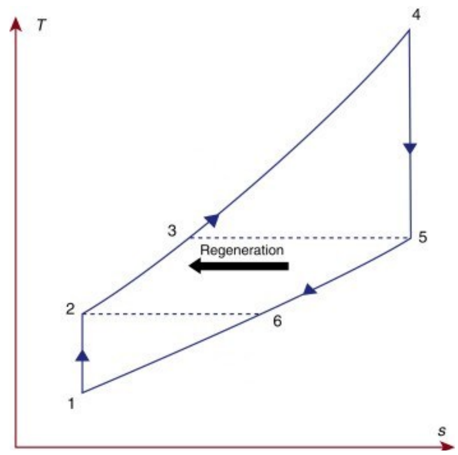



**Fig. 2.** Thermodynamic cycle of RSBC in T-s diagram

The main components required for the realization of SBC and RSBC are heat exchangers (heater, cooler and recuperator), turbomachinery (compressor and turbine) and ducts connecting these components. The real cycle deviates from the ideal one by considering isentropic efficiencies lower than 100% for the compression in C and the expansion in T, respectively, as well as by taking into account pressure losses in the heat exchangers. In a supercritical cycle, the working fluid does not incur a phase transition, thus the pressure and temperature of the fluid must always be kept above its critical point. Due to the already high pressure of the critical point, it is suggested that the minimum cycle pressure is kept as low as possible. However, possible condensation that poses a great risk for the safe and efficient operation of the turbomachinery should be carefully examined [17]. Therefore, there should always be a safety margin between the minimum pressure of the cycle and the critical pressure of $CO_2$, while a similar statement holds for the minimum temperature of the cycle.

### 2.2 Thermodynamic calculation of SBC and RSBC performance

The formulas required for thermodynamic analysis of SBC and RSBC and calculation of their performance, as well as the required data are provided in the Appendix.

## 3 Description of case study

In the present work, a supercritical carbon dioxide Brayton cycle bottoming the main engine of a LNG carrier for waste heat recovery is considered as the case study. The engine of choice is a state-of-the-art one, a six cylinder dual fuel engine, aimed for use at the LNG carrier sector [18] (MAN B&W 6G70ME-C10.5-GA-EGRBP, paired with a MHI MET53-MBII turbocharger). Its operation is based on the premixed Otto principle and is capable of operating on low pressure fuel supply. It also features an exhaust gas recirculation system (EGR), further reducing $NO_X$ emissions. It is designed to reduce methane slip on low pressure dual fuel engines while focusing on keeping the capital expenses low. It is fully Tier III-compliant when running on dual fuel mode, as well as on conventional fuel oils with the help of EGR. Finally, it is capable of producing 16980 kW at 78 rpm at its Specified Maximum Continuous Rating (SMCR) point of operation.

In a WHR system, the heat input rate to the bottom cycle is determined by the temperature and mass flow rate of engine's exhaust gas. To acquire these necessary data at various load conditions of the main engine, CEAS (Computerised Engine Application System) [19], a free software provided by the engine's manufacturer, is utilized. To obtain the aforementioned data, , the engine was assumed to operate in Tier III mode fueled by oil (MDO or MGO) in ISO ambient conditions (ambient air: 25$^{o}$C, scavenge air coolant: 25$^{o}$C). Table 1 summarizes the exhaust gas data at various loading conditions of the main engine according to the CEAS results.

**Table 1.** Performance and exhaust gas data at various loading conditions of the main engine

| Load [% MCR] | Power [kW] | SFOC* [g/kWh] | Exhaust gas flowrate [kg/s] | Exhaust gas temperature [$^{o}$C] |
|---|---|---|---|---|
| 100 | 16980 | 179.0 | 23.4 | 270 |
| 95 | 16131 | 176.1 | 23.0 | 243 |
| 90 | 15282 | 174.0 | 22.6 | 219 |
| 85 | 14433 | 172.5 | 21.8 | 215 |
| 8- | 13584 | 171.5 | 20.9 | 213 |
| 75 | 12735 | 171.1 | 20.0 | 212 |
| 70 | 11886 | 171.0 | 18.8 | 213 |
| 65 | 11037 | 171.0 | 17.7 | 215 |
| 60 | 10188 | 171.2 | 16.4 | 218 |
| 55 | 9339 | 171.5 | 15.1 | 223 |
| 50 | 8490 | 172.0 | 13.8 | 229 |
| 45 | 7641 | 172.6 | 12.3 | 238 |
| 40 | 6792 | 173.4 | 10.8 | 249 |
| 35 | 5943 | 174.4 | 9.1 | 284 |
| 30 | 5094 | 175.6 | 7.5 | 322 |
| 25 | 4245 | 177.0 | 6.2 | 337 |

*Specific Fuel Oil Consumption

Before proceeding, it is important to make the objective of the study clear. A WHR device utilizes the exhaust gas of an engine to produce power. The heat input for such a device comes exclusively from the main engine exhaust gas, thus no further fuel has to be consumed. The ultimate design goal for those devices is to improve the overall efficiency of the main engine in combination with the waste heat recovery device as a combined system. This is achieved by designing a bottoming cycle aiming to produce the maximum possible power. In this way, the exact same amount of fuel is utilized by the main engine in order to produce the maximum possible power. Bottoming cycle efficiency may not necessarily be the main focus when designing a WHR device, as it is possible to design a more efficient yet less productive device that contributes less to the overall efficiency of the system, compared to a device that produces more power with less efficiency. The efficiency of the bottoming cycle may be useful when comparing two WHR devices of similar heat input.

In the context of the case described above, three different studies are performed:

(a) The first study concerns the performance assessment of the SBC, bottoming the main engine at its SMCR operation, for different compressor pressure ratios. The aim is to find the optimal pressure ratio and calculate the required $CO_2$ mass flow rate with the goal of maximizing the SBC net work output; based on these values, a preliminary design of a recuperated SBC is provided. In the course of the calculations of SBC for various pressure ratio values, temperatures $T_2$ and $T_5$ of the SBC are compared; whenever the relation $T_5>T_2$ holds, the recuperator can be utilized and the RSBC is simulated instead; otherwise only the SBC is be considered and solved. For the designed SBC, the overall system performance, as well as the contribution of the SBC to it are assessed.

(b) The second study concerns the performance assessment of the designed RSBC when the main engine operates at partial loads. In this scenario, the optimal pressure ratio and $CO_2$ mass flow rate, found before for maximum performance at full load, are used. Since, mass flowrate and temperature of the engine exhaust gas, both change at partial load operation, it is possible that in some cases the necessary condition for utilizing the recuperator, i.e. $T_5>T_2$, does not hold; in those cases the recuperator is bypassed and the SBC is considered and solved.

(c) The third study concerns the effect of the main engine power to the performance of the combined cycle and is accomplished by examining, through the generated software, a series of similar engines but of different power.

In order to perform the simulations required for the studies mentioned above, several assumptions are made, concerning the steady state modeling of SBC / RSBC and based on the relevant literature [20]:

- The margins for the minimum temperature and minimum pressure of the cycle (state 1) above the critical point are kept at $\Delta T_1$=10K and $\Delta p_1$=0.2MPa, respectively
- The values of 0.85 and 0.9 are used for the isentropic efficiencies of the compressor and turbine, respectively and are assumed to be constant
- The pressure loss coefficient is assumed to be 1% for all heat exchangers involved
- Pressure losses inside the ducts connecting other components are neglected
- Pressure losses of the main engine's exhaust gas inside the heater are neglected and, as a consequence, the performance of the main engine is not considered to affect by the use of the WHR system
- According to the Marine Environment Protection Committee document "Annex 9 Resolution MEPC.281(70)" [21], a value of 42700 kJ/kg is used for the lower heating value of fuel oil (also confirmed by the engine manufacturer's documents)

- An average value of 1.15 kJ/(kgK) is used for the heat capacity of the exhaust gas
- The values of the various parameters used for the SBC and RBC when bottoming the main engine at full load, are also used in the case of part-load operation of the main engine. Thus, only changes in mass flowrate are taken into account, while possible changes in pressure ratio, isentropic efficiencies and pressure loss coefficients are not considered
- The minimum temperature of gas discharge to the environment after the heater is set to 130°C (due to acid dew point of exhaust gas)

**Summary of numerical data for the calculations** (see Appendix for the symbols)

| | | | |
|---|---|---|---|
| $p_{cr}$ =7.38MPa | $\eta_t$ =0.9 | $T_{g,i}$ = (from Table 1) | $K_{r,h}$ = 0.01 |
| $T_{cr}$ =304K | $K_h$ =0.01 | $\Delta p_1$ = 0.2MPa | $\Delta T_g$ =10K |
| $r_C$ =1.5÷5 | $K_c$ =0.01 | $\Delta T_1$ = 10K | $T_{g,min}$ =130°C |
| $\eta_c$ =0.85 | $m_g$ =(from Table 1) | $K_{r,c}$ = 0.01 | $LHV$=42700 kJ/kg |

# 4 Results and discussion

## 4.1 Full load operation of main engine

With the use of the model described in the previous sections, the performance of the SCBC as a standalone WHR system, as well as that of the combined main engine-SBC system, are first evaluated at the engine's SMCR. The power output, thermal efficiency, exhaust gas temperature after the heater and $CO_2$ mass flow rate of the SBC, are calculated for various values of the compressor pressure ratio. Furthermore, the performance of the RSBC is compared to that of the SBC, in order to confirm the conviction that a recuperated Brayton cycle is a more suitable configuration for WHR.

Fig. 3 presents the net power output of SBC and RSBC as a function of compressor pressure ratio, at full load operation of the engine. According to it, the recuperated cycle can be operated only for pressure ratios up to 3.35 due to temperature difference limitations between the turbine and compressor output (i.e. for $r_C$>3.35, the required condition $T_2$<$T_5$ does not hold). Both SBC and RBSC configurations have a similar power output, with the recuperated cycle producing slightly more power for pressure ratios lower than 2.75 and the simple configuration surpassing the recuperated in terms of power production in higher pressure ratios; the latter fact is attributed to the pressure losses in the recuperator.

Fig. 4 presents the corresponding curves of thermal efficiency as a function of the pressure ratio for both the SBC and RSBC. As in Fig.3, thermal efficiency is higher for the RSBC. The SBC surpasses RSBC only for pressure ratios higher than 3.24. A higher thermal efficiency is indeed expected for the RSBC due to the fact that a large part of the required heat is provided internally (regeneration effect). It is also noteworthy that thermal efficiency does not necessarily increase with the increase of pressure ratio due to the irreversibilities of the cycle.

In Fig. 5, the exhaust gas temperature of the main engine at the heater outlet is displayed for both configurations. Due to its higher thermal efficiency and thus lower waste heat usage, the recuperated cycle has a higher exhaust gas temperature at the outlet of the heater for every pressure ratio that it is applicable for. This means that the exhaust gas can be further utilized for other purposes like for steam generation. Another important thing to note, is that the simple configuration cannot operate with the limitations and assumptions of the present model for pressure ratios lower than 2.71, due to the fact that the exhaust gas temperature drops below 130ºC, which is the exhaust gas acid dew point.

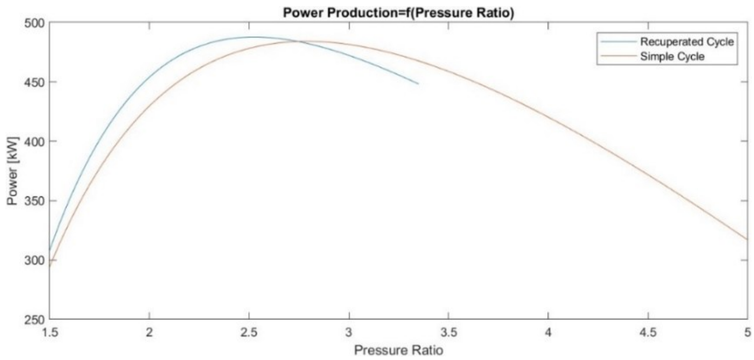



**Fig. 3.** Net power output of SBC as a function of pressure ratio for the simple cycle (red) and the recuperated cycle (blue)

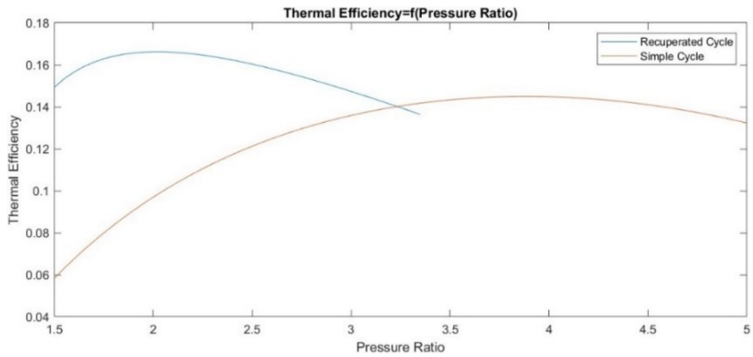



**Fig. 4.** Thermal efficiency of SBC as a function of compressor pressure ratio for the simple cycle (red) and the recuperated cycle (blue)

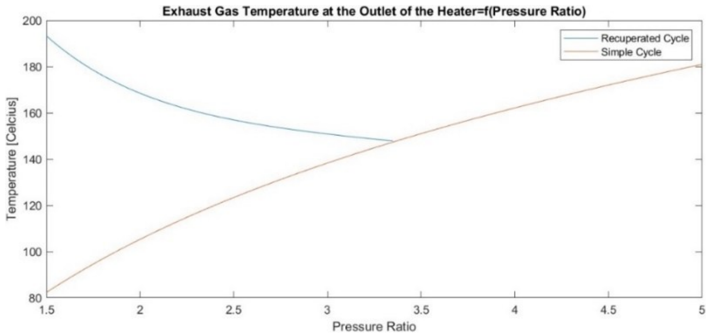



**Fig. 5.** Exhaust gas temperature at the heater outlet as a function of the compressor pressure ratio for a simple cycle (red) and a recuperated cycle (blue)

Fig. 6 presents the calculated $CO_2$ mass flowrate as a function of the compressor pressure ratio for both SBC and RSBC. According to it, the RSBC allows for a larger mass flow rate resulting in a higher power output. Assuming no pressure losses inside the recuperator, the specific net work output is exactly the same for both configurations.

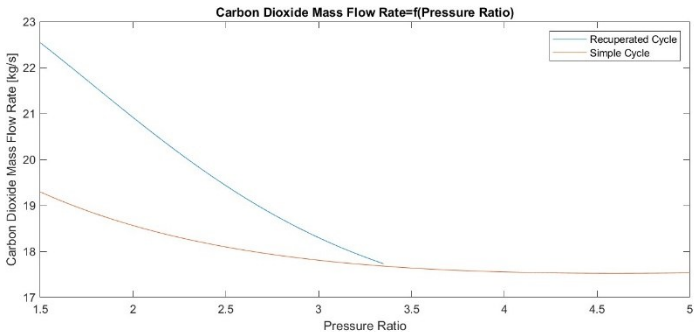



**Fig. 6.** Calculated carbon dioxide mass flowrate as a function of the compressor pressure ratio for a simple cycle (red) and a recuperated cycle (blue)

Summarizing, the optimal operating point of the system is determined with the goal to maximize the power output of the WHR system; this occurs in the recuperated cycle for a pressure ratio equal to 2.55. The characteristics of the RSBC, the main engine and the Combined Cycle (CC) for the above pressure ratio are summarized in Table 2. For the sake of completeness, the corresponding temperature-entropy diagram of the RSBC for the optimal pressure ratio is displayed in Fig. 7.

**Table 2.** Characteristics of RSBC, main engine and CC for the optimal pressure ratio

| RSBC | Main Engine | Combined Cycle |
|---|---|---|
| Pressure ratio 2.55 | Load 100% | Power 17468 kW |
| Min pressure 7.577 MPa | Power 16980 kW | Thermal efficiency 0.49 |
| Max pressure 19.322 MPa | SFOC 179 g/kWh | SFOC 174 g/kWh |
| Min temperature 41 °C | Gas flowrate 23.4 kg/s | Gas temperature 156 °C |
| Max temperature 260 °C | Gas temperature 270 °C | Efficiency increase 2.873% |
| Power 488 kW | Thermal efficiency 0.47 | Power increase 2.873% |
| Thermal efficiency 0.159 | | SFOC reduction 2.792% |
| $CO_2$ mass flowrate 19.3 kg/s | | |
| Heat input rate 3063.1 kW | | |
| Heat recuperation rate 1115 kW | | |
| Cooling rate 2575.3 kW | | |
| Heater effectiveness 0.919 | | |
| Recuperator effectiveness 0.832 | | |

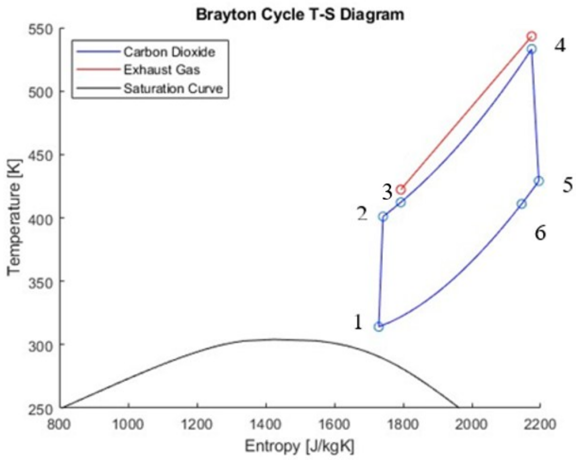



**Fig. 7.** Temperature-entropy diagram of the SBC for the optimal pressure ratio

According to the results of Table 2, the designed RSBC shows excellent performance as a waste heat recovery system at a relatively low pressure ratio. It exhibits an increase of 2.9% in thermal efficiency as a combined main engine-s$CO_2$ system with respect to the main engine efficiency and a similar corresponding reduction in specific fuel oil consumption at full load operation of main engine.

Compared to the RSBC configuration developed by Xie and Yang [22] for use with a smaller marine Diesel engine, both models exhibit about the same efficiency, at similar pressure ratios, which further confirms that in order to achieve the maximum theoretical efficiency of the recompression cycle, a higher temperature heat source is required. Furthermore, as suggested in [22], the system performance can be further enhanced by means of exhaust gas modulation.

A great advantage of the SBC as a WHR system is that, besides its small footprint, it can be cooled by readily available coolants like water or even air in some cases, due to the fact that the minimum temperature of the cycle is always above the carbon dioxide's critical temperature. It is important to notice that in the case under consideration, the minimum temperature is 41°C, which means that the system can be easily cooled by sea water.

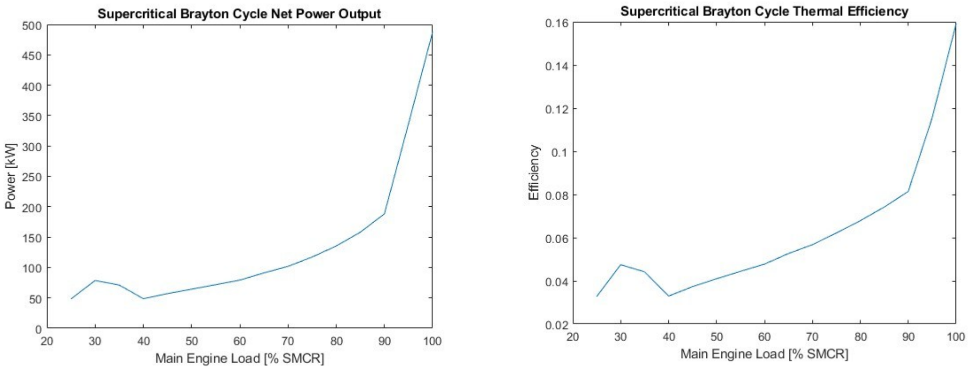
Finally, it is worth noting how such a WHR system can actually reduce the overall energy efficiency of the ship. The main engine exhaust gas at the outlet of the WHR system is 156°C. This means that there is only a narrow margin of 26°C before the exhaust gas starts entering the acid dew point region. Therefore, it would be difficult to find an application further utilizing the exhaust gas. Most modern ships, however, already use WHR systems in the form of boilers called economizers. Using a SCBC as a waste heat recovery method means than an economizer can no longer be used, at least in the context of the present model. Thus, a more detailed study and comparison between those systems has to be conducted in order to determine which one is more beneficial in terms of overall ship energy efficiency.

### 4.2 Part-load operation of main engine

In this section, the performance of the RSBC is studied for partial load operation of the main engine. In this scenario, the optimal pressure ratio found before is kept constant. Since the engine exhaust gas amount and temperature change at partial loads, it is possible that the recuperator may need to be bypassed and turn to a SBC operation; therefore, the simulation algorithm is appropriately modified in order to take account the case of a possible bypass.

In what follows, the net power output and efficiency of both SBC and CC, as well as other parameters are plotted in terms of the various main engine part-load scenarios (from 100% of the SMCR down to 25% of the SMCR). The corresponding simulations show that thermal recuperation cannot be used at loads lower than 95% of the SMCR; thus, for these loads bypass of the recuperator is applied in order to keep the system operational at partial loads.

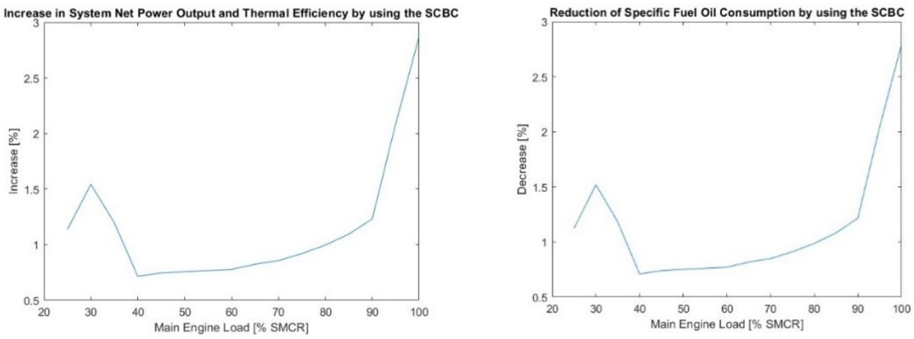
Fig. 8(a) displays the net power output of the SBC against the main engine load; a steep decrease in power is observed in the load reduction range 100%-90%, followed by a smooth decrease for 90%-40% and a sudden spike at lower loads. The higher gradient in the range 100%-90% is due to the significant change in the exhaust gas temperature; it drops 51°C compared to 3°C drop in the 90%-80% range. As for the spike in the 35%-25% range, the explanation is that in this range, the exhaust gas temperature starts increasing significantly, providing a higher cycle heat input rate at that range. In general, since the mass flow rate of the exhaust gas increases with the increase of the main engine load, it is normal that the power output of the cycle has an increasing trend as the engine gets more loaded. Fig. 8(b) displays the corresponding plot of the SBC efficiency for various engine loads, following a similar trend to that of the net power output; it decreases at part-load operation of main engine, despite the fact that the pressure ratio remains constant. This is attributed to the irreversibilities involved in the real cycle.




(a) (b)

**Fig. 8.** (a) Net power output of SBC as a function of main engine load. (b) Thermal efficiency of SBC as a function of the main engine load

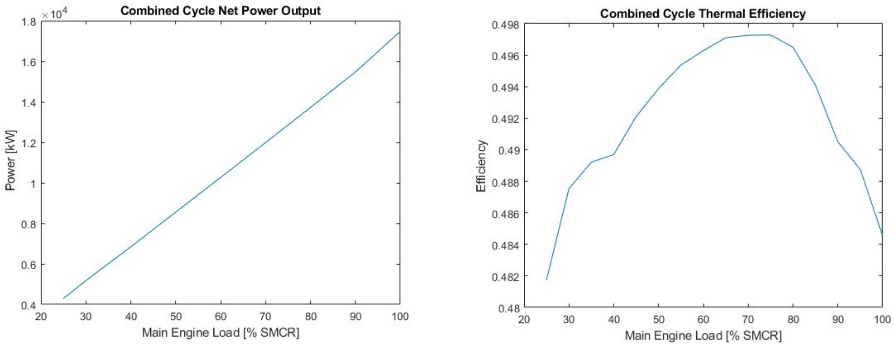
Fig. 9(a) presents the percentage increase of the overall system net power production and thermal efficiency due to the utilization of SBC for WHR at different loads. The corresponding percentage reduction in Specific Fuel Oil Consumption (SFOC) is presented in Fig. 9(b). It is evident that the SBC offers a significant improvement in the overall system performance when used for WHR, especially at higher loads where it can provide an up to 2.9% increase in power and efficiency and a similar decrease in SFOC. Although this may seem not to be a great value, considering the large amount of fuel consumed by such vessels, even a small improvement can result in the long term in significant decrease in GHG emissions and operating costs.




(a) (b)

**Fig. 9.** (a) Percentage increase in overall net power output and thermal efficiency due to the use of SBC for WHR. (b) Percentage reduction of SFOC due to WHR by means of SBC

Fig. 10(a) depicts the plot of the combined cycle power against load; power linearly varies with the variation of load attaining its maximum at the full load operation of the main engine. Fig. 10(b) depicts the corresponding plot of the combined cycle thermal efficiency. According to it, contrary to what happens in power, the maximum overall efficiency of the combined cycle is achieved at about 70% load of the main engine.




(a) (b)

**Fig. 10.** (a) Variation of the combined cycle power against load. (b) Variation of the combined cycle thermal efficiency against load

### 4.3 Effect of main engine power in WHR by SBC

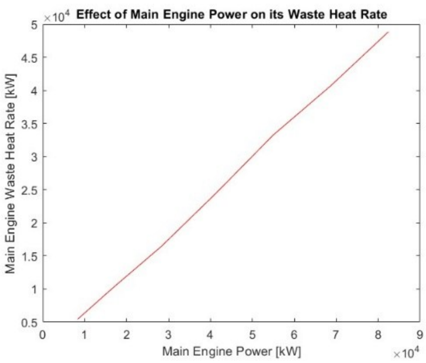
In this section, an attempt is made to assess the effect of the main engine power to the performance of the combined cycle. To this end, a series of seven engines, similar to the main engine selected before is considered [18]. These engines are of gas injection technology (GI-Gas Injection) with power outputs ranging from 8340 to 82440 kW and their characteristics [19] are provided in Table 3. According to this, there is an increase in the exhaust gas amount as the power increases, which is expected, due to the fact that higher engine power corresponds to more working fluid and thus higher exhaust gas mass flow rate. However, the same is not true for the exhaust gas temperature, which is maximum for the low power engine and minimum for the medium to low power engines, while high power engines stand somewhere in between. The exhaust gas temperature is a parameter more difficult to predict, as it depends on a variety of factors like the geometry of the combustion chamber, the air-fuel mixture, as well as several other combustion process parameters. The exhaust gas amount and the exhaust gas temperature obviously play an important role for the available heat input to the bottoming cycle.

**Table 3.** MAN ME-GI Marine Engine Characteristics (EGA= Exhaust Gas Amount, EGT=Exhaust Gas Temperature) [12]

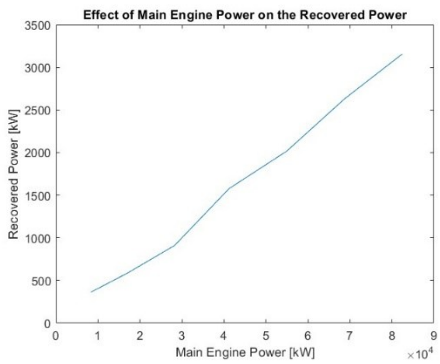
| Model | Power [kW] | SFOC [g/kWh] | EGA [kg/s] | EGT[°C] |
|---|---|---|---|---|
| 6G45ME-C9.5-GI-HPSCR | 8340 | 172 | 17.4 | 270 |
| 6G60ME-C10.5-GI-HPSCR | 17040 | 167 | 36.4 | 245 |
| 6G80ME-C10.5-GI-HPSCR | 28260 | 162 | 58.5 | 242 |
| 6G95ME-C10.5-GI-LPSCR | 41220 | 161 | 79.4 | 265 |
| 8G95ME-C10.5-GI-LPSCR | 54960 | 165 | 112.4 | 255 |
| 10G95ME-C10.5-GI-LPSCR | 68700 | 161 | 132.4 | 265 |
| 12G95ME-C10.5-GI-LPSCR | 82440 | 161 | 158.9 | 265 |

For each of the engines presented in Table 3, a similar study like that of section 4.1 at the engine's SMCR is conducted to obtain a preliminary design of the use of SBC for WHR of the engine. In particular, the optimal pressure ratio of the SCBC is found and the performance of the SBC in terms thermal efficiency and power contribution to the combined cycle is analyzed.

Fig. 11(a) presents the plot of the available waste heat rate ($Q_{WH}=m_g c_{pg} T_g$) of the engine as a function of the main engine power; the former quantity increases almost linear with the increase of the latter. Fig. 11(b) presents the plot of the power recovered by the SBC, which increases accordingly (about linearly) with the increase of main engine power. Thus, the SCBC produces more power when paired to a high power engine.




(a)




(b)

**Fig. 11.** (a) Effect of main engine power in the available waste heat rate.
(b) Effect of main engine power in the power recovered by the SBC

Fig. 12 shows the effect of the main engine power in the predicted compressor optimal pressure ratio. The latter does not exhibit a specific dependency on the engine power. By examining the rest engine data closely, it can be seen that the engines having the same exhaust gas temperature also share the same optimal pressure ratio. Fig. 12

presents the effect of the main engine power in the overall CC thermal efficiency. In a similar way to the optimal pressure ratio, the thermal efficiency does not seem to have a specific dependency on the engine power, but increases with the increase of the exhaust gas temperature. The corresponding plot of the percentage power contribution of SBC to the total CC power against main engine nominal power exhibits exactly the same trends [23].

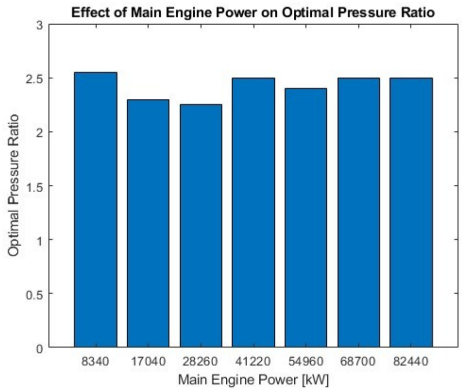



**Fig. 12.** Effect of main engine power in the predicted compressor optimal pressure ratio

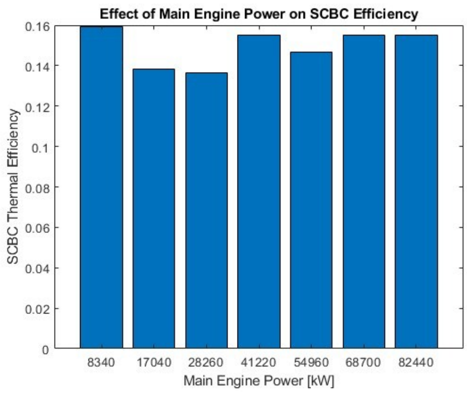



**Fig. 13.** Effect of main engine power in the percentage contribution of SBC in overall CC power output

# 5 Conclusions - Future research

This work investigated the utilization of a carbon dioxide supercritical Brayton cycle for waste heat recovery from a LNG carrier engine. A thermodynamic model was developed and programmed in-house. The performance of simple and recuperated SBC

(RSBC) for WHR of a specific marine engine at its full load operation was assessed and the optimum compressor pressure ratio for power maximization of the RSBC was selected. The designed RSBC exhibits an increase of 2.9% in thermal efficiency of the combined main engine -SBC system and a 2.8% reduction in specific fuel oil consumption at full load operation of main engine. Performance benefits were also demonstrated at part-load operation of the main engine. To assess how the benefits scale with the main engine power at full load, seven similar marine engines of different power were considered and their performances were compared each other, revealing that optimum SBC pressure ratio and efficiency actually scale with the temperature of the main engine exhaust gas.

To further develop the methodology developed and presented herein, the following research directions are proposed:

- Recap the major challenges of commercializing the SBC for maritime applications
- Model the closed gas turbine cycle at partial loads
- Perform advanced exergy analysis of the SBC with the goal of determining the performance limits of the cycle and focusing on the components that need to be further optimized
- Examine adopting preheating by also utilizing the jacket cooling water
- Develop a thermodynamic model for simulating the recompression SBC for waste heat recovery in maritime applications and optimize the flow split ratio
- Compare SBC and Organic Rankine Cycle for WHR of the same engine
- Compare the present results with corresponding ones by Brayton-SBC combined cycles where the marine engine is a gas turbine of similar power
- Perform an thorough preliminary design of the implementation of SBC for WHR in a LNG carrier, involving feasibility and technoeconomic analysis

## Appendix: Thermodynamic calculation of SBC and RSBC performance

In this appendix, the required data and the calculation procedure for the performance calculation of the simple SBC and the RSBC are provided. Figs 14 and 15 depict two such thermodynamic cycles and demonstrate the various states of the working medium along the cycles. It has to be noticed that RSBC can be used only in case that the condition $T_5>T_2$ in the SBC holds.

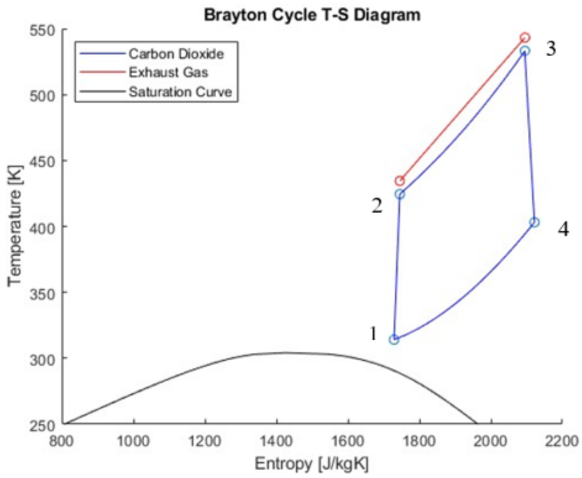



**Fig. 14.** Simple SBC ($T_5<T_2$)

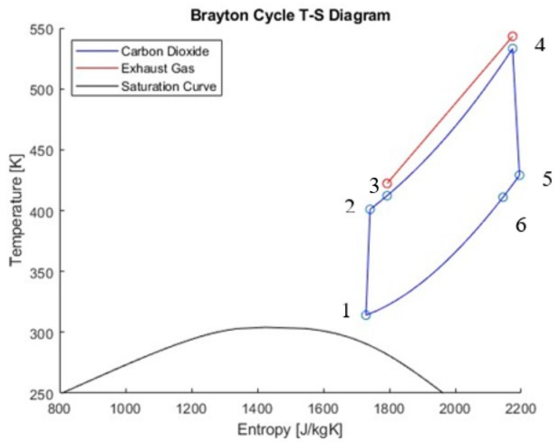



**Fig. 15.** Recuperated SBC ($T_5>T_2$)

In what follows, the symbols for the various thermodynamic quantities are explained, the given data for the problem to solve are listed, and the formulas to calculate the states of the working fluid along the cycle and evaluate its performance are provided, first for SBC (Algorithm I) and then for RSBC (Algorithm II).

**Nomenclature**

T: temperature
p: pressure
h: specific enthalpy
s: specific entropy
r: pressure ratio
W: power
w: specific work
Q: heat rate
ηis: isentropic efficiency
η: thermal efficiency
K: pressure loss coefficient
m: mass flowrate

**Subscripts**

1, 2, …, 6: states along the SBC
cr: critical
C: compressor
T: turbine
h: heater
c: cooler
g: flue gas
r: recuperator
r,h: hot side of recuperator
r,c: cold side of recuperator

**Given data for the thermodynamic calculation of the SBC and RSBC performance**

pcr: critical pressure
Tcr: critical temperature
rC: compressor pressure ratio
ηc: compressor isentropic efficiency
ηt: turbine isentropic efficiency
Kh: heater pressure drop coefficient
Kc: cooler pressure drop coefficient
LHV: Lower Heating Value of fuel
mg: exhaust gas mass flowrate
Tg,i : exhaust gas temperature
Δp1: pressure difference above pcr
ΔT1: temperature difference above Tcr
Kr,c: recuperator pressure drop coefficient, cold side
Kr,h: recuperator pressure drop coefficient, hot side
ΔTg: temperature difference in heater inlet and outlet
Tg,min: minimum allowed gas temperature
cpg: heat capacity of exhaust gas

**Algorithm I: Calculation of SBC performance (states 3=2 and 6=5)**

State-1: $p1=pcr+\Delta p1,\ T1=Tcr+\Delta T1,\ h1=h\ (p1,\ T1),\ s1=s\ (p1,\ T1)$

State-2=3: $p2=rCp1,\ h2s=h\ (p2,\ s1),\ wC=(h2s–h1)/\eta C,\ h2=h1+wC,\ T2=h\ (p2,\ h2),\ s2=s\ (p2,\ h2)$

State-4: $p_4=(1-K_H)\ p_2,\ T_4=T_{g,I}-\Delta T_g,\ h_4=h\ (p_4,\ T_4),\ s_4=s\ (p_4,\ T_4)$

Heater: $q_H= h_4-h_2,\ T_{g,o}=T_2+\Delta T_g,\ Q_H=m_gc_{pg}(T_{g,i}-T_{g,o})=mq_H \rightarrow m=m_gc_{pg}(T_{g,i}-T_{g,o})/q_H$

State-5=6: $p_5= p_1/(1-K_c),\ h_{5s}=h\ (p_5,\ s_4),\ w_T=\eta_T(h_4–h_{5s}),\ h_5=h_4-w_T,\ T_5=T\ (p_5,\ h_5)$

Performance: $w = w_T - w_C,\ W=mw,\ \eta = w/q_H$

In the above cycle, if $T_5 > T_2$, then thermal recuperation is possible. In that case, states 3 and 6 have also to be taken into account, since 3≠2 ($T_3 > T_2$) and 6≠5 ($T_6 < T_5$), as described in Algorithm II below.

**Algorithm II: Calculation of RSBC performance**

State-1: $p_1=p_{cr}+\Delta p_1,\ T_1=T_{cr}+\Delta T_1,\ h_1=h\ (p_1,\ T_1),\ s_1=s\ (p_1,\ T_1)$

State-2: $p_2=r_C p_1,\ h_{2s}=h\ (p_2,\ s_1),\ w_C=(h_{2s}-h_1)/\eta_C,\ h_2=h_1+w_C,\ T_2=h\ (p_2,\ h_2),\ s_2=s\ (p_2,\ h_2)$

State-3: $p_3=(1-K_{R,c})p_2$

State-4: $p_4=(1-K_H)\ p_3,\ T_4=T_{g,i}-\Delta T_g,\ h_4=h\ (p_4,\ T_4),\ s_4=s\ (p_4,\ T_4)$

State-5: $p_5=p_1/(1-K_{R,h})/(1-K_c),\ h_{5s}=h\ (p_5,\ s_4),\ w_T=\eta_T(h_4-h_{5s}),\ h_5=h_4-w_T,\ T_5=T\ (p_5,\ h_5)$

State-6: $p_6=p_5(1-K_{R,h}),\ T_6=T_2+\Delta T_R,\ h_6=h\ (p_6,\ T_6)$

State-3: $p_3=(1-K_{R,c})p_2,\ q_R=h_5-h_6,\ q_R=h_3-h_2 \rightarrow h_3=h_2+q_R,\ T_3=T\ (p_3,\ h_3)$

Heater: $q_H=h_4-h_3,\ T_{g,o}=T_3+\Delta T_g,\ Q_H=m_g c_{pg}\ (T_{g,i}-T_{g,o})=mq_H \rightarrow m=m_g c_{pg}(T_{g,i}-T_{g,o})/q_H$

Performance: $w = w_T-w_C,\ W=mw,\ \eta = w/q_H$

Check: If the temperature of the exhaust gas in the heater outlet drops below the minimum allowed due to acid dew point ($T_{g,o} < T_{g,min}$), set $\Delta T_g = \Delta T_g + 1^oC$ and repeat the calculation.

**Performance of Main Engine (ME)**

$\eta_{ME} = W_{ME} /(Q_{in,ME})=3600000 / (LHV\cdot SFOC_{ME})$ , $W_{ME},\ SFOC_{ME}$ from Table 1

**Performance of Combined Cycle (CC)**

$W_{CC}=W_{ME}+W,\ \eta_{CC} = W_{CC} /(Q_{in,ME}) = W_{CC} / (LHV\cdot W_{ME}\cdot SFOC_{ME})$

$SFOC_{CC} = 1 / (\eta_{CC}\cdot LHV)$

The above described algorithm has been programed in MATLAB utilizing the Cool-Prop free library [16], being linked with the simulation software (further details can be found in [23]).

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