Thermodynamic analysis of waste heat recovery using Organic Rankine Cycle (ORC) for a two-stroke low speed marine Diesel engine in IMO Tier II and Tier III operation

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ABSTRACT

In the present study, a complete thermodynamic model of a two-stroke, low speed, 13.6 MW marine Diesel engine of Winterthur Gas & Diesel has been developed using the engine simulation software Ricardo WAVE. The model has been first validated against experimental data. A Low Pressure (LP) EGR architecture has then been implemented in order to assess the engine performance in the frame of the IMO Tier III regulations. The computational results have been used as inputs to a thermodynamic process simulation model, developed in Engineering Equation Solver, able to quantify the performance of different Organic Rankine Cycle (ORC) architectures and working fluids, with the scope of obtaining the maximum net power output for all engine operating points considered. The outcome of the present study is that, through the combined use of innovative emission reduction strategies, such as LP EGR, and waste heat recovery systems, such as ORC, it is possible to develop marine Diesel engines which exhibit fuel consumption levels comparable to those of Tier II operation, at substantially reduced levels of pollutant emissions. A preliminary economic analysis has yielded annual financial savings in fuel cost of the order of 5% for operation with ORC, as compared to operation without ORC.

1. Introduction

Heavy duty Diesel engines are widely used in several applications, such as power generation, commercial vehicles and ship propulsion. For this reason, they are also among the main contributors to Greenhouse Gases (GHG), as CO_2, and other pollutant emissions, as NO_x, SO_x and Particulate Matter (PM), as presented in the Third IMO Greenhouse Gas Study (2014) [1].

The Third IMO GHG study also reports on the fuel consumption and CO_2 emissions estimated for different types of ships in 2012. The data are presented, after elaboration, in Fig. 1, and indicate that the ship types with high CO_2 emissions (and high fuel consumption) are container ships, bulk carriers, oil tankers, general cargo ships and chemical tankers. These types of ships are typically powered by low speed, two-stroke propulsion units, as those analysed in the present work.

Emission Control Areas are currently established in the Baltic Sea (SO_x), North Sea (SO_x), North America (NO_x and SO_x), US Caribbean Sea, Puerto Rico and US Virgin Islands (NO_x and SO_x), but other areas are currently also under consideration, with particular interest to Mexican Caribbean Sea, Mediterranean Sea and Japan. In these areas, a reduction of up to 76% of NO_x emissions, with respect to Tier II limits, is compulsory, in order to meet IMO Tier III emission levels; Tier II levels still dictate the NO_x emission limits outside the ECAs [2].

Tier II standards are met by combustion process optimization and engine operation improvements, with particular focus on fuel injection timing, injection pressure, injector developments, exhaust valve timing variations (e.g. Miller timing) combined with

$\dot{m}$ mass flow rate, kg/s
p pressure, bar
PR pressure ratio
T temperature, °C
$\dot{W}$ power, kW
x vapour quality

Acronyms
B   Blower
BSFC   Brake Specific Fuel Consumption, g/kWh
C   Compressor
CAC   Charge Air Cooler
COND   Condenser
ECA   Emission Control Area
EGR   Exhaust Gas Recirculation
ENG   Engine
EVAP   Evaporator
EXP   Expander
FMEP   Friction Mean Effective Pressure, bar
GHG   Greenhouse Gas
HFO   Heavy Fuel Oil
HP   High Pressure
HT   High Temperature
IMO   International Maritime Organization
LNG   Liquefied Natural Gas
LP   Low Pressure

LT   Low Temperature
MDO   Marine Diesel Oil
NTUA   National Technical University of Athens
ORC   Organic Rankine Cycle
P   Pump
PP   Pinch Point
SAC   Scavenge Air Cooler
SCR   Selective Catalytic Reduction
SFOC   Specific Fuel Oil Consumption, g/kWh
T   Turbine
TEU   Twenty-foot Equivalent Unit
VGT   Variable Geometry Turbine
WinGD   Winterthur Gas & Diesel
WTS   Water Treatment System

Greek symbols
η efficiency
Δ delta (e.g., ΔT, temperature difference)

Subscripts and superscripts
cf cooling fluid
$cool$ coolant
$econ$ economizer
exh exhaust gas
net net (e.g., ORC power output)
$sub-cool$ sub-cooling
$suph$ super-heating
wf working fluid

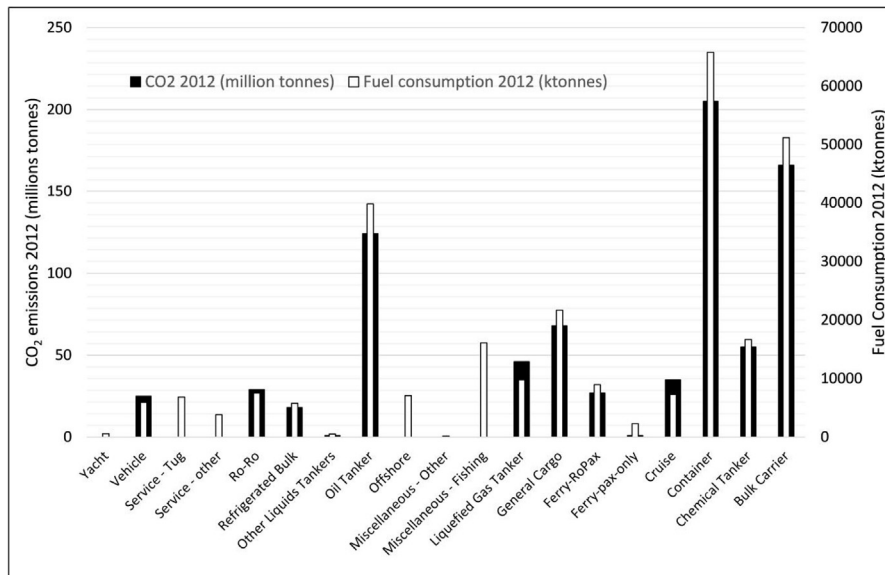**Fig. 1.** *CO_2 emissions and fuel consumption for different types of ships in 2012 (elaborated from the IMO GHG Study 2014 [1]).*

turbocharging strategies and adjustment of cylinder compression ratio [3–5].

On the other hand, Tier III standards require drastic NO_x emission control strategies or technologies, such as water injection, Exhaust Gas Recirculation (EGR) or Selective Catalytic Reduction (SCR) [6]. In the present study, a Low Pressure (LP) EGR concept is investigated, which appears an attractive possibility due to its simple implementation and the potential of reducing NO_x emissions to the required levels of ECAs. Differently from the, more common, High Pressure (HP) EGR system, the LP system recirculates the exhaust gas after the turbine to the inlet of the compressor, at low pressure levels. This system is new and not yet studied in detail.

As is well established, see for example Raptotasios et al. [7], EGR

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decreases NO_x emissions because of the increase of the charge specific heat capacity, which leads to a reduction of the in-cylinder temperature, as well as due to the decreased oxygen concentration, which slows down the rate of NO_x producing reactions. However, at the same time, EGR leads to an increase in fuel consumption (Brake Specific Fuel Consumption, BSFC, commonly expressed in g/kWh), i.e. to decreased engine efficiency, as well as to increased soot formation rates. In this context, the combined use of EGR and waste heat recovery systems, such as ORC (Organic Rankine Cycle) [8], seems to be an effective way to develop engines which can be both clean and efficient.

Waste heat recovery from two-stroke engines is already considered by the shipping industry, as reported in Shu et al. [9]. However, the penetration of technologies as ORC in the sector is rather low, despite the good potential envisaged for low and medium temperature heat recovery [10].

The Organic Rankine Cycle (ORC) is very similar to the common steam Rankine cycle, with the difference that, instead of using water steam, an organic fluid, as for example a refrigerant or a hydrocarbon, is usually used to recover heat at lower temperatures. Good overviews on ORC technology and possible applications, with an emphasis on internal combustion engine waste heat recovery, can be found in Refs. [8,11].

While the research literature on ORC systems for internal combustion engine waste heat recovery is by now rather extensive, few works are available for systems on-board ships, since the application is still not well spread in the market, despite its potential. Moreover, most of the published works are related to heat recovery from four-stroke internal combustion engines for auxiliary power generation, while only a few publications and applications are related to two-stroke ship propulsion units.

The first ORC installed on-board ships has been used to recover heat from the engines of a car-truck carrier ship, as reported by Öhman et al. [12], using an OPCON/Powerbox [13] unit, running with R236fa as working fluid (now banned due to its high environmental impact), engine cooling water as heat source and LT cooling water as heat sink, expecting a 4−6% fuel saving.

Burel et al. [14] analysed the possibility to install an ORC in a tanker where Liquefied Natural Gas (LNG) is used as propulsion fuel, while Larsen et al. [15] proposed a generally applicable methodology based on natural selection principles, to optimise working fluid selection, boiler pressure and Rankine cycle process for marine engine heat recovery.

Larsen et al. [16] proposed a comparison of advanced heat recovery power cycles for large ships, modelling the systems in MATLAB environment, using a genetic algorithm for the optimization procedure; they concluded that a Kalina cycle has no significant advantages on the ORC and steam Rankine systems.

Bonafin et al. [17] performed a study on recovering waste heat from the exhaust gas of a marine dual-fuel engine with power output of 5.7 MW. The working fluid selected was toluene and a simple cycle architecture was considered as the most interesting one in terms of increased power output benefits. An economic analysis has been also reported in Ref. [17].

Baldi & Gabrielii [18] and Baldi et al. [19] proposed the use of optimization techniques for Diesel engine-ORC waste heat recovery systems based on the analysis of typical ship operating profiles. The case studies use, as baseline engines, four-stroke Diesel engines with a power output of 3840 kW and some auxiliary units of 683 kW. Fuel savings potential is discussed for typical vessels' applications.

Song et al. [20] studied the waste heat recovery potential of an ORC to recover heat from the cooling water and the exhaust gas of a medium speed 996 kW marine Diesel engine. The analysis has considered off-design conditions, as well as economic evaluations.

An optimized system using cyclopentane, cooling water as pre-heating source and exhaust gas as evaporating source for the working medium was proposed, obtaining a power output only around 1.4% lower in comparison to the separated bulkier systems.

Yun et al. [21] performed a study of a dual-loop ORC system, with the aim of recovering in parallel waste heat from the exhaust gas of marine Diesel engines, with the highlighted benefit of being more versatile when operating at off-design conditions. They concluded that the dual loop ORC has a power output which is between 3% and 15% higher than a simple single-loop system.

Yfantis et al. [22] considered a four-stroke marine Diesel engine equipped with a Regenerative Organic Rankine Cycle (RORC) to recover exhaust gas heat; they proposed a thermodynamic model to study performance characteristics in the context of the first and second law of Thermodynamics. Different engine operating loads were investigated, with R245fa, R245ca, isobutane and R123 as working fluids. It was found that a subcritical and saturated vapour regenerative cycle has the best performance, both from a first and a second law point of view.

Soffiato et al. [23] proposed an ORC system to recover heat from the cooling jacket water, lubricating oil and CAC of the engines of a LNG carrier ship; exhaust gas has been still used to generate steam. Simple, regenerative and two-stage evaporation architectures have been analysed, obtaining a maximum net power output of 820 kW, achieved using the two-stage architecture, exhibiting double the potential of other architectures, but with higher structural complexity and reliability issues.

Beyene et al. [24] and Sciubba et al. [25] performed a comparative study of a single loop and a dual loop waste heat recovery system for marine engines, in a wide power range (a yacht suitable non-supercharged 300 kW engine and a ship turbocharged 12.6 MW engine), using R245fa and R600 for the secondary recovery loop and water-steam for the primary loop, recovering engine exhaust gas and HT cooling water heat. Regeneration was also shown to improve system efficiency. Simulation results show improved electric power outputs of 8.11% and 2.67% for the small and the large engine, respectively.

Michos et al. [26] analysed the engine fuel consumption effect of fitting an ORC boiler on the exhaust line of a turbocharged V12 engine used for marine auxiliary power generation. Different turbocharging strategies, such as Waste-Gate (WG) and Variable Geometry Turbine (VGT), have been investigated in order to counterbalance the detrimental effect of the increase in exhaust line backpressure. Simple and recuperated ORC architectures have been investigated, using simulation, in order to assess the combined engine-ORC fuel economy improvement. A combined engine-ORC system using VGT turbocharger and acetone as ORC working fluid has been considered the most promising, leading to a possible improvement in fuel efficiency between 9.1 and 10.2%, depending on the ORC boiler engine backpressure.

Two-stroke ship propulsion units have also been considered in waste heat recovery studies, even though a smaller number of works has been published. Hountalas et al. [27] presented a theoretical study on a two-stroke 16.6 MW marine Diesel engine equipped with a Rankine cycle to evaluate the potential benefits for fuel consumption using a simulation model. Exhaust gas and SAC heat sources have been assessed, and a comparison performed between the use of steam and R245ca, obtaining 4.63−4.85% and 5.0−5.2% SFOC improvement, respectively. Pressure drop increase on the gas sides was also considered in Ref. [27].

Choi and Kim [28] analysed the theoretical performance of a dual-loop ORC with trilateral cycle applied to the exhaust gases of a two-stroke propulsion unit for a 6800 TEU container ship, using water in the high pressure loop and R1234yf for the low pressure loop, obtaining a net power output of 2069.8 kW, with a maximum

efficiency of 10.93% and a 6% fuel economy during actual operations.

Yang and Yeh [29] analysed the possibility of recovering jacket cooling heat of a large marine Diesel engine. Results show that R600a performs best, followed by R1234ze, R1234yf, R245fa, R245ca and R1233zd, at low evaporation temperature (58 °C–68 °C).

Wang et al. [30] simulated and analysed a combined ORC-desalination system driven by the SAC heat of a two-stroke engine, using R245fa as a working fluid for the ORC, and obtained up to almost 2800 kW and 245 t/day of desalinated water.

Grljusic et al. [31,32] proposed a supercritical ORC system operating with R123 or R245fa, to recover heat from scavenge air cooler (SAC), jacket cooling water and exhaust gas of a two-stroke 18660 kW propulsion unit for a Suezmax oil tanker, concluding that the system can supply, at full load, enough electrical power for ship requirements, while at part load some additional fuel must be burned in order to reach the power target.

Yuksek and Mirmobin [33] proposed an ORC system (Hydro-current™ 125EJW ORC) to recover two-stroke jacket cooling water heat at 80–95 °C, using R245fa as working fluid, sea water as cooling medium and a turbo-expander; they obtained around 120 kW net power output at design conditions, with a net efficiency of 6.5% and a declared turbine isentropic efficiency of 90%.

Yang [34] evaluated the economic performance of a Transcritical Rankine Cycle (TRC) using different low temperature suitable working fluids (R1234yf, R1234ze, R134a, R152a, R236fa and R290). The heat sources considered were exhaust gas, cylinder HT cooling jacket, scavenge air and lube oil. Best results, corresponding to the lowest levelized energy cost, were obtained for R236fa. Payback period, fuel oil saving, and CO_2 emission reduction were also evaluated.

Larsen et al. [35] also proposed a new concept of an ORC system, aiming at reducing the cost of the bottoming cycle installation using one of the cylinders of the engine for the expansion process. Numerical models have been used in order to assess the maximum power output of the proposed architecture, while 104 different fluids have been evaluated, obtaining best results for R245fa and R1234ze(z). The power output obtained from the ORC cylinder is declared to be similar to that obtainable from Diesel combustion, and an improvement of fuel economy of 8.3% has been deduced.

Andreasen et al. [36] proposed a comparison between organic and steam Rankine cycle for waste heat recovery on large ships, obtaining, from a process simulation campaign, better results with steam as working fluid at high load conditions, and better results with an organic medium at low load points. A turbine type expansion machine has been considered in this study, and some preliminary design considerations have been proposed.

Kyriakidis et al. [37] proposed a work considering the analysis of a steam Rankine cycle, with several pressure levels, applied to a two-stroke ship engine, also considering a high pressure (HP) EGR system on the engine side. They focused the analysis mostly on the ORC system, obtaining 1577 kW net ORC power with two pressure levels and 1641 kW with three pressure levels.

The present paper aims to advance the understanding of waste heat recovery from large two-stroke marine Diesel engines, by means of ORC systems. The analysis considers different architectures and working fluids, as well as the combined use of ORC and LP (Low Pressure) EGR (Exhaust Gas Recirculation) on the engine side to decrease NO_x emissions in ECAs.

To the authors' knowledge, at present, there exists no commercialized LP EGR system for large two-stroke marine Diesel engines. First implementations include a system developed by Japan Engine Corporation [38], and one by Mitsubishi Heavy Industries tested in a 34000 DWT Bulk Carrier [39,40].

Moreover, no literature studies are available, on the performance and modelling of two-stroke marine engines in the presence of both innovative LP EGR and waste hear recovery in terms of ORC. The present study thus addresses the issue of quantifying the advantages of such systems, in terms of both thermodynamic effectiveness and economic savings. To this end, detailed thermodynamic modelling is used, accounting for the principal possible implementations of the technology and for representative simplified ship operational profiles.

The paper is organized as follows. In Section 2, the present modelling is outlined. In Section 3, computational results are presented for the basic operation of the engine (without and with EGR), as well as for operation in the presence of an ORC system. Results of a preliminary economic analysis, quantifying fuel saving costs for a representative case of a ship propulsion system using ORC for heat recovery, are also included. The main findings are summarized in Section 4.

2. Modelling and methodology

The analysis proposed in this work encompasses both the engine and the ORC systems and considers engine operational conditions relevant in terms of both the Tier II and Tier III IMO regulations (without and with the implementation of the LP EGR system). In the following sections, the methodology implemented for the thermodynamic analysis is presented.

2.1. Engine modelling

The engine side analysis has been performed by developing a model using the engine performance simulation software Ricardo WAVE [41]. The engine studied is a WinGD RT-flex58T, 6-cylinder, 13.6 MW brake power, two-stroke, low speed marine Diesel engine.

The baseline engine model, without EGR, has been developed and validated based on the data supplied by WinGD (Winterthur Gas & Diesel) [42], specialized in the development of two-stroke marine engines. A sketch of the baseline engine is presented in Fig. 2a, including the main system components. The main engine dimensions, the cylinder pressure and burning rate profiles, the Friction Mean Effective Pressure (FMEP) and the pressures and temperatures on the gas lines have been supplied in order to develop and validate a model consistent with the actual engine developed by WinGD; the data are not fully reported in the paper, for reasons associated with intellectual property rights.

After developing and validating the baseline model, a model with LP EGR has been implemented, corresponding to the sketch of Fig. 2b. Here, exhaust gas after the economizer is recirculated to the inlet of the compressor, utilizing a blower of imposed fixed efficiency (60%) and a scrubber; the latter is approximated in the present model as a simple heat exchanger, i.e. water injection is not considered. Pressure and temperature drop in the EGR line have been computed utilizing the experience and corresponding guidelines by WinGD. EGR rates between 20% and 40% have been considered at the different engine load points simulated.

The temperature of the scavenge air at the cylinders' inlets has been kept fixed to values suggested by WinGD (30–35 °C), thus determining the heat rejection in the Scavenge Air Cooler (SAC), both during baseline and EGR operations; this is mainly dictated by combustion requirements. A generally increased heat rejection is expected for EGR operation, due to the increased temperatures of recirculated exhaust gas lines, which are thus beneficial for waste heat recovery.

Three main engine load points have been simulated: (a) 100%, at 105 rpm, (b) 75%, at 95 rpm, and (c) 50%, at 83 rpm; these points are representative of engine operation at full and medium-slow
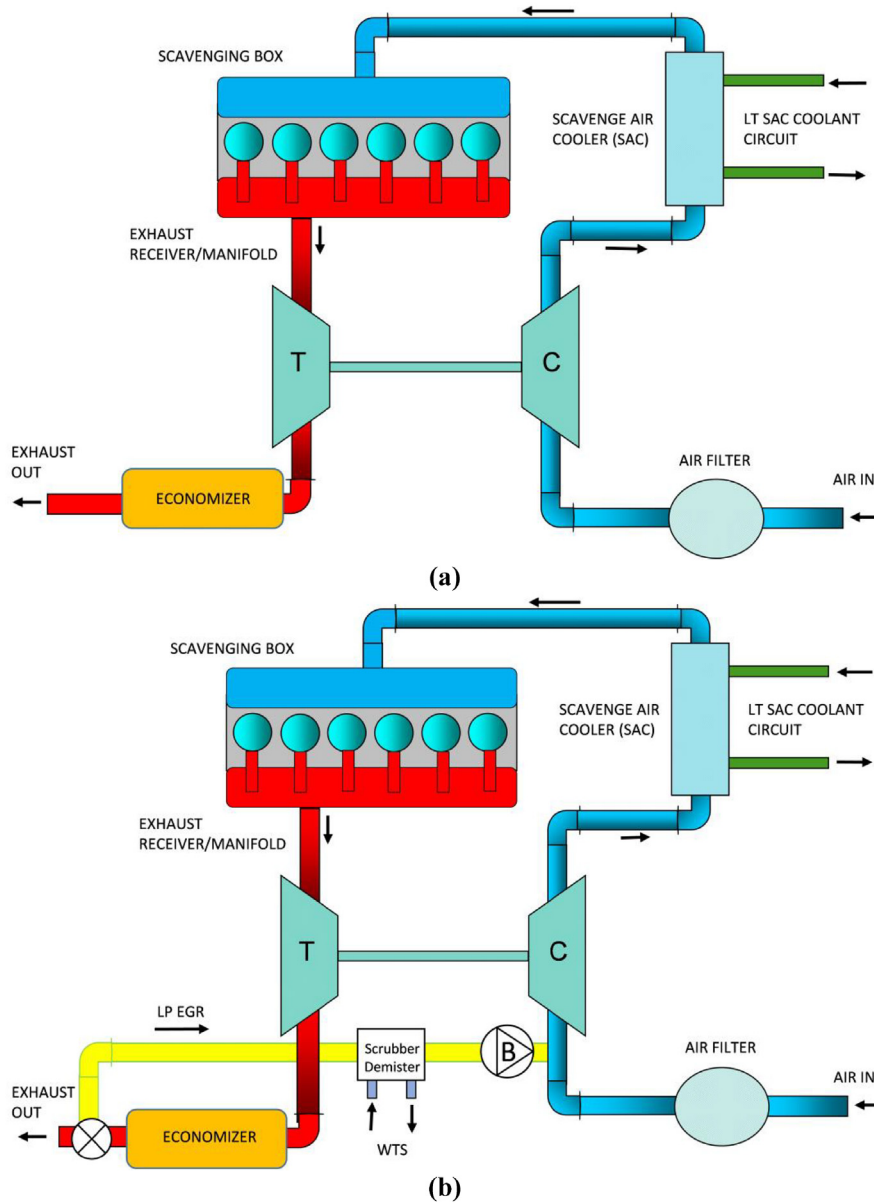
Fig. 2. Sketch of: (a) baseline engine model, (b) engine model with LP EGR.

steaming.

2.2. ORC modelling

Regarding the ORC side thermodynamic analysis of the present study, different process simulation models have been developed, for the considered cycle architectures, using Engineering Equation Solver (EES) [43]. The models have been developed based on mass and energy balances for the main ORC components: expander (EXP), pump (P), evaporator (EVAP) and condenser (COND). A more detailed description of the model foundations can be found in recent works of the present research team [26,44].

Two ORC scenarios have been evaluated, recovering the waste heat of different potential sources of the engine: Scavenge Air Cooler (SAC), exhaust gas (economizer) and high temperature (HT) engine jacket cooling water. The thermodynamic boundary conditions for these heat sources, which are part of the main inputs of the ORC models, have been calculated from the engine thermodynamic

model, both with and without LP EGR (Tier III and Tier II operation, respectively).

Two operational scenarios have been considered (demonstrative sketches are presented in Fig. 3):

- *Scenario 1*: simple ORC architecture recovering exhaust gas from the economizer (concept 1), and simple ORC architecture recovering heat from the HT jacket cooling water and SAC in series, in a so-called two-stage SAC cooling system (concept 4);
- *Scenario 2*: parallel ORC architecture recovering exhaust gas and SAC heat (concept 2), and simple ORC architecture recovering heat only from the HT jacket cooling water (concept 3).

Concept 4, recovering HT engine jacket coolant and SAC heat together, has been considered as a very promising one in terms of thermodynamic performance, due to the high amount of heat available in the cooling water and SAC, even at low-medium temperatures. The concept is also easy to implement and safe, due to

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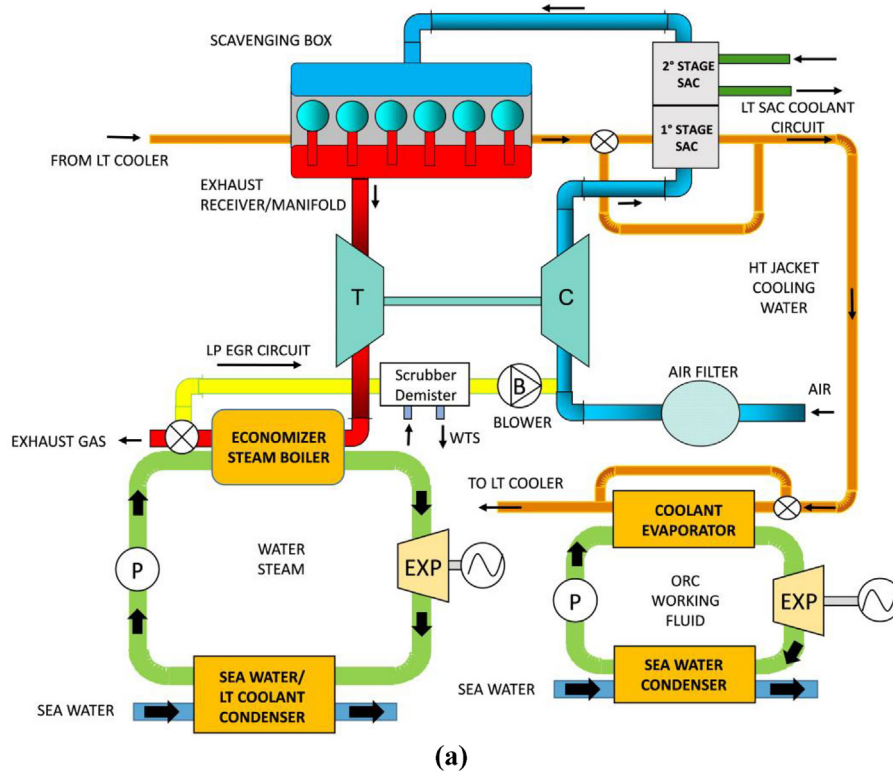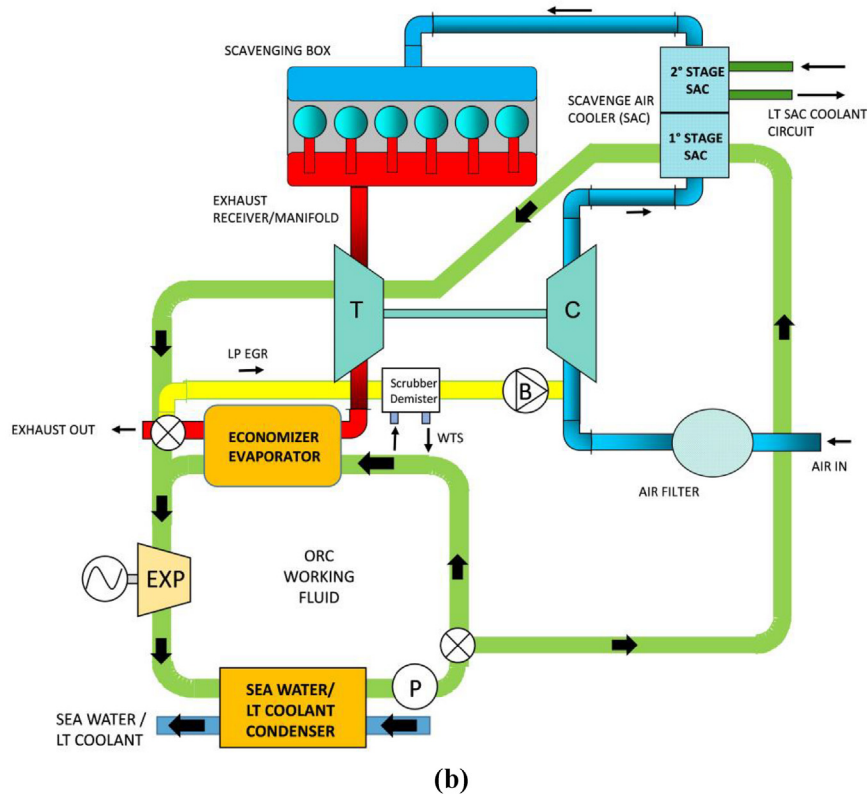
Fig. 3. Combined engine-ORC architectures: (a) scenario 1, and (b) scenario 2. In the case of scenario 2, only the parallel ORC layout (exhaust gas and SAC) has been reported.

the interposition of the cooling water circuit between the engine gas line and the ORC working fluid.

The combined engine-ORC architecture for scenario 1 is presented in Fig. 3a, while for scenario 2, only concept 2 has been reported in Fig. 3b, while concept 3 (only HT coolant heat recovery) has been omitted, for brevity.

In concepts 2 and 4, SAC has been split into two stages, the first one to recover heat through the ORC, and the second one to ensure

Table 1
Model and optimization procedure variables, constraints and assumptions.

Independent optimization variables	
$\dot{m}_{wf}$ (kg/s)	ORC working fluid mass flow
p_{cond} (bar)	ORC condensation pressure
$PR = p_{evap}/p_{cond}$	ORC pressure ratio (p_{evap}/p_{cond})
ΔT_{suph} (°C)	ORC degree of superheating
Constraints and Assumptions	
ΔT_{PP} (°C)	Minimum pinch-point temperature difference for the heat exchangers
	-Concepts 1–2: 10 °C
	-Concepts 3–4: 5 °C
$0.1\ (°C) \leq \Delta T_{suph} \leq 100\ (°C)$	Superheating degree (high superheating generally required for water systems and wet fluids, to avoid high liquid fraction at expansion outlet)
$p_{evap} \leq 30$ (bar)	Maximum evaporation pressure
$p_{cond} \geq 0.1$ (bar)	Minimum condensation pressure. Vacuum accepted in order to increase the power output of the ORC (especially for water systems in concepts 1 and 2)
$T_{exh,out} = T_{econ,,out}$ (°C)	Minimum exhaust gas outlet temperature at the economizer outlet (from WAVE boundary conditions, usually around 180 °C). Economizer heat rejection capabilities respected
$T_{cool,evap,out} = 75$ (°C)	Minimum temperature of the coolant at the evaporator outlet (concepts 3–4). Temperature requirement for engine cooling jacket inlet respected [45]
$x_{EXP,out} \geq 0.9$	Minimum expander outlet vapour quality to avoid high liquid fraction during the expansion process
$T_{max} \leq T_c$ (°C)	Cycle maximum temperature (expander inlet) lower than the fluid critical temperature (no-supercritical conditions evaluated)
$PR \leq 100$	Maximum pressure ratio (high values suitable for multi-stage turbo-expanders)
$\dot{m}_{cf}$ (kg/s)	Cooling fluid mass flow:
	-Concepts 1–2: 20 kg/s of sea water
	-Concept 3: 30 kg/s of sea water
	-Concept 4: 120 kg/s of sea water
$T_{cf,cond,in} = 25$ (°C)	Condenser sea water (cooling fluid) inlet temperature
$T_{cf,cond,out} \leq 50$ (°C)	Condenser sea water (cooling fluid) outlet temperature
$\Delta T_{sub-cool} = 2$ (°C)	Sub-cooling level, to ensure liquid state of fluid at the pump inlet, to avoid cavitation problems
$\eta_{EXP} = 0.7$	Expander isentropic efficiency (turbo-expander assumed)
$\eta_P = 0.6$	Pump isentropic efficiency
Objective	
$\dot{W}_{ORC,net} = \dot{W}_{EXP} - \dot{W}_P$	Maximize the ORC net power output [kW], which in turn minimizes the combined system BSFC
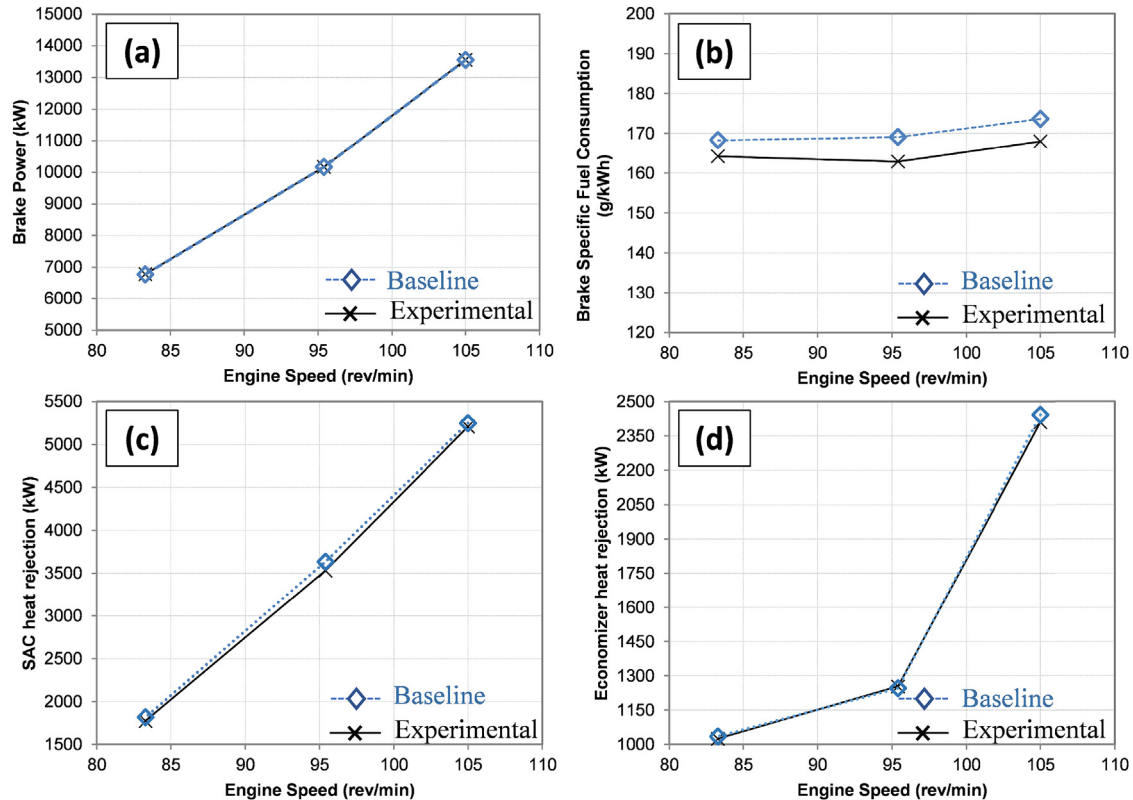
Fig. 4. Validation of the baseline engine model against experimental data, for different values of engine speed: (a) brake power, (b) BSFC, (c) SAC heat rejection, and (d) economizer heat rejection.
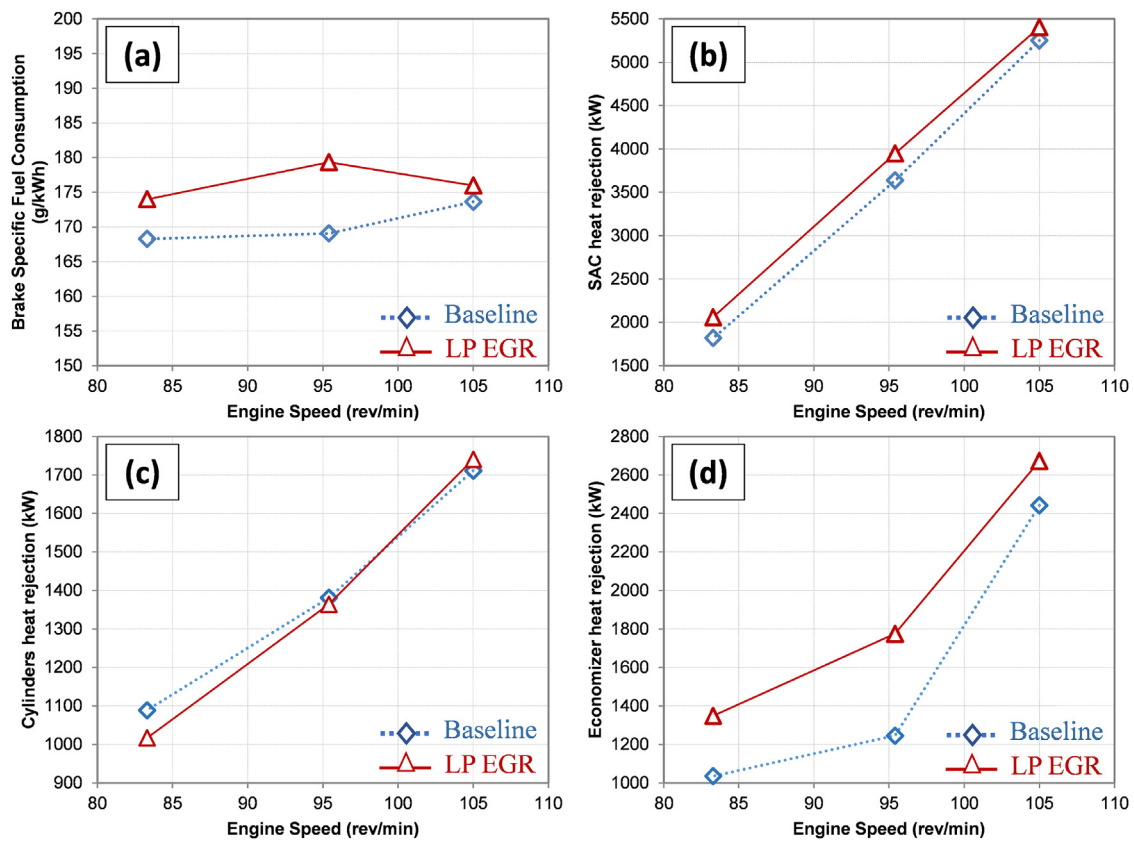
Fig. 5. Simulation model with LP EGR and baseline engine model comparison, for different values of engine speed: (a) BSFC, (b) SAC heat rejection, (c) cylinders heat rejection, and (d) economizer heat rejection.

adequate gas cooling before the cylinders' inlet.

An optimization procedure has been carried out, in terms of a single-objective constrained optimization for maximizing the ORC net power output. This, in turn, corresponds to a maximum reduction of BSFC in the combined engine-ORC process, as the mechanical power produced can be re-introduced into the engine crankshaft (with an assumed mechanical efficiency of about 98%, considering the use of a reduction gearbox).

The optimization has been performed in two steps, using first a genetic algorithm to approximate the global best solution, and, subsequently, refining the solution using a Nelder-Mead Simplex algorithm on local scale. Both algorithms are available in the EES professional version used.

The cooling sea water pump energy consumption has not been estimated in this study; however it should be considered in a complete design of the system.

Regarding the ORC working fluids, for concepts 1 and 2, water has been considered for medium temperature heat recovery. For concepts 3 and 4, R1233zd(E) has been considered, as it is a natural replacement of the most common R245fa refrigerant fluid, suitable for lower temperature heat recovery, as proposed for the engine cooling system. The properties of the fluids have been retrieved from EES internal database.

The boundary conditions and assumptions for the optimization are reported in Table 1.

3. Results and discussion

In the next sections, the main results of the present study are summarized, considering both the engine side, without LP EGR
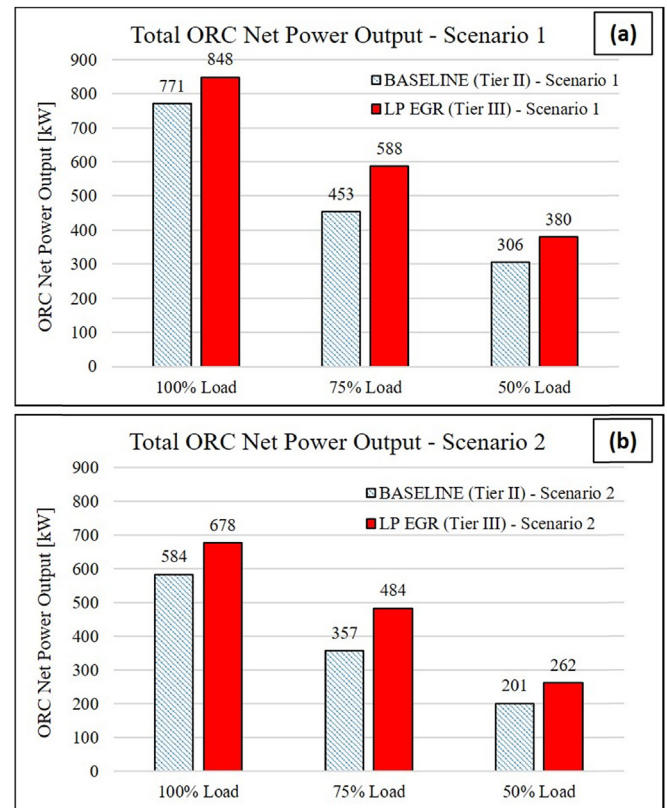

Fig. 6. ORC net power output for the two scenarios, for the baseline and LP EGR engine, for the three-engine operating loads considered.
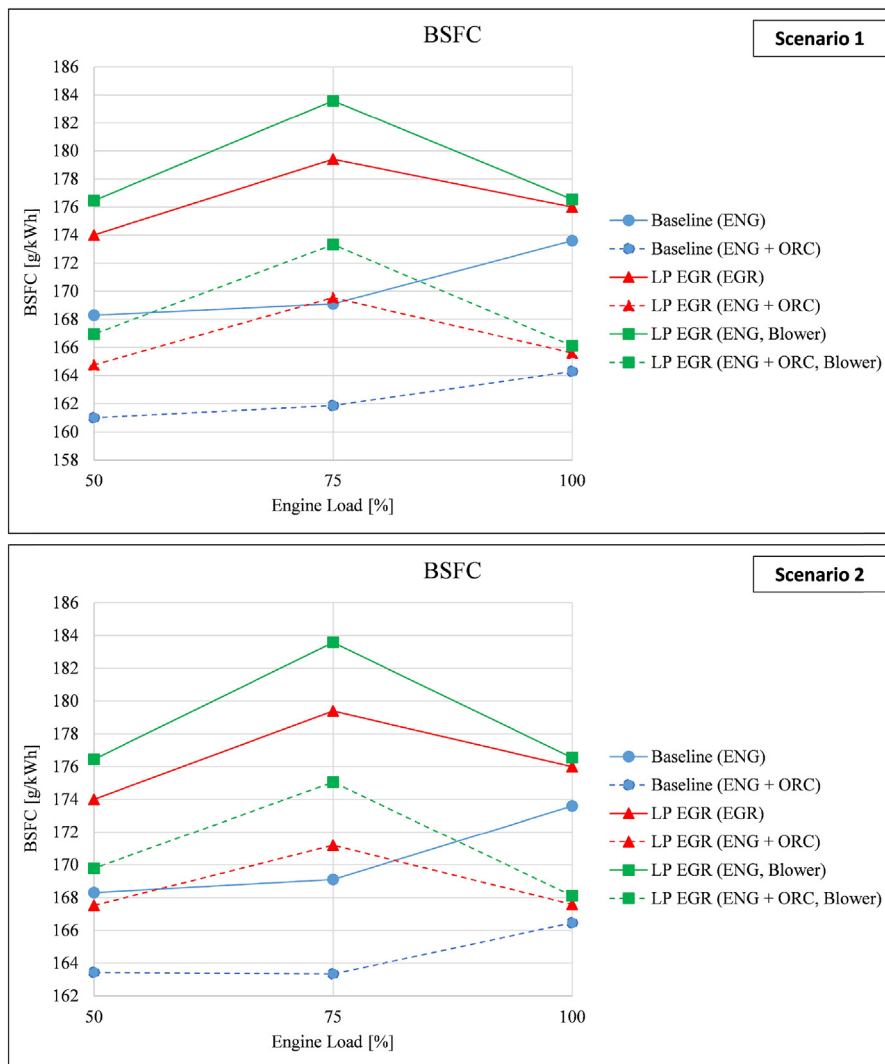
Fig. 7. Engine and combined engine-ORC calculated BSFC for different engine load, for the two considered scenarios.

(Tier II) and with LP EGR (Tier III), as well as the combined engine-ORC BSFC results, in order to demonstrate the positive effect of recovering the engine wasted heat, via an ORC, on the overall system performance. A preliminary economic analysis is also reported, quantifying the annual fuel operational cost savings associated with adding an ORC system to the propulsion unit.

3.1. Engine side

First, a validation of the baseline engine model, without EGR, has been carried out, considering an accuracy of ±5% compared to the supplied experimental data. In Fig. 4, a selection of the validated parameters for the baseline engine is reported.

In a second step, the thermodynamic performance of the engine with LP EGR was calculated, with the recirculation circuit added to the baseline model. In order to keep combustion parameters in line with the experience of the engine manufacturer, as well as to avoid having too low Air-Fuel-Ratio (AFR) inside the cylinders (with a consequent possibility of increased soot formation), the model has been tuned with proper exhaust valve closing and start of injection events; details are not reported here because the corresponding parameter values consist sensitive information of the WinGD design. A more detailed CFD analysis is underway and will be presented in an additional publication.

The results concerning the engine performance and heat rejection, with LP EGR, are presented in Fig. 5. It is noted that, in the

Table 2
Scenario 1: BSFC variation for different operation modes with respect to a reference mode.

ΔBSFC (%)			
Engine Load	LP EGR vs Baseline*	Baseline + ORC vs Baseline	LP EGR + ORC vs LP EGR
100%	+1.3	- 5.4	- 5.9
75%	+6.1	- 4.3	- 5.5
50%	+3.4	- 4.3	- 5.3

Table 3

Scenario 2: BSFC variation for different operation modes with respect to a reference mode.

	ΔBSFC (%)		
Engine Load	**LP EGR** vs **Baseline**[a]	**Baseline + ORC** vs **Baseline**	**LP EGR + ORC** vs **LP EGR**
100%	+1.3	- 4.1	- 4.8
75%	+6.1	- 3.4	- 4.6
50%	+3.4	- 2.9	- 3.7

[a] Calculated from Ricardo WAVE simulations.

presence of EGR, the brake power output has been targeted to be the same as the baseline engine, by means of a PID controller in the model, so as to avoid a reduction in engine performance and keep the comparison meaningful.

The proposed results verify an increase in fuel consumption (BSFC) in the presence of EGR. After tuning the present model, the calculated increase in BSFC was in the range between 1.3% (100% load) and 6.1% (75% load). It is also noted that, in the presence of EGR, an increased heat rejection, mostly in the SAC and in the economizer, is evident, which further motivates the use of waste heat recovery systems, such as ORC.

3.2. Combined Engine-ORC system

In order to overcome the drawback of increased fuel consumption due to the LP EGR, the two already proposed ORC scenarios (1 and 2), have been evaluated using the thermodynamic models developed; boundary conditions for the heat sources have been obtained from the engine simulations with and without EGR.

For the two scenarios considered, the ORC net power output, $\dot{W}_{ORC,net}$ [kW], and the combined systems' BSFC are presented in Fig. 6 and Fig. 7, respectively, both for Tier II and Tier III operation.

Scenario 1, with a traditional tailpipe water Rankine system and the innovative HT coolant-SAC lower temperature heat recovery system, is found to exhibit a better performance.

A general increase in ORC net power output can be evinced in all cases with LP EGR (Tier III operation), mostly due to the higher temperatures of the gas lines, which lead to an increase in heat which can be recovered, while also allowing for an increase in evaporation pressure in the ORC, thus increasing its performance.

The calculated BSFC values are reported in Fig. 7. Concerning the BSFC graph for scenario 1, it is possible to observe how, in absolute values, the use of waste heat recovery systems, with the layouts proposed, tends to almost completely mitigate the BSFC increase effect introduced with the use of LP EGR for Tier III operation. In particular, for load points 100% and 50%, the operation with LP EGR and ORC is even lower in terms of estimated BSFC, compared to the normal Tier II operation without ORC. For load point 75%, for which the proposed engine tuning leads to a more marked increase in BSFC with LP EGR, the use of ORC allows to reduce BSFC to a level comparable to Tier II operation without EGR.

The same trends are observed for scenario 2, with the difference that BSFC benefit introduced with the ORC is slightly lower compared to scenario 1; especially for the 75% load, the ORC

(a)

(b)
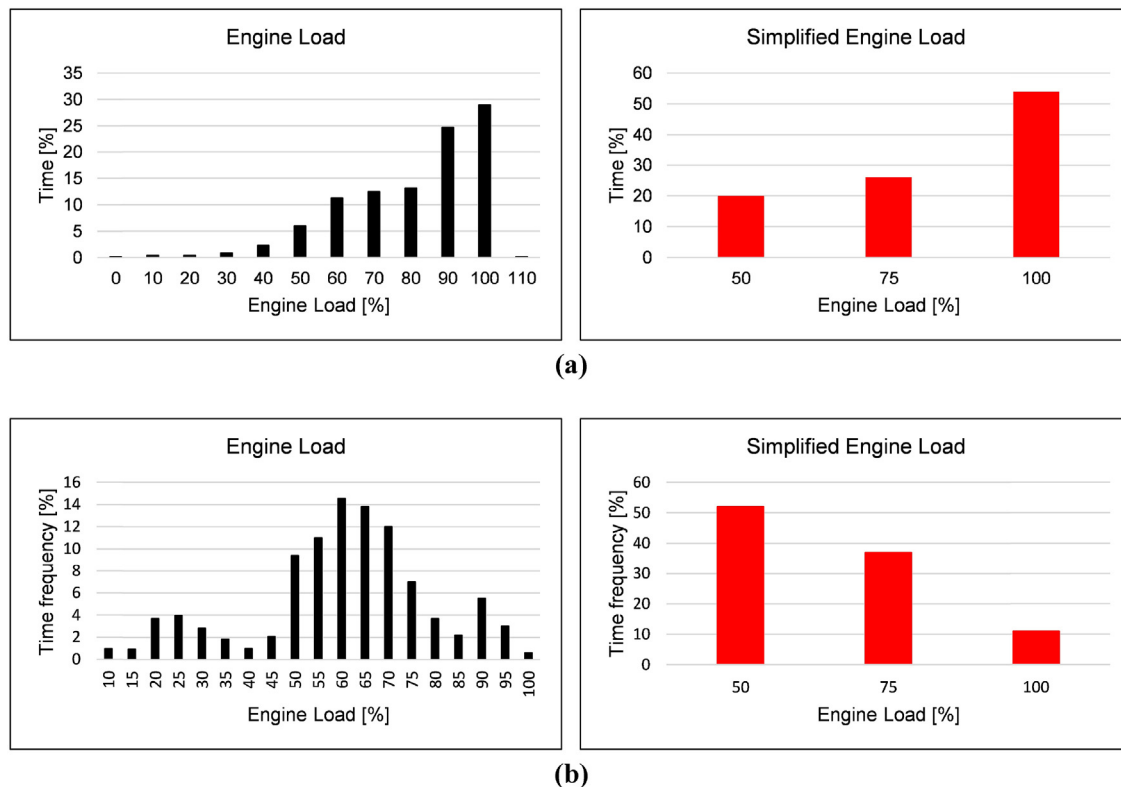
Fig. 8. Load operating profiles: (a) full steaming, and (b) slow steaming.

systems are not able to completely withstand the increase in BSFC due to EGR.

When comparing only the Tier II operation set-ups (baseline ENG, and baseline ENG + ORC), the use of ORCs allows to improve the overall BSFC in a quite marked way compared to the case without a heat recovery system.

It is noted that, in Fig. 7, the green continuous and segmented lines, with square markers, refer to the cases in which the EGR blower power consumption has also been estimated in the overall power balance; the benefits of using ORC are still evident, in particular for 100% and 50% load cases, but also for the 75% case, allowing consistent fuel savings.

The results of the present analysis regarding the BSFC reduction potential are also reported for the two scenarios in Table 2 and Table 3 in a tabulated representation.

3.3. Preliminary economic analysis — scenario 1

A preliminary estimation of the economic benefit of using the ORC systems, as per Scenario 1, has been carried out, consisting in the evaluation of fuel operational cost savings.

The estimation has been carried out on two simplified operating profiles: one representing typical full steaming operations [46], the other representing typical slow steaming operations [47].

Since just three engine operating points (100%, 75% and 50% loads) have been considered in the present analysis, the operating profiles used have been simplified, and are associated with the calculations of these three operating points. This evidently introduces an error; however, the approach is considered adequate for providing first estimates of cost savings. A more detailed analysis, associated with a higher number of operating points, can be considered in future work.

Computed BSFC values have been used as an input for the economic analysis. For all loads, calculations have been performed so as to yield the same brake power of a corresponding baseline case. This means that the additional ORC net produced power, re-introduced in the propeller line, is not considered, but only the estimated fuel savings derived from the use of the waste heat recovery systems are evaluated, even though the real effect would be the increase in power output, for the same fuel mass injected in the engine. This could lead to the choice of slightly reducing the engine load in order to save fuel, changing, however, the boundary conditions of the engine simulations.

Using the additional energy produced via the ORC to generate electricity could be more easily correlated to the savings in terms of fuel costs of on-board electric energy production through the main engine or auxiliary generators.

Since this is a first estimation, the accuracy at this step of the analysis has been considered sufficient in order to demonstrate the approach and the potential economic benefits. Future studies can consider the economic side in further detail, accounting for an overall ship energy management.

A vessel shipping time of 8144 h/year (almost 340 days) has been adopted in the present analysis (Burel et al. [14]), typical for a chemical tanker sailing 8 times/year from Dubai to Hamburg. Cases with 100% time spent outside ECAs (Tier II operations), 100% time inside ECAs (Tier III operations) as well as for a typical time percentage within ECAs have been considered (Burel et al. [14]).

The fuel prices for Marine Diesel Oil (MDO) and Heavy Fuel Oil (HFO, IFO 380) have been assumed to be 543 $/mt [48] and 339 $/mt, respectively, [49].

The operating profiles assumed in the present study are reported in Fig. 8 [46,47], together with the simplified ones, obtained by grouping different engine load steps in bigger intervals and summing up the contributions.

Regarding the residence within ECAs, the data of Burel et al. [14] have been used to identify useful information regarding ships powered by large low speed two-stroke Diesel propulsion units. Here, the representative case of a Handysize type tanker has been chosen, with an estimated sailing time spent in ECAs of 12.5%; this justifies the necessity of using NO$_x$ emission reduction technologies, such the LP EGR architecture investigated in the present study. Given the fuel costs considered, the calculated BSFC levels and the simplified operational profiles, the annual operational fuel costs have been calculated. Results are presented in Fig. 9 (full steaming) and Fig. 10 (slow steaming), for exclusive Tier II or Tier III operation, as well as for the case of 12.5% of sailing time spent in ECAs. The results reported concern both MDO and HFO, for systems without and with ORC.

The results presented in Figs. 9 and 10 verify substantially increased operation costs for MDO, compared to HFO; this is evidently a consequence of the higher level of MDO price. The large decrease in annual cost for slow steaming operation is also verified. Interestingly, the results show a significant decrease in annual fuel cost for operation with ORC, as compared to operation without ORC (up to 5.7% for full steaming, and up to 5.5% for slow steaming). Cost reduction with ORC is higher for operation in ECAs (Tier III), due to the good performance of ORC in recovering the increased levels of rejected heat associated with LP EGR. Nonetheless, for Tier II operation, use of ORC still gives cost reductions between 4% and 5% for slow and full steaming. As expected, for a 12.5% of sailing time in ECAs, cost saving values do not substantially deviate from those of Tier II operation.
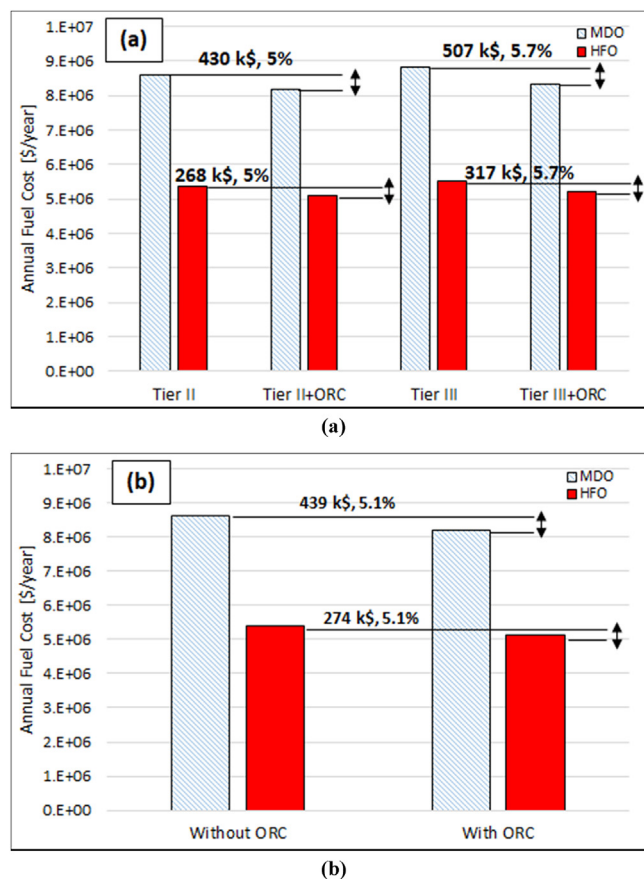

Fig. 9. Annual fuel costs for full steaming operation using HFO or MDO. Cost savings associated with using ORC are indicated. (a) Exclusive Tier II or Tier III operation, (b) 12.5% sailing time spent in ECAs.
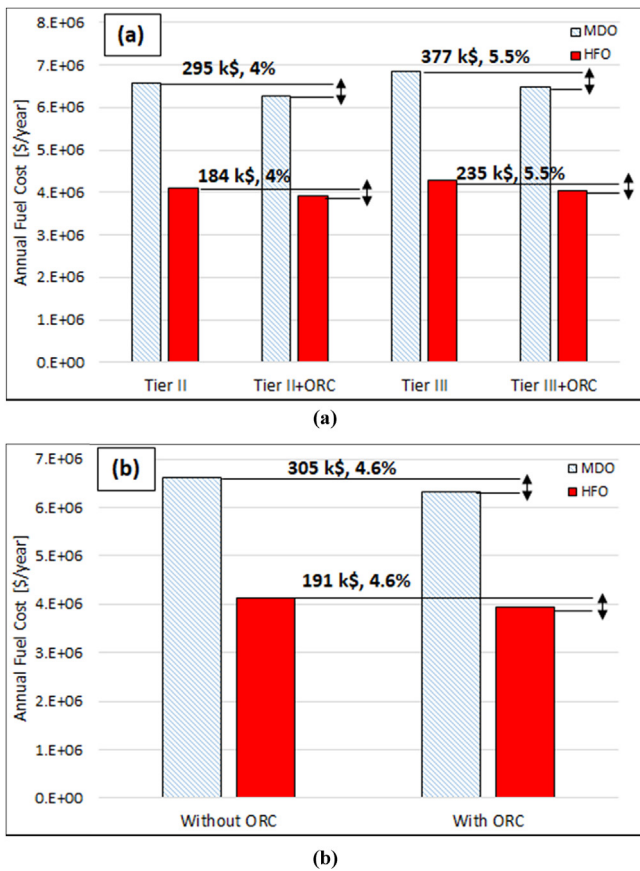
Fig. 10. Annual fuel costs for slow steaming operation using HFO or MDO. Cost savings associated with using ORC are indicated. (a) Exclusive Tier II or Tier III operation, (b) 12.5% sailing time spent in ECAs.

While the present preliminary economic analysis clearly demonstrates the potential for reduced fuel costs associated with the use of ORC systems in two-stroke low speed ship propulsion units, future work should include a more detailed approach for calculating cost savings. In this context, more detailed engine operational profiles should be considered, together with cost estimates associated with the acquisition, installation and maintenance of ORC systems, as well as the use of the additional available energy, towards an accurate evaluation of ORC system investments.

4. Conclusions

In the present study, the potential of introducing ORC systems in large two-stroke marine Diesel engine propulsion units has been assessed, both for Tier-II and Tier-III operation. To this end, a complete thermodynamic model of a two-stroke marine Diesel engine was first developed and validated for loads of 100%, 75% and 50%. Next, the model was extended in order to account for LP EGR. Increase in BSFC of a few percent for EGR operation was verified for the three operation points considered.

For operation using ORC, four different architectures have been assessed and divided in two combined scenarios, in order to recover most of the engine wasted heat, with the goal of minimizing fuel consumption, and maximizing ORC net power output. R1233zd(E) has been assumed the best choice for the proposed application and lower temperature heat sources, together with common water-steam for higher temperature heat recovery.

When combining the different ORC architectures, in order to try

to fully exploit the engine waste heat available, a combined system (scenario 1) with a water-steam Rankine cycle on the exhaust gas side and an ORC recovering HT jacket cooling water and SAC heat, in an innovative two-stage SAC configuration, has been estimated to bring 5.4% fuel economy benefit in Tier II operation, and 5.9% in Tier III (at full load), while at the same time keeping system complexity at a reasonable level. The results achieved show the possibility of mitigating the increase in fuel consumption effect of EGR operations through the use of waste heat recovery systems.

A preliminary economic analysis for the case of a representative tanker has shown that ORC can provide annual financial savings in fuel cost of about 5%.

Future work can consider improved combined engine-ORC systems' layouts, including, for example, dual loop and two-stage pressurization architectures, as well as other engine emission reduction strategies and technologies. Finally, more detailed investment evaluation studies of ORC systems are needed.

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