

A review of emissions reduction technologies for low and medium speed marine Diesel engines and their potential for waste heat recovery

Simone Lion^{a,*}, Ioannis Vlaskos^{b,1}, Rodolfo Taccani^c

^a University of Trieste, Department of Engineering and Architecture, Via A. Valerio 6/1, 34127, Trieste, Italy

^b Winterthur Gas & Diesel (WINGD), P.O. Box 414, Schützenstrasse 3, CH-8401, Winterthur, Switzerland

^c University of Trieste, Department of Engineering and Architecture, Via A. Valerio 6/1, 34127, Trieste, Italy

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ABSTRACT

Reducing emissions from internal combustion engines is becoming one of the most important tasks for engine manufacturers and transport regulatory organizations. In particular, the marine transportation sector is one of the most polluting, due to the intense maritime activity and the use of low-quality fuels, burned in Heavy Duty Diesel Engines, for ship propulsion and auxiliary power generation. In order to reduce the global shipping environmental impact, the IMO (International Maritime Organization) is restricting NO_x and SO_x ships' emissions through the introduction of the IMO Tier III legislation, which requires to consider a wide spectrum of emissions reduction technologies and strategies, which are going to have an impact on the engine performance and fuel consumption.

In this work, the main solutions being currently developed or adopted for low and medium speed Diesel engines have been reviewed from a qualitative, and sometimes quantitative, point of view, but, in comparison to previous literature, focusing more on their potential with respect to possible waste heat recovery systems utilization, such as, in particular, steam Rankine cycles and Organic Rankine Cycles (ORC). Indeed, even though many of the considered emissions mitigation technologies lead to a certain amount of penalty in fuel economy, the use of waste heat recovery systems to recover wasted engines energy could become interesting in order to develop more efficient but, at the same time, cleaner engines.

1. Introduction

Heavy Duty Diesel Engines are widely used in several applications, such as power generation, commercial vehicles and ship propulsion. However, they are also among the main contributors to Greenhouse Gases and other pollutants emissions, such as CO₂, NO_x, SO_x and Particulate Matter (PM) as can be observed from the data reported in the Third IMO Greenhouse Gas Study (2014) [1], which demonstrates also the marine shipping sector being stable around 2–3% regarding the percentage of the total emitted CO₂ and GHG, as shown in Tables 1 and 2.

Moreover, the Third IMO Greenhouse Gas Study (2014) [1] reports also the data regarding NO_x and SO_x emissions (as well as other pollutants not reported in this work) in a period between 2007 and 2012, showing a tendency to reduction of these two typical harmful compounds, due also to the implementation of more effective technologies, and to the introduction of cleaner fuels with lower amount of sulphur compounds (Fig. 1).

The study reports also the fuel consumption and CO₂ emissions estimated for different types of ships used in the marine shipping sector in

2012. The data have been reported, after elaboration, in Fig. 2. From the qualitative analysis of the statistics, it is possible to evince how the most CO₂ polluting and fuel consuming ships are: container ships, bulk carriers, oil tankers, general cargo ships and chemical tankers. The interesting outcome of this assessment is the observation that these types of ships are typically powered by low speed two-stroke propulsion units, which are also one of the main prime movers in the marine sector. Some of the emissions and fuel consumption can come partially also from typical four-stroke auxiliary power units, used in ships to supply power for internal use or for diesel-electric propulsion architectures.

From all these considerations, it can be inferred how it is essential to implement and develop new efficient emission reduction technologies in order to reduce emitted pollutants, as well as new technologies that could improve the ship and engine fuel efficiency. Between these last mentioned, waste heat recovery systems, already developed and applied in industrial stationary power generation applications, will have a predominant role in the very next future of the shipping industry.

* Corresponding author at: Department of Engineering and Architecture, Via A. Valerio 6/1, 34127, Trieste, Italy.

E-mail addresses: simone.lion@phd.units.it, simone_lion@hotmail.com (S. Lion), Ioannis.vlaskos@wingd.com (I. Vlaskos), taccani@units.it (R. Taccani).

¹ Previously Ricardo Deutschland GmbH.

Nomenclature

K	Propeller Law Constant
$\dot{m}_{f,sw}$	Sea water mass flow (ORC cooling), kg/s
$\dot{m}_{f,wf}$	ORC working fluid mass flow, kg/s
N	Engine speed, rpm
P_{cond}	ORC condensing pressure, bar
P_{ratio}	ORC pump pressure ratio
\dot{W}	Power, kW
$\dot{W}_{ORC,net}$	ORC net electric power output, kW
ΔT_{suph}	ORC superheating degree, °C
η_{ORC}	ORC thermal efficiency, %

Acronyms

AFR	Air-Fuel Ratio
AIG	Ammonia Injector Grid
AT	After Turbine
BDC	Bottom Dead Centre
BSFC	Brake Specific Fuel Consumption, g/kWh
CAC	Charge Air Cooler
CAD	Crank Angle Degrees
CAPEX	Capital Expenditure
DCU	Decomposition Unit
DME	Dimethyl Ether
DWI	Direct Water Injection
ECA	Emission Control Area
EEDI	Energy Efficiency Design Index
EGB	Exhaust Gas Bypass
EGR	Exhaust Gas Recirculation
EPA	Environmental Protection Agency
EVC	Exhaust Valve Closure
GHG	Green House Gases
GWP	Global Warming Potential
HDDE	Heavy Duty Diesel Engine
HAM	Humid Air Motor

HFO	Heavy Fuel Oil
HPT	High Pressure Turbine
HT	High Temperature
IMO	International Maritime Organization
IVC	Inlet Valve Closure
LNG	Liquefied Natural Gas
LPG	Liquefied Petroleum Gas
LPT	Low Pressure Turbine
LT	Low Temperature
MCR	Maximum Continuous Rating
MDO	Marine Diesel Oil
MEP	Marine Environmental Protection Committee
MHI	Mitsubishi Heavy Industries
ODP	Ozone Depletion potential
OPEX	Operating Expenditure
ORC	Organic Rankine Cycle
PM	Particulate Matter
PT	Power Turbine
SAC	Scavenge Air Cooler
SAM	Scavenge Air Moisturizing
SCR	Selective Catalytic Reduction
SEEMP	Ship Energy Efficiency Management Plan
SFOC	Specific Fuel Oil Consumption, g/kWh
SOI	Start of Injection
ST	Steam Turbine
TDC	Top Dead Centre
TEG	Thermos-Electric-Generation
US	United States
VGT	Variable Geometry Turbine
WFE	Water-in-Fuel Emulsion
WHR	Waste Heat Recovery
WG	Waste-Gate
WTS	Water Treatment System
WINGD	Winterthur Gas & Diesel

The International Maritime Organization adopted the MARPOL Annex VI first in 1997 [2], in order to limit the main air pollutants contained in ships exhaust gas and, in particular, SO_x, NO_x and PM. In 2005, the Marine Environmental Protection Committee (MEPC), decided to revise the MARPOL Annex VI in order to strengthen the emission limits, considering future technological improvements. In 2008, finally, the revised MARPOL Annex VI entered into force. Emission Control Areas (ECAs) and IMO Tier II and Tier III emission limits have been proposed [3,4].

ECAs 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 are planned to be established in the next future, with particular interest to Mexican Caribbean Sea, Mediterranean Sea and Japan.

Table 1

CO₂ shipping emissions statistics (compared to global CO₂, reported in millions of tonnes CO₂) [1].

Year	Global CO ₂	Total Shipping CO ₂	% of Global	International Shipping CO ₂	% of global
2007	31,409	1100	3.5	885	2.8
2008	32,204	1135	3.5	921	2.9
2009	32,047	978	3.1	855	2.7
2010	33,612	915	2.7	771	2.3
2011	34,723	1022	2.9	850	2.4
2012	35,640	938	2.6	796	2.2
Average	33,273	1015	3.1	846	2.6

Table 2

GHG shipping emissions statistics (compared to global, reported in millions of tonnes equivalent CO₂) [1].

Year	Global CO _{2e}	Total Shipping CO _{2e}	% of Global	International Shipping CO _{2e}	% of global
2007	34,881	1121	3.2	903	2.6
2008	35,677	1157	3.2	940	2.6
2009	35,519	998	2.8	873	2.5
2010	37,085	935	2.5	790	2.1
2011	38,196	1045	2.7	871	2.3
2012	39,113	961	2.5	816	2.1
Average	36,745	1036	2.8	866	2.4

In the ECAs where the SO_x limit is in force, IMO introduced special fuel quality requirements, the sulphur compounds amount in the fuel must be below 0.1% in mass. For this reason, cleaner fuel must be used, unless efficient scrubbing technologies are used when employing Heavy Fuel Oils (HFO) for propulsion, in order to respect sulphur limits.

With respect to NO_x emissions, the limits imposed by the IMO Tier regulations depend on the maximum engine operating speed (N , rpm) as shown in Table 3. The Tier I and II thresholds are global, while the Tier III standards must be applied only in NO_x ECAs.

Tier II standards are expected to be met by combustion process optimization and engine operations improvements, with focus, for example, on fuel injection timing, injection pressure, injector and spray developments, exhaust valve timing (e.g. Miller timing) and cylinder

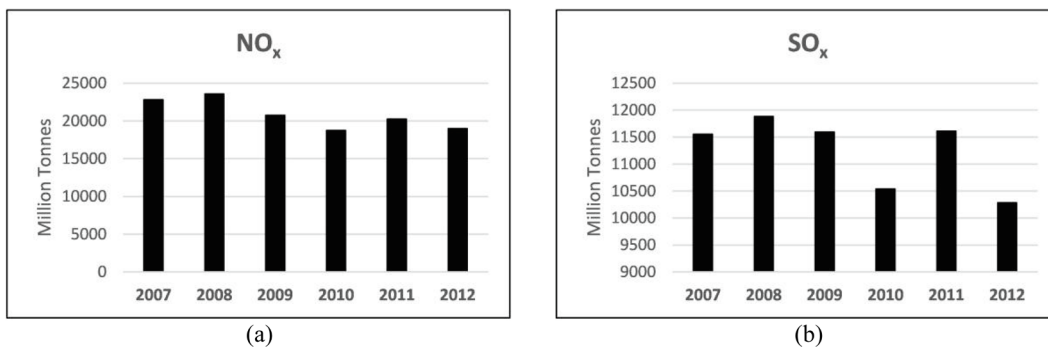


Fig. 1. NO_x (a) and SO_x (b) emissions trend in the 2007–2012 period [1].

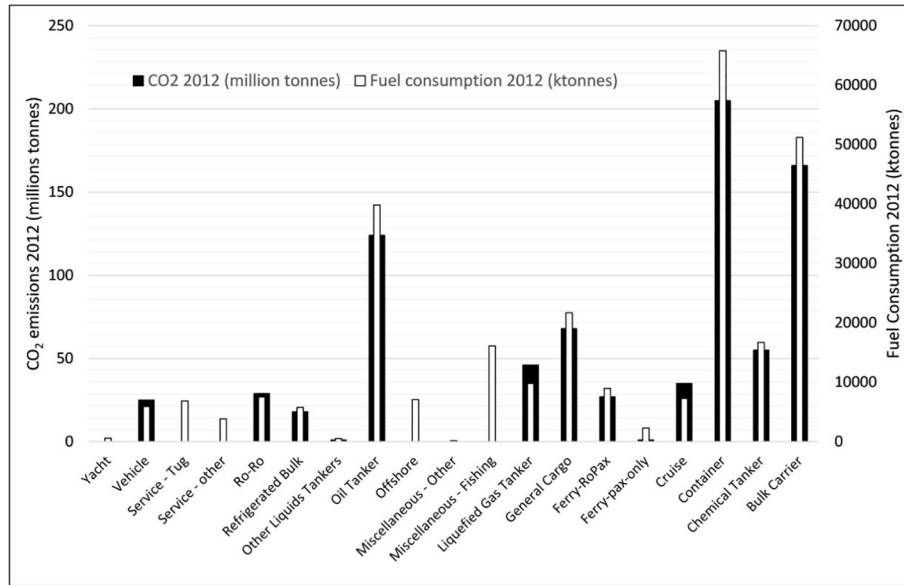


Fig. 2. CO₂ emissions and fuel consumption for different types of ships in 2012 (IMO GHG Study 2014) [1].

Table 3

IMO Tier NO_x regulation limits [4].

IMO Tier	Date	NO _x Limit, g/kWh		
		N < 130 rpm	130 ≤ N < 2000 rpm	N ≥ 2000 rpm
Tier I	2000	17.0	45 · N ^{-0.2}	9.8
Tier II	2011	14.4	44 · N ^{-0.23}	7.7
Tier III	2016	3.4	9 · N ^{-0.2}	1.96

compression ratio, supported by advanced turbocharging strategies.

However, Tier III standards are expected to require dedicated NO_x emission control strategies or technologies, such as water injection, Exhaust Gas Recirculation (EGR) or Selective Catalytic Reduction (SCR).

Tier III regulation imposes 80% NO_x reduction compared to IMO Tier I, for low speed engines.

Moreover, with the MARPOL Annex VI introduction, the Energy Efficiency Design Index (EEDI) [5] has been made mandatory for new ships, and it aims at promoting the use of more energy efficient and less polluting engines and ships equipment, thus including also waste heat recovery systems, in order to reduce CO₂ emissions. The Ship Energy Efficiency Management Plan (SEEMP) also establishes a mechanism for operators in order to measure and control GHG emissions from the already existing shipping fleet.

In this scenario, various emissions reduction strategies and technologies are currently developed and applied to both low and medium

speed diesel engines used for marine ship propulsion and power generation. These technologies commonly lead to a reduced engine fuel efficiency, but decreased pollutants. For this reason, their combined use with waste heat recovery systems such as steam Rankine cycles, Organic Rankine Cycles (ORC) and more advanced systems, could lead to an improvement in the positive trade-off between fuel efficiency and engine pollution.

In literature, several papers and reports are available about the description of the working principles and performance of different emissions reduction strategies, however none of them considers the effective potential that these technologies could have in relation to the possible utilization of waste heat recovery systems, in order to propose overall more efficient and less polluting engines.

This work is not meant to be an introduction to waste heat recovery systems, such as steam Rankine and ORC, but rather a critical overview of which could be the potential for waste heat recovery, using these systems, when considering a combination with new emission reduction technologies. An overview of ORC systems operating principle and applications can be found, for example, in [6–8].

In this paper an overview of the main technologies used and developed in the marine sector is proposed, assessing qualitatively and, in some cases, quantitatively their potential for waste heat recovery. Moreover, an overview of studies and implementations of ORC technology for marine low and medium speed engines waste heat recovery has been also proposed, in synergy with the evaluation of slow steaming operational practice, with the scope of properly designing waste heat recovery systems which could be suitable for each typical application

and operational regime.

2. Heavy duty Diesel engines

In this section, some statistics about typical HDDE used in marine applications have been reported, considering both two-stroke and four-stroke units, for propulsion and also for auxiliary energy generation, in order to give an overview of the engine architectures and characteristics available in the sector, which then will influence the right choice and design of the emissions reduction technologies and waste heat recovery systems.

The analysis is based on the re-elaborated data obtained from the EnginLink database [9]. The data have been considered for a timeframe between 1972 and 2016 (February), due to availability reasons, and have been reported divided in two different categories, based on the engine power output.

The category of 200–1000 kW (Fig. 3), is typical of smaller marine engines applications or genset, thus also considering high power output outboard motors, and the category with power output higher than 1 MW (Fig. 4), is representative of high tonnage ships propulsion engines and marine gensets for on-board auxiliary power generation.

Regarding the smaller power output range, the engines statistics report how IMO Tier III emissions levels are usually met with the use of SCR, with high percentage of engines still not having any emission reduction technology installed (88.6%). The fuel used is mostly gasoline (65.7%), due to the presence in the statistical data of outboard engines for recreational small marine applications, or small boat inboard engines. The main emissions compliancy standards are IMO Tier I-II-III, EU Stage II, US Pre-SI Phase 2, US SI Phase 1–2 and US Tier 2–3 [4]. The engines are mostly naturally aspirated (62.9%), with some models turbocharged (30.9%) or supercharged (6.2%), with 4 to 12 cylinders.

In the higher power range, the engines usually meet also IMO Tier III emissions with SCR aftertreatment only (43.5%), with EGR showing basically negligible percentages. Also in this case, a high percentage of engines present no emission reduction technology, thus showing the need for further development and installation of SCR and EGR systems to meet new emission standards. The fuel used is mostly diesel (Marine Diesel Oil, 67.5%), followed by Heavy Fuel Oil (HFO, 29.6%) and some

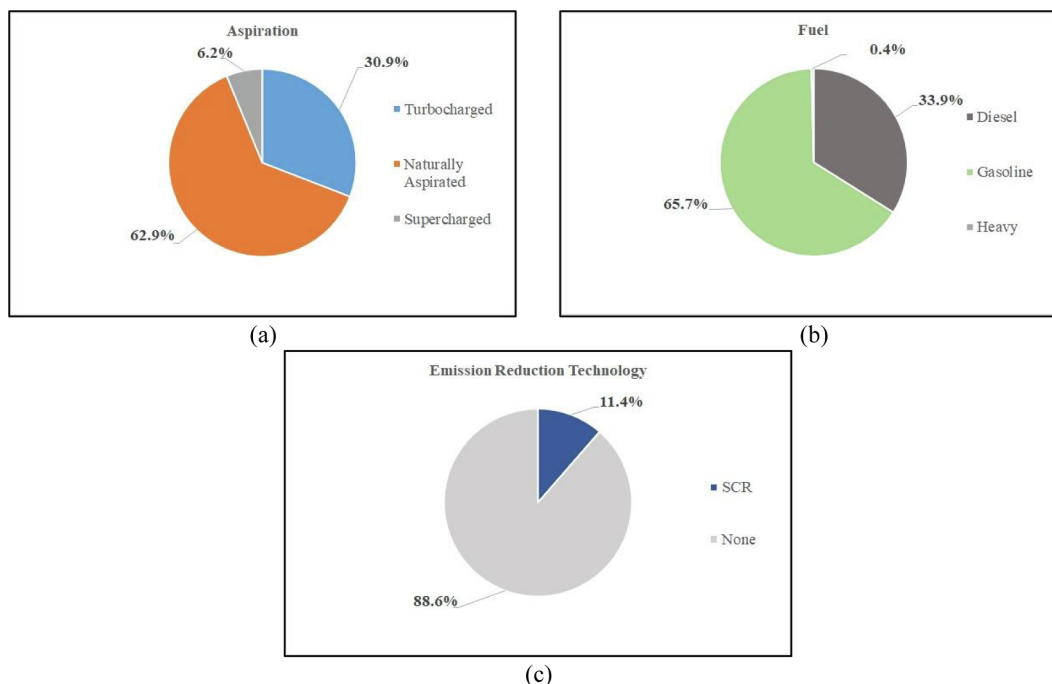


Fig. 3. Marine engines statistics. Power range: 200–1000 kW. Elaborated from EnginLink [9].

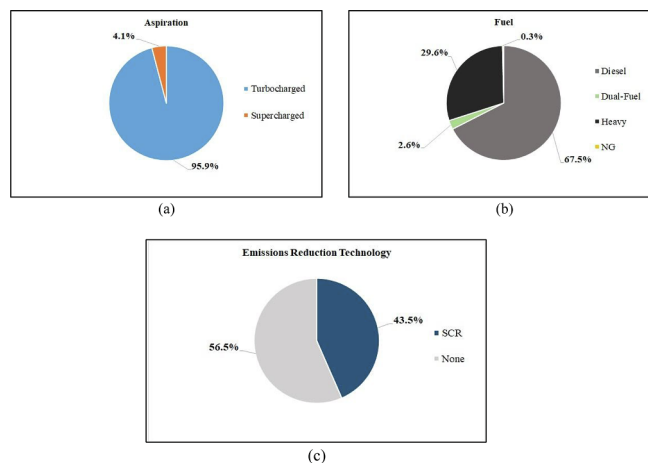


Fig. 4. Marine engines statistics. Power range: higher than 1 MW. Elaborated from EnginLink [9].

small percentage of dual fuel (2.6%). Natural Gas (NG) shows an almost negligible percentage (0.3%), but it is slowly spreading into the shipping industry as a promising alternative. The main emissions compliancy standards are IMO Tier I-II-III, EU Stage II and US Tier 2–3–4 [4]. The engines are almost all turbocharged (95.9%), with a small percentage of supercharged (4.1%), with 5–20 cylinders.

In the marine sector, mainly low and medium speed HDDE are used, both for ships propulsion and auxiliary power generation. Generally, low speed engines are two-stroke MW size propulsion units, running on MDO or HFO, while gensets are usually four-stroke engines. In Fig. 5, typical heat balances have been reported for a WINGD 6RT-flex58T-D 13.6 MW low speed two-stroke engine at full load conditions (Fig. 5a, [10]) and a Wärtsilä W6L20 1.2 MW four-stroke genset, again at full load conditions (Fig. 5b, [11]).

As it can be observed from the engine energy balances, MW size two-stroke engines for ship propulsion can reach a brake thermal efficiency higher than 50%, while four-stroke units are typically in the range between 40 and 50%. For these types of big power output

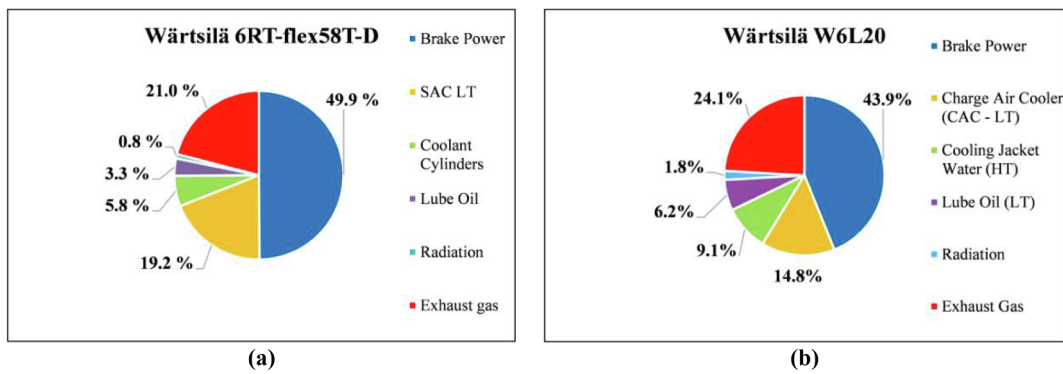


Fig. 5. Typical heat balances for two-stroke low speed (a) [10], and for four-stroke medium speed (b) [11], marine diesel engines.

engines, a high potential for waste heat recovery can be expected, considering heat sources as exhaust gas, EGR (when employed), Charge Air Cooling (CAC, four-stroke), Scavenge Air Cooling (SAC, two-stroke), cooling jacket water and lubrication oil. A higher potential regarding exhaust gas heat recovery is expected for four-stroke units, due to the higher temperature of the exhaust gas after the turbine. However, the large amount of gas mass flow available in two-stroke engines could also lead to possible advantages when considering heat recovery systems, even though, often, exhaust gas economizers are already installed on the engine tailpipe in order to recover heat for the production of on-board steam or electricity with well-known Rankine cycles. However, heat sources such as high temperature (HT) jacket coolant, or lower temperature (LT) scavenge air or charge air circuits sources could be, in principle, exploited due to the high availability of wasted heat, even if at lower temperature. In this case, the ORC technology could become interesting in order to recover this, otherwise, lost energy.

Some data for two-stroke and four-stroke engines typical possible ORC heat sources and heat sinks are reported in Table 4 (from references [10,11] and experience).

3. Emission reduction technologies

Several emissions reduction technologies are available or currently under development in order to meet IMO Tier III standards.

MAN ([12–17]) presented a series of reports investigating the performance and technical feasibility of some different strategies, focusing in particular on EGR and SCR for the reduction of NO_x and SO_x , also through the use of scrubbing technologies, but also considering engine combustion tuning strategies, water injection, waste heat recovery and gas utilization.

Also Wärtsilä and Winterthur Gas & Diesel, WINGD (e.g. [18–21]) prepared presentations and reports concerning possible emission reduction strategies for low and medium speed marine engines.

The US EPA (Environmental Protection Agency) presented some reports ([22,23]) regarding NO_x emissions reduction technologies as well as considering the economic impact of several strategies which can be applied to marine Diesel engines to meet IMO Tier III standards. The ICCT (International Council on Clean Transportation) published a white paper [24] evaluating the techno-economic feasibility of different technologies to reduce GHG emissions from ships, not only considering engine related solutions, but other actions such as, for example, hull coatings, water flow optimization, weather routing and efficient route optimization, air hull lubrication, wind and solar powered propulsion (hybrid ships). These strategies have been considered also by Perera et al. [25], who investigated their impact on the ships efficiency indexes such as the EEDI.

Also, several researchers considered emissions reduction topics in marine applications. For example one of the first studies was proposed, more than twenty years ago, by Geist et al. [26] who presented an overview of NO_x reduction techniques to be applied to marine Diesel

engines, developed and studied at New Sulzer Diesel Ltd (later Wärtsilä and now WINGD). They declared, at that time, that IMO Tier I emission standards could be met with well-chosen engine primary methods only, such as combustion tuning, fuel injection tuning, compression ratio, Miller supercharging, excess air ratio, scavenging air temperature, wet technologies and EGR. Secondary methods, such as SCR, are declared to be more efficient but also more expensive, thus being suitable for future emission legislations (e.g. the actual IMO Tier III).

Lamas et al. [27] presented a short overview of NO_x formation mechanisms, marine emission legislations and NO_x reduction strategies for low speed 2-stroke Diesel engines, considering both primary and secondary measures, declaring how EGR and water addition are the most used technologies, leading to decreased NO_x but increased hydrocarbons and carbon monoxide in the engine exhaust.

Yang et al. [28] proposed a simplified methodology to provide ships owners with a tool in order to choose the most suitable NO_x and SO_x emissions strategies, quantitatively analysing the data collected from shipping companies, shipyards and marine academies, with regards to emission reductions percentages, costs, space requirements and return of investment.

Wik et al. [29] presented a publication which main objective is identifying the various options available for reducing engine SO_x and NO_x emissions and clarifying the main criteria that engine manufacturers consider in order to choose the best technology, both from a technical and economic point of view. They proposed also some case studies for different types of vessels.

Yfantis et al. [30] presented an overview of NO_x emissions reduction technologies in the marine shipping industry, reporting advantages and disadvantages of each of those considered.

Emission reduction technologies can be basically divided in primary methods (engine related) and secondary methods (engine retrofits). In the first category, technologies as advanced turbocharging strategies coupled with Miller timing, combustion tuning, wet methods, alternative fuels usage, EGR can be classified, while in the second category technologies such as scrubbers, SCR and waste heat recovery can be considered. Other strategies can also be applied and can be appropriate

Table 4

Typical heat source and sink data for four-stroke and two-stroke marine diesel engines.

Heat Source/Sink	four-stroke	two-stroke
	Temperature [°C]	
Exhaust Gas Temperature After Turbine (AT)	300–500	250–300
Cooling Water Jacket Temperature (HT)	80–90	80–90
Scavenge Air Cooler Cooling Circuit Temperature (LT)	/	25–36
CAC Cooling Circuit Temperature (LT)	40–65	/
Lube Oil Circuit	65–80	60–75
Sea Water (Heat sink)	–5 to +40	
Engine Room Air	Up to 40–45	

to ships operations or ships design, such as slow steaming, route and weather routing optimization, hull optimization and coating. A conceptual scheme of the possible engine-related emission reduction technologies and strategies for four-stroke and two-stroke engines is presented in Fig. 6. Another overview on emissions reduction technologies is proposed by Balcombe et al. [31], considering also various solutions, as LNG and other alternative fuels (such as hydrogen and fuel cell related technologies), wind, solar and nuclear propulsion, hull coatings, and analysing their carbon saving potential.

In the next sub-sections, the main emission reduction technologies have been analysed, considering in particular those who are having more impact on possible waste heat recovery systems. DOC, DPF and scrubbers are for this reason not evaluated in the details.

3.1. Advanced turbocharging, charge air cooling and Miller timing

Turbocharging is the oldest developed method to increase engine efficiency. Turbochargers use the enthalpy of the exhaust gas to rotate a turbine which is connected to a compressor, in order to rise the engine boost pressure, increase engine volumetric efficiency and thus performance, reducing fuel consumption.

Turbochargers are nowadays well-developed technologies, which are used in most medium and low speed heavy duty Diesel engines, and, in particular, are essential for the scavenging process in two-stroke low speed Diesel engines, which can be equipped with up to 4 units for engines up to 20 cylinders.

Two-stroke turbocharging strategies can be basically of two types: constant pressure (used by almost all two-stroke low speed marine Diesel engines) and pulse.

In the constant pressure type, all cylinders' exhaust in the common exhaust receiver, which dampens out the gas pulses, thus maintaining almost a constant pressure, eliminating multiple complex exhaust manifolds systems, typical of pulse configurations, and leading to higher turbine efficiency and lower BSFC. Higher flexibility in the positioning of the turbocharger is also possible. This configuration has low performance at part load and during transition periods due to the slow response of the turbocharger, but this could be not a very big drawback during constant speed and load operations typical of a ship.

Charge Air Cooling (or Scavenge Air Cooling in two-stroke engines) is used to decrease temperature and increase air density after the compressor in order to improve engine volumetric efficiency and performance. The most common type of CAC (or SAC) is a water-cooled

design with finned tubes in a casing carrying seawater or fresh water, over which the engine intake air passes. A mist-catcher is also usually placed after the SAC in order to collect condensed water from the gas flow [32].

Large turbochargers for low-medium speed ships Diesel engines can be manufactured based on radial or axial turbines, usually connected to radial compressors.

Single stage turbocharging architectures are able, at the moment, to supply boost pressures up to 4–6 bars, with compression ratios up to 5.8:1 [33]. Due to the improvement of turbochargers efficiency in the last decades, up to 70% and more, considerable BSFC reduction has been achieved, and some other concepts and modifications have been applied to two-stroke low speed engines. An example is the “low port” system [34], in which, with a very efficient turbocharger, the required amount of scavenge air for combustion can be supplied to the cylinders using a lower amount of exhaust gas energy. For this reason, the height of the scavenging air ports can be reduced, and the opening of the exhaust valve delayed, increasing the working expansion stroke, thus increasing engine efficiency and reducing fuel consumption, not sensibly reducing exhaust gas temperature. This configuration is also beneficial for waste heat recovery, due to the availability of almost the same exhaust gas energy to be recovered, if compared to a normal-port configuration.

Waste Gate (WG) turbochargers are not very commonly used in low speed two-stroke engines, due to the fact that the operational profile of a ship usually does not show many transition periods, leading mostly to steady state operating points [34]. Exhaust WG could be even a disadvantage because the loss of exhaust gas reduces the turbocharger efficiency, thus leading to higher thermal load of the combustion chamber components and lower boost for the scavenging process. Moreover, at part-load conditions (especially lower than 30%), auxiliary blowers are used to support turbochargers operations.

Variable Geometry Turbines (VGT) can also be used in order to improve engine performance at part-load conditions (e.g. slow steaming operations), allowing optimisation of the performance for the considered operating point, with a smooth continuous control over speed and load points. MAN supplies a VTA (Variable Turbine Area) turbocharger, with a nozzle ring with adjustable vanes [35]. This allows to increase boost pressure in the low and medium speed/load range, thus increasing engine efficiency. MAN declares up to 8.5% fuel efficiency improvement at loads between 25% and 70%.

Heim [34] reported that also VGT turbochargers are not very

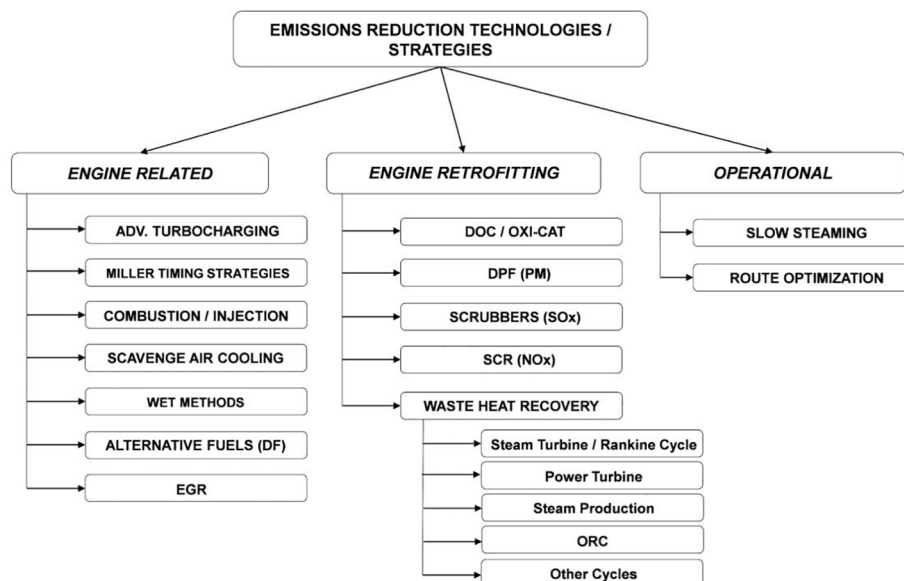


Fig. 6. Emission reduction technologies and strategies conceptual scheme.

practical for two-stroke low speed engines running on HFO, due to fooling problems due to unburned and lubricating oil in the gas stream. Increased cost is another drawback. For these reasons VGT are generally used in four-stroke engines.

Some other technologies variants have been considered in recent developments. For example, a hybrid turbocharger [36] comprises an electrical generator embedded in the turbocharger body (Mitsubishi Heavy Industries, MHI). In this way, the turbocharger can be used as a generator and also as a motor through the application of bidirectional frequency converters (AC-DC). The hybrid turbocharger can also be used in order to reduce the required electric power needed for the auxiliary blower at part load conditions and is declared to operate with useful power output for engine loads higher than 75%. However, MHI declares the hybrid option more efficient than using an auxiliary blower, leading to a benefit in terms of both improved fuel efficiency and decreased blower configuration power consumption [37]. Highly efficient turbochargers could also allow to use some of the exhaust energy for power generation instead for engine boosting purpose (e.g. up to 13% of energy use declared possible from MHI).

Due to the higher exhaust gas temperatures, advanced turbocharging strategies, as for example two stage turbochargers, are mostly used and developed for medium speed four-stroke engines. Indeed, exhaust gas temperature at the outlet of the first turbine stage, especially in case of low speed two-stroke engines, is quite low (250–300 °C). For this reason, two stage configurations are less spread in the low speed marine propulsion engine sector but could become interesting in the next future with the development of very efficient turbochargers. Now, these technologies are still under development. The idea behind is to increase the engine power per cylinder and reduce the total number of cylinders of the complete engine required for the ship propulsion. This is very similar to what is called “downsizing” in some four-stroke heavy duty diesel engine and passenger cars applications.

Some of the benefits of two stage turbocharging are: intercooling (in order to decrease second stage compression work), lower mechanical stresses, optimization of turbocharger operations over a wider range of speeds and loads, due to the operational flexibility of this configuration.

Some of the drawbacks are low-load efficiency worse than single stage configurations, more complex architectures with lower reliability, increased space required for installation, lower heat available for steam production and waste heat recovery purpose.

Two stage turbocharging efficiencies are expected to be in the range between 70 and 80% [38]. Because of this high efficiency levels, two stage turbocharging are commonly considered in four-stroke engines also in combination with Miller timing strategies [39] in order to take profit of the higher boosting pressure capabilities during the longer intake valve closure period used to reduce NO_x emissions.

In case of Miller timing strategies, the idea is to compress the scavenging air to a pressure higher than that needed for the engine operation, but the cylinder filling is reduced, as well as the effective compression ratio, using a suitable timing of the inlet valve (Inlet Valve Closure, IVC). For example, an opportunity is using early IVC (Intake Valve Closing) before the BDC (Bottom Dead Centre), for internal sub-cooling. This, of course, is valid for four-stroke medium speed engines, in which, a maximum NO_x reduction of around 40% to 50%, compared to Tier I levels, can be expected [30,39,40].

In two-stroke low speed engines, Miller timing strategies can be implemented retarding the Exhaust Valve Closure (EVC) and increasing scavenging boost pressure [41].

Because of late EVC, a part of the scavenging charge is pushed out from the cylinder. For this reason, in order to maintain similar Air-Fuel Ratio (AFR), Miller timing strategies need a higher intake pressure than commonly supplied with single stage turbochargers architectures. Miller strategies are commonly used in combination with highly efficient turbochargers or two stage turbocharging architectures to supply enough boost for operations.

If the maximum achievable boost pressure supplied by the

compressor is too low, Miller timing strategies result in worse engine performance and increased fuel consumption.

As reported by Feng et al. [42], in their simulation analysis about a two-stroke low speed 3.6 MW brake power marine engine, Miller timing in combination with single stage turbocharging has the potential of around 45% NO_x reduction at full load, but with a penalization in power and SFOC.

Again Feng et al. [42], proposed to implement Miller timing and two stage turbocharging, obtaining from their 1D-3D performance and combustion coupled analysis results, better results than with single stage turbocharging architecture (higher NO_x decrease and lower SFOC and power penalty).

As a final statement regarding waste heat recovery, it can be said that two stage turbocharging architectures are usually not beneficial for combined use with waste heat recovery systems such as steam Rankine or ORC, due to the fact that the two stages expansion results in a decreased amount of energy that can be recovered from the exhaust gas at the turbine outlet, due to the lower gas enthalpy at the low-pressure turbocharger outlet. Possible waste heat recovery systems, such as Power Turbines (PT), steam Rankine or ORC could be used when considering Waste-Gate systems in operations, eventually considering recovering by-passed exhaust gas heat, which would be in this case not utilized.

3.2. Engine combustion tuning

A reduction of NO_x emissions compared to Tier I level can also be obtained by tuning and by optimizing engine internal parameters, especially related to combustion. These strategies are helpful in order to target Tier II emission levels, however, they are not enough in order to meet Tier III limitations, because they allow a maximum NO_x reduction of 20–30% compared to Tier I [28,30,43]. For this reason, these methods have not been evaluated in the details, even though they could have impact on possible waste heat recovery systems, especially when affecting exhaust gas temperatures (and available enthalpy). A wide combustion engine-related literature is available anyway for consultation (e.g. [44]).

A non-detailed overview of the main possibilities that can be considered in the combustion optimization is reported here below [27]:

3.2.1. Delay of SOI/decrease of injection duration

A delayed injection leads to lower peak cylinder pressures and temperatures. Retarding injection leads also to lower mixture combustion residence time for NO_x formation.

A retard of injection of several CAD can lead up to 20–30% NO_x emissions reduction, but with penalization of engine performance and increase in fuel consumption.

3.2.2. Pre-injection

Pre-injection can be used to decrease the ignition delay period, thus decreasing temperature and pressure during the preliminary phase of combustion, also reducing particulate matter emissions. Fuel consumption can be slightly reduced if the pilot injection timing is optimized.

3.2.3. Modification of fuel injectors

Reducing spray cone angle can lead to reduction of NO_x formation, but to an increase in fuel consumption, due to the lower amount of air enclosed in the spray, leading to a less homogeneously prepared mixture for premixed combustion phase, decreasing combustion efficiency.

Increasing dimensions of fuel injectors tips also decreases NO_x formation but increases fuel consumption due to the spray position concentrated more next to piston bowl, decreasing combustion efficiency, pressure and temperature. The number of injectors holes and diameters can also be optimized as a trade-off between NO_x formation and fuel consumption.

3.3. Wet technologies

The introduction of water in the combustion chamber reduces NO_x emissions formation due to the increase of the specific heat capacity of the in-cylinder gases, having water a higher specific heat capacity compared to air.

Mostly three possibilities are available: Water-in-Fuel Emulsion (WFE), Direct Water Injection (DWI), Intake Air Humidification – Humid Air Motor (HAM, or Scavenging Air Moisturizing, SAM) [19,32].

Wet technologies have been intensively used, together with engine parameters tuning and combustion related strategies, in order to reduce emissions from Tier I to Tier II levels but seem not to be enough for Tier III emissions limits compliancy. Moreover, water addition is controversial, due to the observation that water droplets impacting on the cylinder walls can immediately destroy the lubrication oil film, and also that water in the cylinder can lead to corrosion problems. These problems, however, are due only to liquid water. Indeed, once water is in a vapour state, it can no longer affect the lubrication oil film, and does not condense unless saturation is reached, being this problem not very important at high combustion temperatures [45].

Water availability problems are less important for ship applications rather than for vehicles, however, freshwater systems (desalination) could be needed, due to the need of large amount of water for NO_x reduction.

3.3.1. Water-in-fuel emulsion (WFE)

The WFE consists in adding water to the fuel, thus creating an emulsion, before the injection in the combustion chamber. This leads to a combined decrease of both NO_x and soot emissions. WFE has been largely tested on large two-stroke Diesel engines.

An adequate emulsifier is needed in order to emulsify large quantities of water (up to 50%) into distillate fuels, while still allowing safe and reliable operation and a homogeneous emulsified mixture of water droplets in the fuel phase [45,46]. While some sources declare a NO_x reduction potential of maximum 15–25% [19], some other sources report higher NO_x reduction potential up to 50% [28,45,47].

Some advantages and disadvantages of the WFE technology are

Table 5

Wet technologies: advantages and disadvantages.

Wet Technology	Advantages	Disadvantages
WFE	<ul style="list-style-type: none"> ● Only marginal increase in SFOC ● Reduced smoke formation at low load ● Low water consumption compared to HAM ● Water quality not crucial as in HAM ● Enhanced fuel spray atomization and mixing ● Lower impact on turbocharger compared to HAM (drift towards compressor surge and speed increase) ● Equipment can be used in order to reduce viscosity of fuels ● <i>Low impact on WHR possibilities</i> 	<ul style="list-style-type: none"> ● Low-medium NO_x reduction potential (15–25%) without engine modifications ● Increased smoke formation and poor engine performance in case of switching off the system (low flexibility) ● Increased mechanical stress on the fuel injection system in case «standard» nozzles are used ● Limitations due to injection system ● Limited experience on long-term
DWI	<ul style="list-style-type: none"> ● High NO_x reduction potential (50%) ● Low water consumption compared to HAM ● Water quality less crucial than with HAM ● Intake duct design not changing ● No corrosion/fouling issues for SAC ● Flexible system (switch on–off) ● Less impact on turbocharger compared to HAM (drift towards compressor surge and speed increase) ● <i>WHR possibilities unaffected</i> ● Better long-term experience 	<ul style="list-style-type: none"> ● High fuel consumption penalty ● Increased smoke formation especially at low loads (need of injecting lower amount of water at low loads with consequent decrease of NO_x reduction) ● More complicated and expensive system compared to HAM ● Challenges in terms of piston top and injector corrosion with high sulphur fuels ● Engine must be re-designed in order to allow efficient operations (not a good retrofit)
HAM	<ul style="list-style-type: none"> ● Only marginal increase in SFOC ● Less complicated and expensive system compared to DWI ● Flexible system, better control of water flow rate ● Could be developed in order to increase knock-margin in gas engines operations 	<ul style="list-style-type: none"> ● Lower NO_x reduction potential (10–40%) compared to DWI and well-designed WFE ● High water consumption compared to DWI and WFE ● Very clean water needed to avoid fouling and corrosion of the SAC and intake manifolds ● <i>WHR possibilities affected (less cooling water heat available for production of vessel clean water, exhaust gas can be used to evaporate water for intake fumigation)</i> ● Affecting turbocharger, especially regarding surging problems. Not possible with pulse charging systems ● Increased smoke at low load (less water injected) ● Limited long-term experience

reported in Table 5 [19], together with the other two wet methods considered and introduced in the next paragraphs.

3.3.2. Direct water injection (DWI)

Direct water in-cylinder injection requires an independent water injector placed in the cylinder head and allows high quantity of water to be injected without affecting too much engine performance. However, the position of the water injector must be carefully considered in order to avoid liquid water impingement on the combustion chamber walls. The DWI system is more flexible compared to WFE since the water injection can be easily switched off.

Wärtsilä developed the system for medium-speed engines after initially considering ammonia instead of water. Water showed similar potential as ammonia but with less health hazards and environmental concerns [19].

The Stratified Fuel Water Injection (SFWI) concept has also been considered combining DWI and WFE using a single injector to inject water and fuel directly in the combustion chamber [48].

3.3.3. Intake air Humidification – Humid air Motor (HAM)

The introduction of water in the scavenging intake air is the simplest wet method, but also the least efficient between the considered for NO_x reduction. In particular, if water is not fully evaporated, the droplet of liquid water could reach the cylinder walls creating problems with the lube oil film, and possible corrosion problems [45]. For this reason, it is better to consider vaporized water injection instead of liquid water, possibly using the exhaust gas heat of the engine for the evaporation process, thus, however, being not very beneficial for waste heat recovery configurations heat availability.

A recent review of water injection methods applied to internal combustion engines has been proposed by Zhu *et al.* [49].

3.4. Dual or multi-fuel engines

The use of Liquefied Natural Gas (LNG) as fuel in shipping leads to several environmental advantages. An LNG fuelled ship reduces the emissions of NO_x by at least 85% to 90% (using a gas only engine), and

SO_x and particles by close to 100% compared to today's conventional marine fuel oils or heavy fuel oils [50]. The most critical issues of using LNG as a fuel for ships are bunkering and safe fuel supply to the engines, as well as the optimization of new engine combustion systems able to fully take advantage of the new possible fuel. Ships new design or retrofitting of older vessels are also at the moment considered. Mostly LNG is considered as fuel, but Liquefied Petroleum Gas (LPG), Dimethyl Ether (DME), methanol, ethanol are also under investigation. A review of ship propulsion systems using LNG is reported by Fernandez *et al.* [51].

First developments have been carried out mostly on four-stroke engines (for example the Wärtsilä Dual Fuel DF engines, capable of running both on Diesel or gas modes), but currently two-stroke engines gas combustion systems are under development, focusing in particular, as reported by Wärtsilä [52], on pre-mixed lean burn combustion using a Otto cycle (WINGD concept), or on the typical Diesel Cycle with pilot Diesel injection and direct gas injection (MAN concept). The two different concepts main features are reported in Table 6 [52].

MAN uses a Diesel combustion mode with high pressure gas injection (300 bar), called ME-GI injection system [17,53]. Compared, for example, to HFO, gas leads to a cleaner exhaust, with very low or no sulphur content (SO_x almost negligible, thus reducing scrubber needs). However, it seems that EGR or SCR emission reduction strategies are still needed to meet IMO Tier III emissions level. HC emissions increase in percentage with gas operations, but still are low in absolute numbers, due to the efficient combustion in two-stroke low speed engines.

A comparison between HFO and LNG operation impact on emissions has been reported, as example, in Table 7 for a MAN 70 cm bore 6S70ME engine (without aftertreatment) [53].

WINGD [54], for their Otto Cycle LNG combustion concept, declares 30% reduction of CO₂, 85% reduction of NO_x, 99% reduction of SO_x and 95% reduction of PM emissions.

In vessels, LNG is stored in cryogenic insulated tanks with temperature around -162 °C at near-ambient pressure. Both with a low-pressure or high-pressure LNG gasification system, the LNG needs to be gasified to Natural Gas (NG) before being sent to the engines. This gasification process requires heat to be supplied to the LNG vaporizer, which in turns is acting as a heat sink for the process.

Often the regasification is performed using, as hot source, sea water, with an average temperature between 0 and 25 °C depending on season and place, which is still high enough to evaporate the low temperature cryogenic LNG [17]. This is a cheap method in ships to evaporate the LNG. There is still however loss of potential to produce work, due to the high temperature difference between heat source and heat sink in the regasification system, and this is where, eventually, a waste heat recovery system could help to improve overall system efficiency.

The available energy (or exergy) can be recovered using different power cycles. An example could be an ORC system optimized for very low temperature sources. For example, Tomków *et al.* [55], presented the process simulation results of an ORC system running on ethane fluid, used to recover sea water heat, taking profit of the low temperature LNG heat sink, in which the ORC condenser acts as a vaporizer for the LNG line. The system benefits from low exergy losses in particular in the condenser, however a good trade-off between LNG

Table 6
LNG 2-stroke combustion systems.

Otto Cycle (WINGD)	Diesel Cycle (MAN)
<ul style="list-style-type: none"> ● Processes: scavenging, compression/gas admission, ignition and expansion ● Injection of gas at mid-stroke position ● Low pressure injection (< 10 bar) ● High impact on NO_x emissions (lower combustion temperatures achievable) ● Meets IMO Tier III without aftertreatment ● Possible methane slip problems 	<ul style="list-style-type: none"> ● Processes: scavenging/compression, pilot Diesel injection and high-pressure gas injection, expansion ● Injection of gas close to TDC ● Air is compressed, thus high-pressure gas injection is needed (300 bar) ● Lower NO_x reduction (declared by Wärtsilä) ● Requires aftertreatment to meet IMO Tier III levels ● No methane slip problems ● Possible improvement in SFOC in gas mode compared to Diesel mode

Table 7
Comparison between HFO and LNG operation for a MAN 6S70ME 2 S engine operating on LNG Diesel mode [53]

HFO		LNG		Δ (%)
Load 100%	g/kWh	Load 100%	g/kWh	
CO ₂	577	CO ₂	446	-22.7
O ₂ (%)	1359	O ₂ (%)	1340	-1.4
CO	0.64	CO	0.79	+23.4
NO _x	11.58	NO _x	8.76	-24.4
HC	0.19	HC	0.39	+105.3
SO _x	10.96	SO _x	0.88	-92
PM (mg/m ³)	0.54	PM (mg/m ³)	0.34	-37

vaporization pressure and LNG compressor power absorption must be considered.

Sung *et al.* [56] proposed a dual loop system (considering two separated circuits) to be applied to dual fuel engines, in order to recover, in the high temperature ORC loop, waste heat from engine exhaust gas, and in the low temperature loop, heat from the engine coolant, then rejecting heat to the cold LNG loop. Recuperated and pre-heated architectures have been also considered. 5.17% additional power compared to the baseline engine have been obtained using a process simulation model and n-pentane and R125 as working fluids.

The LNG vaporizer can be, theoretically, also used as possible heat sink for a higher temperature ORC recovering heat from coolant or higher temperature heat sources of the main engine or power generators. This could increase the ORC power output, increasing the recoverable enthalpy drop in turbine, but, at the same time, probably requiring more complicated expansion machine design (due to the very high expansion ratios).

Moreover, very low temperature suitable fluids could be needed in order to recover sea water heat (heat source) with LNG vaporization as the heat sink. More possibilities are available when recovering higher temperature heat sources.

A selection of the fluids that can be used for LNG "cold" recovery is presented in Table 8 where on shore installation are also included.

Ships applications are today not often considered, since gas engines are not yet spread in shipping. Potential for this type of application will increase in the next future if LNG will become more popular for ships fleets.

A recent overview of the prospect and challenges of the combined use of LNG and waste heat recovery systems has been reported by He *et al.* [57].

3.5. Selective catalytic reduction (SCR)

A way of meeting IMO Tier III targets is installing a Selective Catalytic Reactor (SCR) on the engine exhaust line. NO_x are reduced catalytically, through the injection of ammonia (urea), to nitrogen (N₂) and water (H₂O) [16]. The catalyst in the SCR reactor consists of blocks with a large number of channels, providing a large surface area, in which the catalytic reactions happen [14].

The technology is mature for stationary power plant applications

Table 8

Working fluids, heat sources and heat sinks for ORC exploiting LNG cold potential.

Fluids	Heat Source	Heat Sink	Refs.
propane	Sea Water	LNG Vapor.	[58]
ethane	Sea Water	LNG Vapor.	[55]
R245fa, R123, HFE7000, HFE7100, n-pentane, n-hexane	Exhaust Gas	Sea Water	[56]
propane, R143a, R125, R41, R152a, R134a, R218	Engine Coolant	LNG Vapor.	[56]
C4F10, CF3I, R236ea, R236fa, RC318	Exhaust Gas	LNG Vapor.	[59]
R22, R134a, R152a, propane, isobutane, R245fa, R123, isopentane	Exhaust Gas	LNG Vapor.	[60]

Table 9

Typical SFOC and SFOC penalty for LP and HP SCR Tier III operations (MAN CEAS, [61]).

SFOC [g/kWh] – relative to Tier II standard engine (LL EGB) – Engine: 6S60ME-C8.5 (14.9 MW)			
Fuel	≤3.5% Sulphur	≤0.1% Sulphur (ECAs)	≤0.1% Sulphur (ECAs)
MCR [%]	SFOC [g/kWh] Tier II operation (LL EGB)	SFOC penalty [g/kWh] Tier III operation (HP SCR)	SFOC penalty [g/kWh] Tier III operation (LP SCR)
100	167.5	+ 0.5	+ 1.0
75	161.6	+ 1.0	+ 1.0
50	159.5	+ 1.5	+ 1.0
25	166.1	+ 2.0	+ 1.5

but not completely for marine applications. For example, MAN is developing the technology together with Hitachi Zosen Corporation, focusing the activity on systems for large two-stroke engines [16]. WINGD is also developing the technology both for four-stroke [20] and two-stroke engines [52].

Because of the high thermal efficiency of large two-stroke engines, the temperature of exhaust gas at turbine outlet is usually lower compared to four-stroke engines (see Table 4). For this reason, and especially when considering operations using HFO, it is necessary to ensure the right temperature of the gas at the SCR inlet, which, following MAN guidelines, should be around 330–350 °C [16]. The temperature can be reduced up to 300–310 °C in case of low sulphur fuel content (< 0.1%) [14], still allowing a safe margin against sulphur deposition (ammonia bisulphate) and corrosion problems.

Considering what reported above, generally, two different SCR systems possibilities are available: HP SCR (High Pressure) and LP SCR (Low Pressure).

In the HP SCR configuration, the vaporizer (urea spray) and the SCR reactor are placed before the turbo in order to take profit of the higher temperature available for an effective catalytic process. When Tier II mode is possible, the SCR system is bypassed by valves. The exhaust gas passes directly to the turbocharger. When operating in Tier III mode, the exhaust gas is passed through the SCR reactor [14]. The cylinders and SCR by-pass valves are opened in low-load operating mode in order to decrease the amount of air for combustion (at fixed amount of fuel and scavenging pressure) thus increasing the exhaust temperature, otherwise temperature levels would be too low at low load points (below 50%) for correct SCR operations. Auxiliary blowers in this case must also be oversized due to the possible need of additional air to stabilize the system during SCR operations, because of the high heat capacity and thermal delay introduced by the SCR before the turbine.

When restricting the fuel sulphur content, during SCR operations in ECAs, to 0.1% or less, it is possible to install the SCR in a low-pressure configuration (LP SCR), after the turbine, which provides more flexibility. Following what reported by MAN for their system [14], in this case, a mixer (AIG, Ammonia Injection Grid) and a Decomposition Unit (DCU) are needed. The DCU is made of a blower, a burner and a vaporiser. The reducing agent is injected in the vaporizer forming a mixture of ammonia vapour which is injected into the mixer, forced by

the blower and heated up by the burner. Part of the exhaust gas is bypassed to the lower pressure side of the turbine to increase the temperature for the SCR. This will increase the SFOC based on temperature level required. The DCU is used to dissolve the deposits of ammonia bisulphate, which cannot completely be avoided in this configuration. Due to formation of deposits, usually the boiler (or economizer), during SCR operations, could be bypassed, because of the too low temperatures [14]. Formation of ammonia bisulphate deposits in the boiler could lead to massive increase of backpressure in the exhaust line, thus decreasing the engine performance and increasing fuel consumption. For these reasons, the LP SCR configuration is not beneficial for waste heat recovery, especially if considering low load or slow steaming operations.

Examples of the MAN HP and LP SCR configurations are presented in [14].

Some typical data representing SFOC penalty percentage increase, due to Tier III ECAs SCR operations, have been reported for a MAN two-stroke low speed ME-C engine [14] in Table 9. The Tier II SFOC data are related to a 6 cylinders MAN 6S60ME-C8.5 14.9 MW engine and are obtained from the online MAN software CEAS [61], considering Low Load Exhaust Gas Bypass (LL EGB) setup.

Tests on a SCR system have also been conducted by Mitsubishi Heavy Industries (MHI) and have been carried out on a test rig equipped with a low-speed Mitsubishi-Akasaka 3UEC37LA engine (1103 kW, 188 rpm) at 100% load using ammonia as reducing agent. The tests aimed at a NO_x reduction of 80%. De-nitration rates for the SCR showed a slightly higher results than for Exhaust Gas Recirculation (EGR) but comparable in range (up to 80–90%) [62].

3.6. Exhaust Gas Recirculation (EGR)

Exhaust Gas Recirculation (EGR) is commonly applied in EURO VI four-stroke HDDE in the vehicle transportation sector. EGR, together with SCR (Selective Catalytic Reduction), is gaining also more interest for two-stroke ship propulsion engines, as one of the most effective technologies to reduce NO_x emissions for IMO Tier III legislation compliance. EGR could also lead to benefits regarding waste heat recovery systems applications, as already considered for smaller HDDE applications in literature [63–66], but not yet well spread in the marine sector.

Usually, EGR could be internal or external, depending if it is accomplished through particular valve timing or scavenging port height sizing, for the first case, or through the use of an external circuit to recirculate the gas to the intake side of the engine [30], for the second case.

One of the main effects of EGR on Diesel engine combustion and, in particular, NO_x emissions reduction, is due to the increase of the charge mixture specific heat capacity (higher when recirculating CO₂ and H₂O) which leads to a reduction of the in-cylinder gas temperature, thus reducing nitric oxides emissions. Another important effect is due to the decreased oxygen concentration in the intake due to the recirculated exhaust gas containing CO₂ and H₂O, which leads to a decrease of combustion rate and temperature.

Despite the EGR benefits considering NO_x reduction, the recirculation of exhaust gas leads to a general increase in fuel consumption and soot emissions. Moreover, in particular when using HFO, the high

sulphur content of the fuel can lead to corrosion and wear issues in the engine components, and in particular for the EGR cooler, particularly used in the HP (High Pressure) EGR configuration, and the EGR blower, used to push the recirculated gas from the exhaust to the intake side, both in HP and LP (Low Pressure) configurations.

To try to overcome these issues, usually, a scrubbing system is used in order to clean the recirculated exhaust gas. The system could be either a dry or wet system, with the second usually preferred in marine applications. The water used for scrubbing the exhaust gas can be fresh water, usually used in ECAs, recycled and cleaned on-board, or sea water, used outside ECAs and discharged back into the sea after cleaning. The EGR scrubber is usually more compact compared to the main exhaust scrubber, and this in particular for the HP configuration, in which gas pressure is higher compared to the LP case, which is recirculating gas after the turbine, so almost at ambient pressure, or even slightly below [67,68].

The scrubbing system needs a Water Treatment System (WTS) to comply with IMO criteria for washed water discharge [16]. The contaminated scrubber water needs to be cleaned for soot particles to avoid clogging up the system and the water generated during combustion, condensed in the EGR cooler, needs to be discharged to the sea, in clean conditions. Simple and compact installation for the WTS unit is essential for on-board applications.

As already introduced, the EGR system could be either a HP or LP configuration. The HP system recirculates the gas from the turbine inlet to the SAC inlet, while the LP system recirculates the exhaust gas from the turbine outlet to the compressor inlet. The HP system is used and developed, for example, by MAN [14], while the LP system is under development, in particular by Mitsubishi Heavy Industries [47]. Some simplified schematics for the main configurations are shown in Fig. 7: HP EGR in Fig. 7(a), while LP EGR in Fig. 7(b).

It is out of the scope of this work to classify all possible EGR architectures, in particular for the HP configuration, in which different layouts are available depending on how many turbocharger units are used in order to supply the right boost pressure to the engine, and depending on the switching ability between Tier II and Tier III operations, this last feature being very important for new engines which should be able to easily run below both emission regulations limits, inside and outside ECAs. More detailed information can be found in [14,16,69].

In case of HP EGR, usually an EGR cooler is used in order to cool the recirculated gas to a temperature compatible to the temperature of the intake air (150–250 °C) before the SAC, from a temperature before the turbine of around 300–450 °C, depending on the engine operating point [10]. For this reason, a high amount of heat is rejected to the engine cooling circuit, being this an energy loss for the overall system, and an additional thermal load to be rejected into the coolant. In this case, a waste heat recovery system such as a steam Rankine cycle or an ORC could be beneficial in order to recover the wasted heat and produce additional useful energy, in the form of mechanical or electrical power (or producing steam for ships usage), thus counterbalancing the typical EGR fuel consumption increase drawback.

The EGR cooler is usually made of stainless steel and can be subjected to highly corrosive environment, especially when HFO is used, even though, in ECAs, a low sulphur fuel should be used, thus decreasing the issue. Also, the position of the scrubber and the demister (used to trap condensed water and eject it from the gas line) should be carefully considered before or after the EGR cooler, depending on deposition and fouling problems, and considering system packaging constraints.

In particular, Alfa Laval – Aalborg [70] proposes a high pressure economizer (HPE) for EGR waste heat recovery, called EGR-HPE. The EGR-HPE is a pressurized boiler to be used in an engine system with HP EGR, developed in collaboration with MAN.

Additionally, when recirculating exhaust gas to the intake before the SAC, depending on the degree of cooling of the gas in the EGR

cooler, the temperature of the mixed gas in the intake could slightly increase, which, together with the high amount of mass flow of gas available in large low speed two-stroke Diesel engines, can lead to a large increase in heat rejection in the SAC, thus being a possible advantage in case of using low temperature suitable waste heat recovery systems, such as ORC.

An example of what reported above can be observed in Fig. 8, in quantitative form. The data have been obtained from MAN CEAS [61] and refer to a comparison between Tier II and Tier III operations with HP EGR for an MAN 6S60ME-C8.5 engine (14.3 MW brake power at full load, 105 rpm, with EGR by-pass configuration).

From the analysis of the data reported in Fig. 8, it is possible to observe how the SAC heat rejection increases of about 62% at full load conditions, 108% at 50% load and even up to 322% at 25% load conditions, when considering Tier III HP EGR operations in comparison to Tier II operations without EGR. No significant heat rejection increase is observed concerning coolant and lube oil circuits.

The reported data clearly show how recovering heat from the EGR cooler and/or the SAC, using a bottoming cycle such an ORC, could be a benefit regarding overall system performance.

Both in HP and LP EGR configurations, an EGR blower is needed to push the recirculated gas through the EGR circuit and equilibrate the pressure between the exhaust and the scavenging receivers (in Fig. 7 the blower has been reported as B). MAN is involved in the development of an efficient EGR blower, based on a high-speed radial turbo-compressor wheel [16], capable of withstanding the challenging conditions of the EGR circuit. As reported by MAN, the blower must be highly efficient over a wide flow range (0 – 40% EGR rate usually considered for IMO Tier III operations), should have a fast-dynamic response, should be corrosion resistant, reliable, compact, leakage proof. In some configurations the blower could become part of a turbo configurations, as

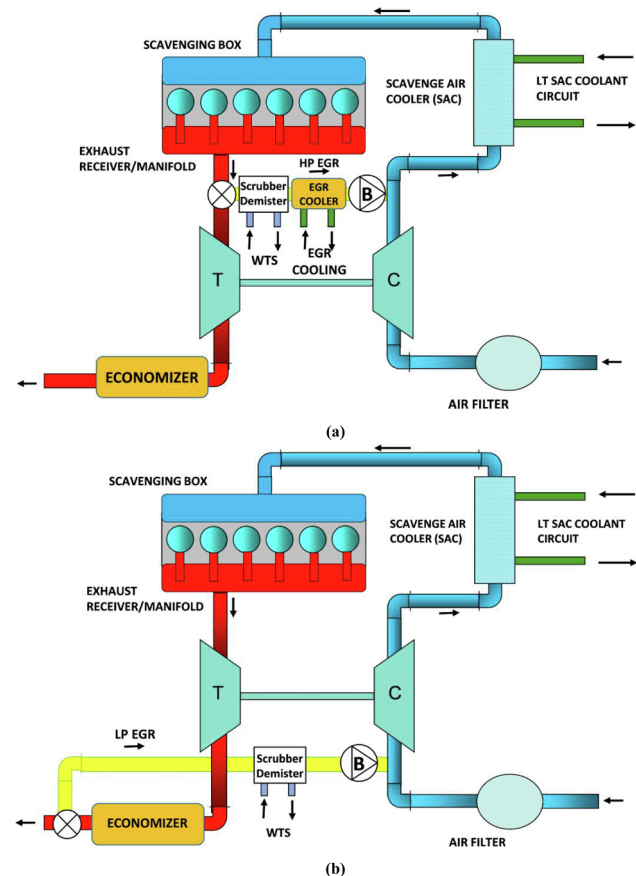


Fig. 7. Simplified schemes of HP (a) and LP (b) EGR configurations.

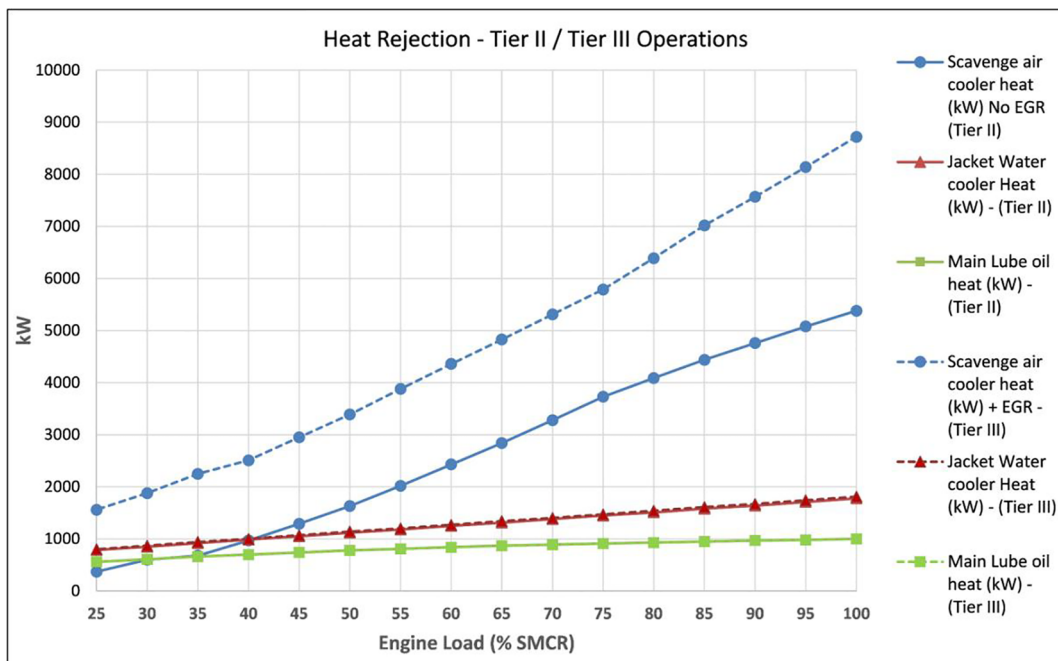


Fig. 8. Comparison between heat rejection in Tier II and Tier III (EGR by-pass configuration) for a MAN 6S60ME-C8.5 14.3 MW two-stroke low speed Diesel engine. Data extracted and post-processed from MAN CEAS software [61].

reported by Codan et al. [41], or could be just electrically driven, thus allowing an higher controllability.

Another consideration can also be drawn from the data obtained from CEAS calculations for the previously considered MAN 6S60ME-C8.5 engine. In particular, during Tier III operations with HP EGR, further power consumption absorbed when operating the blower and the WTS system [61]. For the considered engine, for example, an EGR blower electric power consumption of around 108 kW can be expected at full load, while around 88 kW at 50% load and 51 kW at 25% load. Moreover, the WTS system also requires power to be consumed, e.g. 57 kW at full load, 41 kW at 50% load and 33 kW at 25% load. The sum of the reported consumed power leads to an overall power need of 165 kW at full load, 129 kW at 50% load and 84 kW at 25% load, which must be summed to the increased fuel consumption expected during EGR operations.

For this reason, waste heat recovery systems can become even more attractive in order to counterbalance also this increased power consumption (around 1.1% of the produced engine brake power at full load) required.

Compared to LP EGR, HP EGR recirculates exhaust gas before the turbine, on the high-pressure side, with the drawback of reducing the exhaust gas mass flow which rotates the turbine, thus leading to a boost and engine performance decrease. A more efficient turbocharger is then required, or cylinder by-pass strategies must be adopted. Two stage turbocharging architectures can also be considered in synergy with EGR strategies in order to improve the boosting capabilities of the engine, as reported in the study by Feng et al. [42].

As mentioned before, the EGR tends to increase engine fuel consumption. Some data have been reported in Table 10, as an example, for a 4 cylinders MAN 4T50ME-X test engine, with HP EGR, 7 MW brake power and 123 rpm at 75% load, not considering EGR blower and auxiliaries power consumption [16].

As observed from the data, EGR rates up to more than 40% are usually required for 2-stroke large low speed marine Diesel engines; an higher amount compared to commercial vehicles smaller HDDE, as also reported by Codan et al. [41].

LP EGR is another concept, which currently is less spread in marine applications and still facing development and testing stages, as reported

by MHI [47,71,72], but promising a very good compromise between NO_x reduction capabilities, fuel consumption and easiness of construction. To the authors' knowledge, not many information is available in literature regarding LP EGR in large marine two-stroke applications.

In the LP EGR concept, the exhaust gas is extracted from the low-pressure side of the turbine (outlet), after the economizer, and recirculated to the low pressure side of the compressor (inlet), as shown in Fig. 7. For this reason, one of the main concerns of the technology is that it could lead to compressor failure, due the possible soot and condensate droplets which could damage the turbomachine (pitting and corrosion). Effective scrubbing and demisting systems are thus very important when considering this configuration. Another drawback of the LP EGR is related to the resulting bulkier system compared to the HP one, due to the lower pressures acting on the turbine outlet side, thus requiring more space for on-board installation. Moreover, OPEX (Operating Expenditure) is higher for LP-EGR, mostly due to the more expensive low sulphur fuel needed for operations without damaging the compressor.

However, the LP EGR architecture has also many other advantages. For example, it is simpler in structure compared to HP EGR, due to the lower system pressures involved and the lower temperatures, having also a lower CAPEX (Capital Expenditure). It requires a lower number of components, and the EGR cooler and the pre-scrubber are not needed. The EGR blower required power is lower compared to HP EGR, since the turbocharger suction pressure is used to draw gas through the compressor. The turbine operation is less affected compared to the HP EGR case because of all the exhaust gas is passed through in order to

Table 10
MAN 4T50ME-X HP EGR NO_x reduction and Specific Fuel Oil Consumption (SFOC) data [16]

	NO _x [g/kWh]	ΔSFOC [g/kWh]	EGR [%]
Without EGR	17.8	0	0
With HP EGR (max without modif.)	2.3	+4.9	39
EGR Reference Point	3.7	+3.0	36
Tier III Setup (cylinder by-pass)	3.4	+0.6	41

obtain the energy to run the compressor. However, a turbo-matching is still needed because of the possible increased boost required during EGR operations. In addition, just a simple valve is needed to divert exhaust gas to the EGR circuit, while for HP EGR more complicated control systems are needed.

On the waste heat recovery side, possible waste heat recovery systems (e.g. steam Rankine or ORC) are penalized, due to the absence of the EGR cooler heat rejection (possible bottoming cycle heat source at higher temperature) and to the lower temperatures available in the EGR line. However, all the exhaust mass flow can be fully exploited in the economizer for steam or electricity production, and an augmented SAC heat rejection could still be available to be recovered in a bottoming cycle.

MHI [72], during their LP EGR testing campaign, reported a NO_x reduction potential up to more than 80%, thus comparable with IMO Tier III emissions regulations. At the same time, MHI declared up to 91% soot removal and up to 99% SO₂ removal potential for their LP EGR scrubbing system, and compatibility with compressor clean operations. The fuel consumption increase is also reported to be lower than 1.5–2%, but the operating point is not specified.

Finally, it is possible to observe how HP EGR should be more beneficial regarding combined emissions reduction technology and waste heat recovery systems use, mostly due to higher temperature involved in the processes and higher amount of heat available and a higher temperature. However, also LP EGR could show large benefits when considered in synergy with possible bottoming cycles, in order to recover the increase heat rejection expected at least in the SAC. Several different configurations for possible waste heat recovery bottoming cycles, such as steam Rankine and ORC, should be assessed, regarding the combined use of different heat sources (e.g. exhaust, SAC, EGR, coolant, lube oil), the choice of the right working fluid(s) and the proper cycle layout, in order to maximize the obtainable power output, with the scope of mitigating the adverse fuel consumption effect introduced with any of the considered emission reduction technologies.

3.7. Slow steaming

Even though slow steaming cannot be really considered an emission reduction technology, but rather a strategy that can be implemented by fleet and ships owners in order to reduce fuel consumption and thus pollutants emissions, it has been considered in this work due to the impact it can have on the application of waste heat recovery systems.

Indeed, the main engines, running at different load and speed points, lead to complete different boundary conditions for the proper design of the heat recovery bottoming cycle.

Slow steaming is the practice, often used in the marine shipping industry, to decrease ships speed in order to reduce fuel consumption and emissions, at the price of delivering goods later than with common full steaming practices, due to the increased travel time.

Ships fuel consumption is heavily dependent on vessel steaming speed. Increasing the speed of 2 knots can increase the fuel consumption of 50% per unit distance travelled. At the same time fuel costs represent most of the shipping operating costs, thus, usually, slow steaming strategies are applied depending on the fuel price, in order to save costs [73].

Several different considerations affect typical ships speed, for example: air resistance, wave resistance, hull frictional resistance, hull design and so on.

For these reasons, depending on the typical ship design, there is always a relationship between the ship speed and the required engine power to be supplied to the propeller. This relationship is often called “propeller law” (for fixed pitch propeller) and can be written as [32,74]:

$$\dot{W} = K \cdot N^3 \quad (1)$$

where \dot{W} is the propulsion power, K is the propeller law constant and N is the engine rotational speed.

A MAN survey analysis [75], conducted with 149 interviews of shipping container fleet respondents, showed how ships using slow steaming operations are running their engines at loads between 30 and 50%, while the majority of ship fleet companies which answered the survey declared also to combine slow steaming with full steaming operations, with only 6% employing only slow steaming, this leading to the need of designing engines capable of operating over a wide range of operational points in an effective way, as so the heat recovery systems.

Slow steaming is beneficial regarding fuel cost savings, but can lead to engine operating problems as for example: fouling of the exhaust boiler, low temperature in the exhaust gas boiler affecting possible waste heat recovery potential, soot deposit on moving parts, premature wear and tear of engine parts, under or over lubrication, lower engine performance and combustion efficiency, performance and combustion efficiency loss due to low-quality fuel.

As reported by MAN [75], some measures for implementing slow steaming strategies can be: turbocharger cut-out, slide fuel valves,

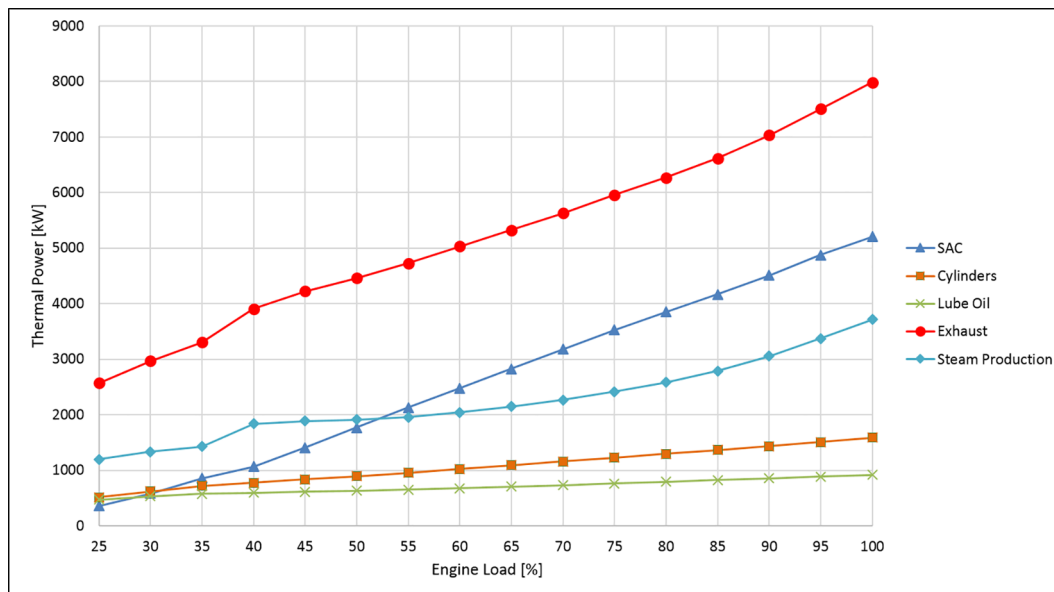


Fig. 9. Heat rejection and thermal power for different engine loads, for a WINGD RT-flex58T-D, two-stroke 13.6 MW engine [10].

engine derating, cylinders deactivation, propellers upgrade, lubrication oil system upgrades.

Slow steaming operations have for sure impact on possible waste heat recovery systems, since lower scavenging and exhaust gas mass flows and temperatures lead to lower waste heat recovery potential, and even sometimes the economizer should be by-passed due to too low exhaust temperatures and possible fouling problems. For ultra-slow steaming, auxiliary blowers could also be needed during operations in order to supply the right combustion air. Some typical heat sources data have been reported for different engine loads and speed for a two-stroke propulsion unit in Fig. 9, showing heat recovery potential at different operating points, especially when considering slow steaming strategies. The data have been reported for a WINGD 6RT-flex58T-D, two-stroke, 13.6 MW brake power engine, with single stage turbocharger for IMO Tier II operations, and have been obtained using the engine performance analysis WINGD GTD software developed by the engine manufacturer [10].

As it can be observed in the graph in Fig. 9, with a reduction in engine load down to 30–50% there is a sensible reduction in the thermal power available in all the possible heat sources that can be considered for waste heat recovery purpose. The «Cylinders» heat source can be considered, in first approximation, as an estimation of the heat that is transferred from the cylinders to the coolant of the engine, thus being an estimation of the coolant thermal power available to be recovered with possible waste heat recovery systems.

The engine fuel consumption benefit can be evinced from the Table 11 and Fig. 10 has also been reported, showing a benefit especially when considering 50% load point. The engine, as already introduced, could be further optimized in order to run at slow steaming conditions. The data reported in Table 11 and Fig. 10 are not for a slow steaming optimized engine.

The steam production power term is calculated with an economizer outlet temperature of 170 °C, and the data reported in Table 11 are a comparison with the 100% load point, showing a marked reduction of heat rejection for all possible bottoming waste heat recovery suitable heat sources.

3.8. Comparison between emissions reduction technologies

From the literature data considered in this work, the Table 12 has been produced, and can be considered as a comparison among the described technologies and strategies, focusing the attention on the NO_x and SO_x emissions reduction potential, the technology maturity, the fuel efficiency, costs and impact on waste heat recovery technologies usage. The data reported in the table can also be extracted from some interesting reviews reported in references such as [22,23,25,28,30,31].

4. Waste heat recovery in ships

For most HDDE used for ship propulsion and power generation, a maximum of around 50% of the fuel energy is converted into useful shaft power for propulsion or electricity production, while the

Table 11

Heat rejection and BSFC comparisons between 30% and 50% loads (referred to 100%) for a WINGD RT-flex58T-D two-stroke 13.6 MW engine [10].

Load	30%	50%
BSFC [g/kWh]	0.1%	-2%
SAC [kW]	- 89%	-66%
Cylinders [kW]	- 61%	-44%
Lube Oil [kW]	- 42%	-32%
Exhaust [kW]	- 63%	-44%
Steam production [kW]	- 64%	-49%
Exhaust temperature [°C]	- 1%	-4%
Exhaust mass flow [kg/s]	- 62%	-41%

remaining is lost through the exhaust gas, CAC or SAC, cooling circuit, lubrication circuit or to the ambient through radiation [76,77].

In case of ships, it is easier, compared to automotive applications to recover engines wasted heat, due to the more stable operating profile. Indeed, engines, both for propulsion and auxiliary power generation, tend to run for longer periods at constant load and speed, compared to what happens in vehicles, unless some particular applications are considered, such as long-haul trucks or some off-highway commercial vehicles.

Literature about waste heat recovery on board ships, with particular focus on two-stroke propulsion units, is rather sparse [76], probably mostly due to the fact that two-stroke engines show lower potential than four-stroke engines (e.g. on-board auxiliary units) when recuperating exhaust gas heat, due to the lower temperature levels (Table 4) and to frequent utilization in shipping of heat recovery steam generators. The general trend is using a gas Power Turbine (PT) and/or a Steam Turbine (ST) as main waste heat recovery systems, together with steam production for general ships usage. A combination of the technologies can also be proposed as reported, for example, by MHI [78].

In the power turbine case, the kinetic energy of the engine exhaust gas is used to drive a gas turbine. A parallel configuration with the turbocharger, or a series turbocharger-PT configuration are possible, depending on the engine architecture. By-pass valves can be used to control the flow through the turbocharger and PT. At partial load, where most of the gas flow is needed to drive the turbo, the exhaust gas flow could be insufficient to operate also the PT. Mechanical or electrical coupling configurations with the main engine could be used. Compared to ST, the PT has a lower capital cost (CAPEX) and shorter construction time and maintenance [76].

As reported by Choi *et al.* [79] the steam turbine (ST) configuration usually uses low temperature exhaust gas around 270 °C (after the turbine) as a heat source. Superheated steam is produced exploiting a multi-step heat exchange through a HP and LP feed water heater, using then a mixed-pressure turbine.

Ma *et al.* [80] proposed a single pressure steam Rankine system, evaluating both design and part-load conditions, for a 9K98ME-C7 MAN two-stroke engine. First and second law thermodynamic analysis have been considered. The results showed 8 bar as the most appropriate boiling pressure, showing an increased engine efficiency from 48.5% to 53.8%.

Dimopoulos *et al.* [81] proposed a detailed thermo-economic optimisation of a waste heat recovery system for a two-stroke engine for container ships propulsion, while again Dimopoulos *et al.* [82] also investigated the exergy analysis as a way to optimize and further understand the combined Diesel engine-waste heat recovery system. Dimopoulos *et al.* [83], proposed also a tool, DNV COSSMOS, to model, simulate and optimise marine energy systems from and energy and

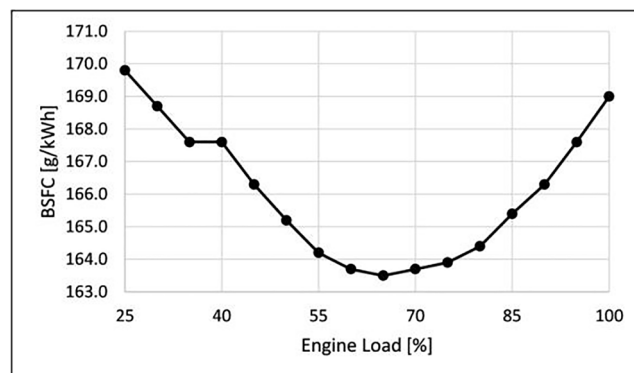


Fig. 10. BSFC [g/kWh] at different load points for a WINGD RT-flex58T-D two-stroke 13.6 MW engine [10].

Table 12
Comparison among emissions reduction technologies and strategies.

Emission Reduction Technology/ Strategy	NO _x reduction potential (%)	SO _x reduction potential (%)	Technology Maturity	Fuel Efficiency Impact (%)	Initial Cost (CAPEX)	Operational Cost (OPEX)	WHR System Impact	Refs.
Optimization Engine Internal Parameters	20-30	/	Conventional	+0 ÷ 3	Low	Low	Neutral	[26,27,40]
Miller Timing + 1 Stage Turbocharging	40-50	/	Conventional	+2 ÷ 3	Low	Low	Negative	[39,42]
Miller Timing + 2 Stage Turbocharging	40-50	/	Conventional/Not very spread	+1 ÷ 2	Medium	Medium	Negative	[38,39,42]
Water-in-Fuel-Emulsion (WFE)	15-50	/	Developed/limited long-term experience	+2 ÷ 3	Medium	Medium	Neutral	[19,46,48]
Direct Water Injection (DWI)	40-50	/	Developed and used (especially in medium speed 4 S engines)	+7 ÷ 10	Medium	Medium	Neutral	[19,46,48]
Humid Air Motor (HAM)	10-40	/	Developed/Limited long-term experience	+2 ÷ 3	Medium	Medium	Negative	[19,46,48]
Dual/Multi-Fuel (LNG) - Diesel/Orto	25-85	92-99	Developed/trial phase, first commercialization	Depending on combustion	High	High	Neutral	[43,50-54]
HP EGR + Scrubbing	80-93	90-99	Developed/first commercialization	+1 ÷ 3	High	Medium	Positive	[14,16]
LP EGR + Scrubbing	80-85	90-99	In development/testing phase	+0 ÷ 2	High	Medium/High	Neutral/Positive	[71,72]
SCR (LP & HP)	80-95	/	Developed/first commercialization	+0 ÷ 2	High	High	Negative	[14,16]
Slow Steaming	Depending on operation	Depending on operation	In use	-2 ÷ 0	Low	Low	Negative	[73,75]

thermo-economic point of view.

Several configurations are available from the main two-stroke engines manufactures for waste heat recovery using PT and/or ST. Wärtsilä [19], for example, proposed mainly three different configurations, considering service steam production only, service steam production and a ST or a combined steam production ST-PT system, evaluating the possibility of producing electricity or re-inserting the generated power to the propeller shaft through the combined use of an electrical motor, and declaring a possible overall engine system thermal efficiency increase up to 11.4%. MHI [78] also developed a combined PT-ST waste heat recovery system. For a 45.7 MW two-stroke engine they declared 5.5% maximum increased power production due to the ST, and 3.7% maximum power increase due to the PT.

MAN [15] also proposed mainly two configurations: a configuration with PT (or ST) only with steam production, and a configuration with combined PT-ST system, with a double pressure (or a single pressure) steam evaporator. Some results [84] have been reported in Table 13, regarding the combined PT-ST system for a two-stroke MAN S60ME-C8.5 engine in IMO Tier II configuration, depending on the engine size, showing a possible increase in produced power between 7.3 and 8.3% compared to the baseline engine.

Larsen *et al.* [85] proposed a comparison between advanced heat recovery power cycles used in combined cycle configurations for large ships powered by two-stroke Diesel engines. In particular, a Kalina cycle, a steam Rankine and an ORC have been evaluated, recovering exhaust gas and CAC heat, showing how, in terms of performance, the Kalina cycle has no significant advantages compared to the ORC and the steam Rankine cycle, introducing even a more complicated system layout. The ORC, using R245ca as working fluid, contributed with 7% power increase compared to the baseline engine, while the Kalina and steam Rankine 5%. A power turbine generator has also been considered leading to only 2.5% power increase.

As reported by Shu *et al.* [76] and Singh *et al.* [77], other waste heat recovery technologies can be used in ships applications such as re-refrigeration (absorption or adsorption cycles) [86,87], Thermo-Electric-Generation (TEG) and desalination, while as already introduced above, also optimized Kalina cycles [8,85,88,89] and Goswami cycles [90-92], could be used and developed in order to exploit wasted engine heat to produce additional power or combined power and cold energy for general ships usage.

Recently also ORC systems have been introduced in marine applications, in particular to consider low temperature heat sources in synergy with the already used PT and steam Rankine Cycles (or ST), or in order to be used instead of the steam turbine, fitted on the low temperature two-stroke exhaust line [76,93]. An overview of studies about this type of systems for marine application is reported in the next section.

5. Organic Rankine cycles (ORC)

The ORC is very similar to the well-known steam Rankine cycle, with the difference that instead of using water steam, usually an organic fluid, as for example a refrigerant or a hydrocarbon, is used in order to recover heat at lower temperatures.

Many articles have been published in literature about ORC as waste heat recovery systems for internal combustion engines but only some works are available about systems on-board ships, since the application is still not well spread in the market but could show potential for future implementation.

Moreover, most on the works are related to heat recovery from four-stroke internal combustion engines for auxiliary power generation, while only few papers regard two-stroke ship propulsion units.

In this section of the paper, an overview of the literature regarding mostly two-stroke marine engine waste heat recovery is proposed. Furthermore, some ORC architectures have been discussed, considering advantages and disadvantages, especially regarding engines

Table 13

Recoverable power with PT-ST combined configuration for a MAN S60ME-C8.5 two-stroke engine [84]

Cylinders Number	Engine Power		Power Turbine (PT)		Steam Turbine (ST)		Full WHRS with combined PT-ST	Percentage compared to engine power
	%SMCR	kW	kW _e	%	kW _e	%	kW _e	%
5	100	11,900	431	3.6	595	5.0	978	8.2
	75	8,295	274	3.3	447	5.4	651	7.8
6	100	14,280	520	3.6	763	5.3	1,180	8.3
	75	10,710	334	3.1	543	5.1	787	7.3
7	100	16,660	610	3.7	896	5.4	1,383	8.3
	75	12,495	397	3.2	640	5.1	925	7.4
8	100	19,040	701	3.7	1032	5.4	1,588	8.3
	75	14,280	461	3.2	739	5.2	1,064	7.5

configurations with EGR.

A recent overview about ORC waste heat recovery in marine applications can be found in Mondejar *et al.* [94]. The differences compared to the review proposed in this paper are mostly related to the fact that emission reduction technologies are not considered, as well as the possible relationship and synergy with waste heat recovery systems. In particular, in this section different combined engine-ORC system architectures have been considered, focusing the interest on those involving the recovery of EGR and SAC heat, which seems to have potential for further improving overall system's efficiency.

The first ORC installed on-board ships has been used to recover heat from the engines of car-truck carrier ship as reported by Öhman *et al.* [95], using an OPCON/Powerbox [96] unit, running with R236fa as working fluid (now banned), engine cooling water as heat source and LT cooling water as heat sink, expecting a 4–6% fuel saving.

Burel *et al.* [43] analysed the possibility to install an ORC in a tanker where LNG is used as propulsion fuel, while Larsen *et al.* [97] proposed a methodology based genetic algorithms, to optimise working fluid selection, boiler pressure and Rankine cycle process for marine engine heat recovery.

Again Larsen *et al.* [85] proposed a comparison of advanced heat recovery power cycles for large ships, modelling the systems in MATLAB environment, using a genetic algorithm for the optimisation procedure, and concluding that a Kalina cycle has no significant advantages on the ORC and steam Rankine systems.

Bonafini *et al.* [98] proposed a study about recovering waste heat from the exhaust gas of marine dual-fuel engine with power output of 5.7 MW. The selected working fluid is toluene and a simple cycle architecture has been considered the most interesting in terms of increased power output benefits. An economic analysis was also carried out.

Baldi *et al.* [99,100], in two different works, proposed the use of optimization techniques for Diesel engine-ORC waste heat recovery systems based on the analysis of typical ships operating profiles. The case studies use, as baseline engines, MaK 8M32C four-stroke Diesel engines with a power output of 3840 kW and some auxiliary units of about 683 kW. Fuel saving potential is considered for some typical vessels' applications.

Song *et al.* [101] 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 produced by Hudong Machinery Co., Ltd. Economic evaluations as well as off-design conditions are considered. An optimized system using cyclopentane, cooling water as preheating source and exhaust gas as evaporating source for the working medium is proposed, obtaining only around 1.4% lower power output compared to the separated bulkier systems.

Yun *et al.* [102] proposed a study about a dual loop ORC system with the aim of recovering waste heat in parallel from the exhaust gas of marine Diesel engines, with the highlighted benefit of being more versatile when operating at off-design conditions. The conclusion is that the dual-loop ORC has a power output that is between 3 and 15% higher than a simple single loop system.

Yfantis *et al.* [103] proposed a thermodynamic model to study the first and second law of thermodynamics performance characteristics of a four-stroke marine Diesel engine equipped with a Regenerative Organic Rankine Cycle (RORC) to recover exhaust heat. Different engine operating loads are investigated, as well as R245fa, R245ca, isobutane and R123 as working fluids. A subcritical and saturated vapour regenerative cycle is found to have the best performance both from first and second law point of view.

Soffiato *et al.* [104] proposed an ORC system to recover cooling jacket water heat, lubricating oil and CAC of the engines of a LNG carrier ship. In this case 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 configuration and showing double the potential of the other architectures, but with higher structural complexity and reliability issues.

Sciubba *et al.* [105,106] analysed the comparison between a single-loop and a dual-loop waste heat recovery system for different power range marine engines (a yacht 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. Regeneration is also proposed to improve system efficiency.

Michos *et al.* [107] 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 Turbine Geometry (VGT), have been investigated in order to counterbalance the detrimental effect of the increased exhaust line backpressure. Simple and recuperated ORC architectures have been investigated, through simulation, in order to assess the combined engine-ORC fuel economy improvement. A combined engine-ORC system using VGT turbine and acetone as ORC working fluid has been considered the most promising, leading to a possible improved fuel efficiency between 9.1 and 10.2%, depending on the ORC boiler engine backpressure.

Mondejar *et al.* [108] proposed a quasi-steady state analysis of an ORC system recovering heat from the main engines of a passenger vessel over a typical North Sea operating scenario, obtaining as a result that the heat recovery system should be able to cover up to the 22% of the required onboard average electrical power demand (hotel load).

Uusitalo *et al.* [109] conducted also an analysis based on a typical cruise ship operating profile, but recovering the steam cycle condensed heat with an additional ORC loop.

Two-stroke ship propulsion units have also been considered in waste heat recovery studies, even if a smaller number of papers has been published.

Hountalas *et al.* [110] 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. Pressure drop increase

on the gas sides has also been considered.

Choi *et al.* [79] 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 *et al.* [111] analysed the possibility of recovering jacket cooling heat of a large marine Diesel engine. Results show that R600a performs in the best way, followed by R1234ze, R1234yf, R245fa, R245ca and R1233zd, with very low evaporation temperature (58–68 °C).

Wang *et al.* [112] simulated and analysed an ORC-desalination combined system driven by the SAC (Scavenging Air Cooler) heat of a two-stroke MAN 12S90ME-C9.2 ship engine, using R245fa as a working fluid for the ORC and obtaining up to almost 2800 kW and 245 t/day of desalinated water.

Grlijsic *et al.* [113,114] proposed a supercritical ORC system operating with R123 or R245fa to recover scavenge air (SAC), jacket cooling water and exhaust gas of a two-stroke 18.7 MW 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.

Yang *et al.* [115] evaluated the economic performance of a Transcritical Rankine Cycle (TRC) using different low temperature suitable working fluids (R1234yf, R1234ze, R134a, R152a, R236fa and R290) obtaining the best results, and lowest levelized energy cost, with R236fa. Payback period, fuel oil saving, and CO₂ emission reduction are also evaluated. Considered heat sources are exhaust gas, cylinder HT cooling jacket, scavenge air and lube oil.

Larsen *et al.* [116] also proposed a new concept of 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 103 different fluids have been evaluated, obtaining the best results with 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 considered feasible.

Andreassen *et al.* [117] proposed a comparison between organic and steam Rankine cycle for waste heat recovery on large ships, obtaining the best results with steam as working fluid, at high load conditions, and best results with an organic medium at low load points. A turbine type expansion machine is considered in the study, and some preliminary design considerations have been proposed.

Kyriakidis *et al.* [118] presented a work regarding the theoretical optimization of two different concepts of steam Rankine cycle for a 23 MW two-stroke marine Diesel engine, considering also an HP EGR configuration and optimizing the cycle pressure levels, obtaining around 1.6 MW useful power output.

Suarez de la Fuente *et al.* [119] proposed an analysis of ORC recovering waste heat from a two-stroke marine engine operating in the Arctic Sea, an exploiting the possibility of using cold air or sea water as cooling fluids, thus improving the system thermal efficiency, due to the low ambient temperature conditions. An additional paper on the topic has been also proposed in [120], considering the forward movement of the ship in order to support the air cooling of a possible ORC system,

thus reducing fan parasitic power consumption.

Akman and Ergin [121] analysed the possibility of using different ORC systems concepts to recover the heat from a two-stroke MAN 68.6 MW large engine, obtaining the boundary conditions for the heat sources from the CEAS MAN software, and recovering jacket cooling water, scavenging air and exhaust gas heat. A possible 6.9% reduction in ship emissions has been calculated.

Baldasso *et al.* [122] proposed a study regarding a 2500 TEU regional feeder ship powered by a 10.5 MW MAN two-stroke engine using LNG as a fuel, and considering configurations both with LP and HP SCR or EGR (not specified if HP or LP). The results obtained shows better installations in terms of ORC costs and performance when using LP SCR or EGR systems.

When considering ORC to recover wasted heat from large low speed two-stroke marine Diesel engines, some peculiarities must be considered compared to four-stroke engines. In particular, as reported in Table 4, exhaust gas temperatures are much lower in two-stroke engines compared to four-stroke, even though mass flows are higher because, often, the size of these engines is larger, thus making heat recovery, in terms of energy, interesting.

Moreover, recently, some ORC manufacturers and shipping companies are considering using ORC technology in order to recover two-stroke waste heat. In particular, the challenge is recovering jacket coolant water of the engines, in the 80–90 °C temperature range, which seems to become more attractive as a heat source, because basically freely available, easy to use and with less issues of flammability concerns in comparison to the gas exhaust line, in which a leakage of the working fluid from the ORC plant could become a problem.

Some ORC commercial-ready products for marine engine applications, and in particular two-stroke low speed engines, targeting mostly low temperature heat sources, have been reported in Table 14, even though the market in this sector can still be considered in an early phase of development.

In order to assess the potential of low temperature ORC systems for two-stroke engines, some preliminary calculations have been performed, using a process simulation model developed in Engineering Equation Solver (EES) [128], for an engine cooling water driven ORC based on the data obtained for a 6 cylinders WINGD RT-flex58T, 13.6 MW brake power two-stroke low speed marine Diesel engine. The boundary conditions for the study have been taken from the WINGD GTD online software [10], concerning especially temperatures and mass flows for the high temperature and low temperature cooling circuits of the engine, based on suggested design conditions at engine full load.

In particular, two different possible schemes have been evaluated: a scheme with the ORC system in parallel to the low temperature sea water cooler (Fig. 11-1) and a scheme with an ORC recovering directly the HT jacket coolant heat in a configuration to by-pass the conventional HT-LT jacket water cooler (Fig. 11-2). The second configuration has been considered the most promising because of the higher temperature available (90 °C) and the still high amount of volume flow of coolant available to be recovered (at nominal conditions). A more detailed explanation of the thermodynamic models evaluated can be found in previous authors' works [66,107].

A genetic algorithm has been used in order to maximize the ORC power output based on three different main independent variables: ORC working fluid mass flow, pump pressure ratio and superheating degree. The condensing pressure has been fixed based on the amount of sea

Table 14
Commercial-ready ORC products for marine engine low-medium temperature WHR.

Company	Model	Heat Sources	Working Fluid	Power Output [kW _e]	Refs.
Opcon Marine	Powerbox™	coolant, SAC, exhaust gas	Ammonia	400–800	[6,96]
Calnetix	Hydrocurrent™ 125 EJW	coolant	R245fa	125	[123–126]
Climeon	Ocean™	coolant, LT exhaust (after economizer)	n.a.	150–1000	[127]

cooling water (100 kg/s) and considering an inlet sea cooling water temperature of 32 °C (nominal conditions). A more accurate simulation considering also the variable sea cooling water mass flowrate and temperature should be carried out in a following phase of the research, when considering an overall system approach.

Pinch points in the heat exchangers have been constrained not to be lower than 5 °C (liquid–liquid heat exchangers), while pump efficiency has been set to 70%, turbine expander efficiency to 80% and the electric generator efficiency to 95%. The HT engine coolant at the outlet of the ORC evaporator has been set to 75 °C as minimum allowable level. The design optimization requires, indeed, that the coolant at the engine inlet has a temperature of 75 °C.

R245fa and R1233zd(E), suitable for low temperature ORC, have been tested with the process simulation model, and the results have been reported in Table 15. In particular, R1233zd(E) seems to be one of the most suitable replacements for the R245fa, due to the similar thermodynamic properties [129] and the lower environmental impact (lower GWP, Global Warming Potential).

The optimisation results show how, for the considered boundary conditions, the two fluids show very similar thermodynamic performance, with a power output of 116 kW_e.

Just for a matter of comparison, the concept (1) of Fig. 11 with R245fa, leads to only around 33 kW_e power output.

Also, a model considering an ORC like the one of concept (2) in Fig. 11 but with regeneration has been simulated. Cooling the HT jacket water down to 75 °C, and optimizing the cycle parameters, very similar results in terms of electric power output compared to those regarding the proposed layout without regeneration have been achieved. For this reason, the use of an additional heat exchanger is not worth to be considered, leading this to an increase in the overall system costs.

In general, however, the high volumes of cooling water used for large two-stroke ship diesel engines, can carry a quite high amount of heat, even if a low temperature, which could be exploited with ORC systems, in order to further produce electricity for auxiliary loads. The trend for this kind of developments is also envisaged by the data reported about manufacturers and developers of these systems in Table 14.

Due to the high amount of gas mass flow available in large two-stroke low speed marine Diesel engines, the Scavenge Air Cooler (SAC) heat rejection becomes a potentially useful heat recovery source for low-medium temperature ORC systems, as reported, for example, in Fig. 8, which shows high amount of heat dissipated in the SAC.

Additionally, when considering, for example, emission reduction technologies such as, in particular, Exhaust Gas Recirculation (EGR), the recirculated exhaust gas, which has to be cooled down in the SAC to temperatures compatible with the engine intake boundary conditions for the combustion process, tends to even increase the available heat which can be recovered in the scavenge air cooling process, and this is usually valid both for the HP and LP EGR configurations. Moreover, when considering HP EGR, an additional medium–high temperature heat source can be available from the HT EGR cooler in order to increase the ORC net power output possibilities and, thus, the overall system performance. In case of LP EGR system, the heat rejected to the WTS system could be also exploited, but the process could become more complicated due to the water cleaning treatment system. For the reasons described, more advanced ORC layouts and concepts could be investigated, considering also SAC and EGR as heat sources, in combination with coolant and exhaust gas. Some of these concepts can be observed in Fig. 12.

In Fig. 12(a), a configuration with exhaust gas evaporation and HT coolant preheating has been reported. In Fig. 12(b), a configuration with parallel boilers evaporation can be observed, in order to take profit of both exhaust gas and HP EGR cooler heat sources, using a unique evaporation pressure controlled by a pump, in order to recover heat from two heat sources characterized by similar temperature levels.

In Fig. 12(c) an innovative concept with two-stage SAC has been

presented. In this case, the HT jacket coolant is used to recover heat from the first stage SAC, in order to increase the cooling fluid temperature. The coolant is then sent to the ORC in order to produce useful power. This configuration could be useful regarding also engine thermal management benefits with possible reduction of cooling circuit auxiliaries power consumption. The second stage SAC is used to allow to reach the required scavenging air temperature for a proper combustion. The Tier III operations with, in this case, LP EGR, increases the heat which can be recovered in the SAC. This system has been quantitatively investigated by the authors in [130], considering operations with (Tier III) and without (Tier II) LP EGR. The system has been considered innovative and with the potential of improving a traditional coolant-only configuration.

In Fig. 12(d), a layout with two-stage SAC is considered, together with a dual loop (or cascaded) ORC system, in order to recover also exhaust gas. In this case, the dual loop architecture allows a better exploitation of two different temperature levels heat sources, but with a rather more complex system. In all cases with exhaust gas recovery, the working fluid could be water steam (as in steam Rankine cycles already used in ships) or another ORC fluid suitable for medium–high temperature heat sources recovery.

All the systems proposed are mostly using a combination of exhaust gas, EGR and coolant heat. In particular, the recovery of EGR heat is not only beneficial in terms of ORC performance, but also in terms of overall engine thermal management, recovery heat that otherwise would have to be dissipated in the cooling system and converting it to useful power. This makes the overall engine system at the same time more efficient and cleaner, from an emissions' point of view.

A configuration with SCR on the exhaust could be also considered. In this case, a careful positioning of the ORC exhaust gas heat exchanger must be investigated, due to light-off and temperature constraints of the SCR system itself.

In general, when considering a combined engine–ORC configuration, the overall system could be optimized, in terms of layout and operating parameters, in order to achieve the best compromise in terms of power and emissions.

6. Conclusions

The work proposed in this paper aims to present the emissions mitigation technologies and strategies which are currently applied or under development in the marine sector, for heavy duty Diesel engines used for ship propulsion or auxiliary power generation.

The interest is mostly focused on two-stroke propulsion units used for container ships, bulk carriers and oil tankers, which, from an analysis of IMO GHG reported data, seem to be the most polluting type of ships in the marine sector, for what concerns, in particular, CO₂, NO_x and SO_x. These ships are also among those responsible of most of the fuel consumed in the marine shipping industry. In principle, similar technologies can be applied also to medium–high speed four-stroke engines, considering different operational boundary conditions.

The scope of the work has been to assess the various technologies from a technical point of view, highlighting the advantages and the drawbacks of most of them, considering, in particular, a synergy with possible waste heat recovery systems, which can be used to improve

Table 15
HT jacket cooling water ORC WHR for a WINGD RT-flex58T two-stroke low speed engine. Process simulation results with R245fa and R1233zd(E).

Working Fluid	$\dot{m}_{f,wf}$	$\dot{m}_{f,sw}$	P_{cond}	P_{ratio}	ΔT_{suph}	η_{ORC}	$\dot{W}_{ORC,net}$
	[kg/s]	[kg/s]	[bar]	[-]	[°C]	[%]	[kW _e]
R245fa	8.8	100	2.6	2.6	5.9	6.5	116
R1233zd(E)	8.6	100	2.2	2.5	12.1	6.5	116

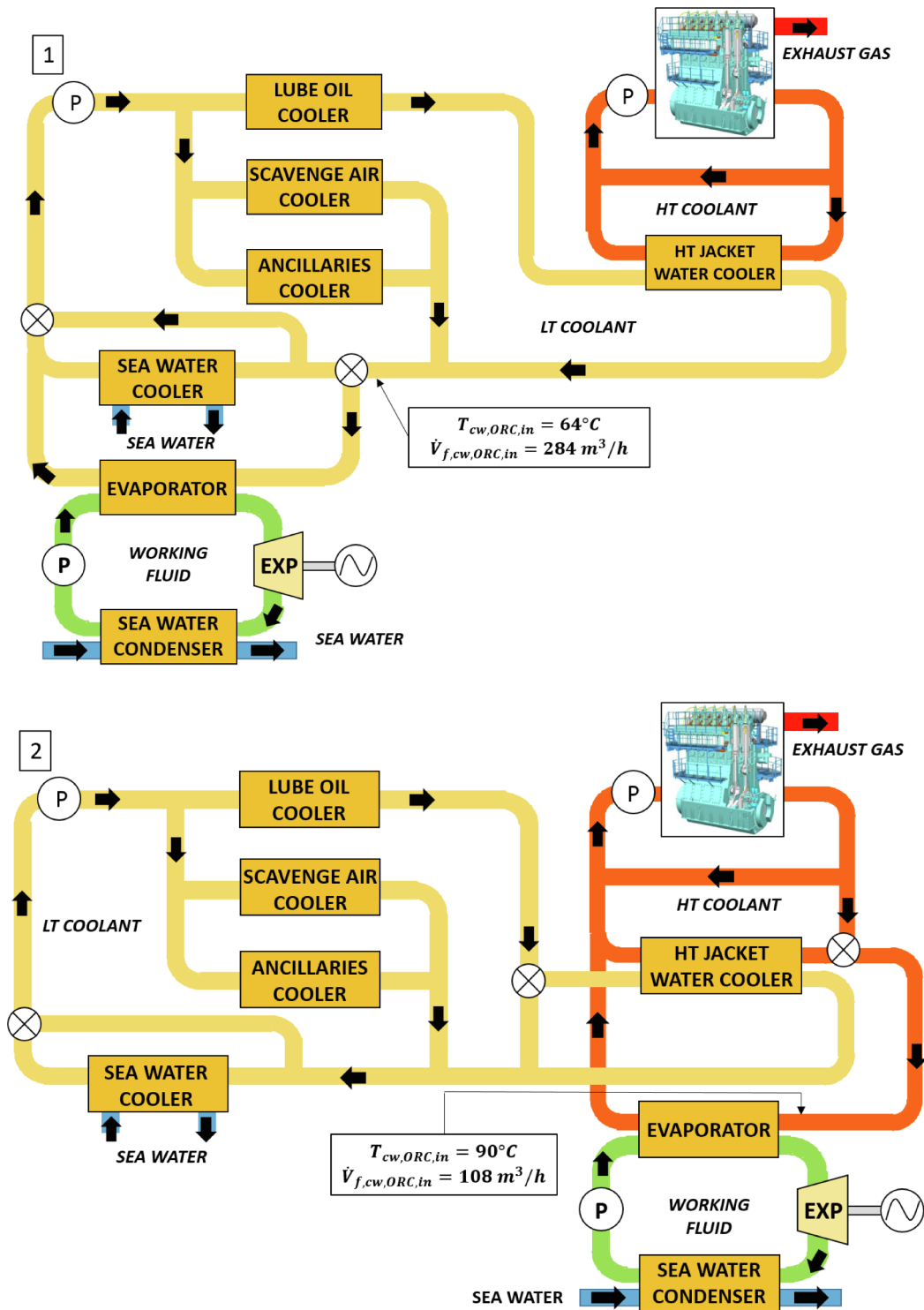


Fig. 11. ORC configurations for coolant water WHR of a WINGD RT-flex58T-D 13.6 MW two-stroke engine. Base cooling fluid circuit configuration elaborated from WINGD GTD [10].

fuel economy, thus evaluating the engine and ORC not anymore as separated systems, but rather as a combined entity which should be optimised in terms of performance and under emissions constraints.

Power Turbines (PT), Steam Turbines (ST), steam production and waste-heat-based desalination systems seem to be, now, the most used waste heat recovery technologies in the marine sector.

However, Kalina cycles, Goswami cycles, TEGs and ORC seem to have potential to further improve ship efficiency, taking profit of all the

heat sources available from the heat wasted from the main and auxiliaries' engines, due to their innate cycle efficiency limitations. Between these last, ORCs have good potential, in terms of performance and system costs and dimensions.

All emissions reduction technologies considered in this work lead to a certain fuel economy penalty (usually in the range between 0 and 10%), but with the benefit of decreasing pollutants released into the environment. However, they could also lead to benefits for what

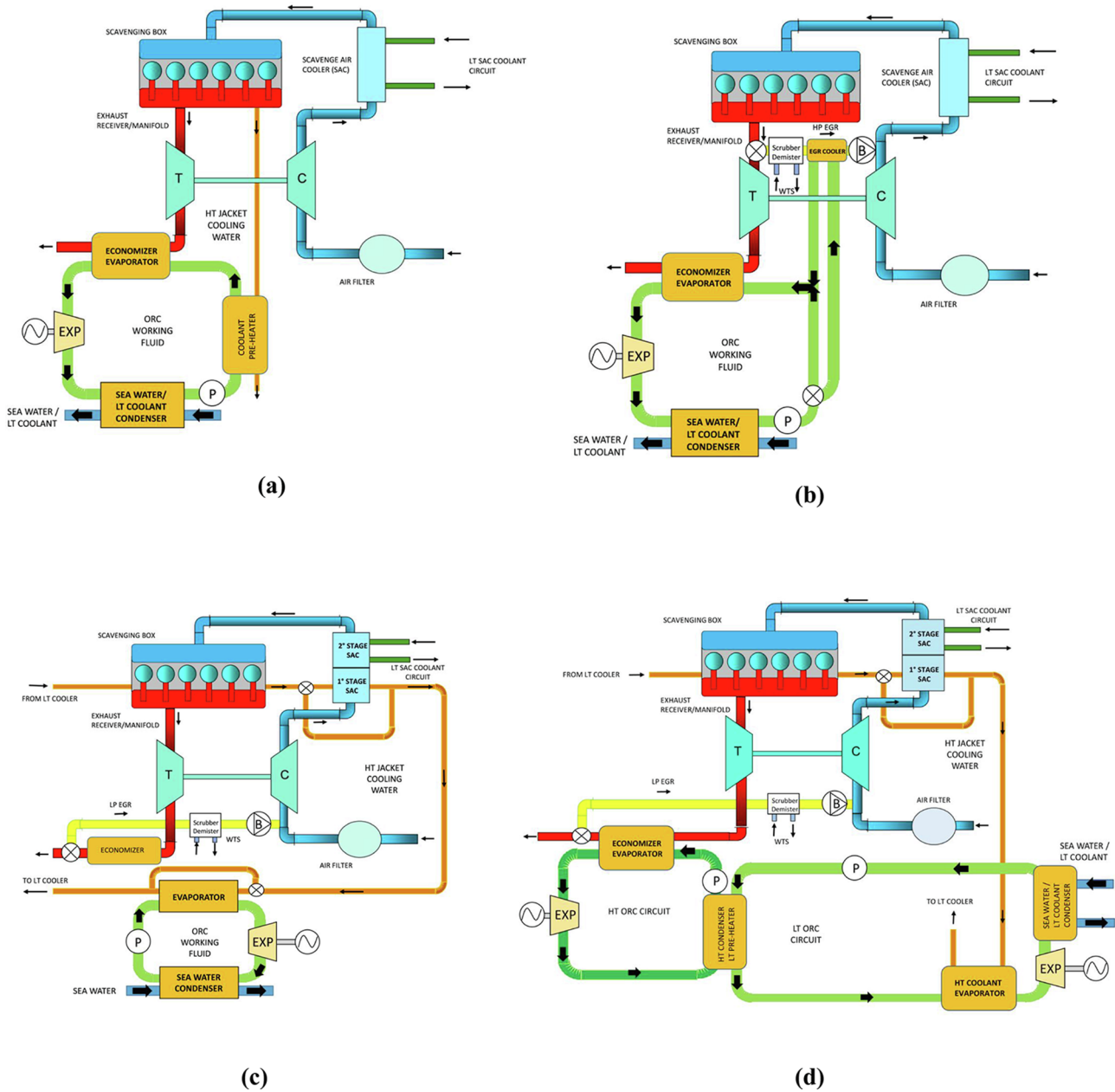


Fig. 12. Different ORC configurations for two-stroke low speed engines waste heat recovery in Tier II and Tier III configurations. Exhaust gas evaporation with HT coolant pre-heating (a), Parallel boilers configurations with exhaust gas and HP EGR evaporation (b), 2-stage SAC for combined HT coolant and SAC heat recovery (c), 2-stage SAC, dual loop system, for combined recovery of exhaust gas, HT coolant and SAC (d).

concerns the utilization of waste heat recovery systems. For this reason, a combined use of this technologies and waste heat recovery systems can mitigate the fuel consumption disadvantages with the main goal of developing more environmentally friendly but, at the same time, efficient propulsion and energy generation systems.

In this scenario, and for two-stroke low speed Diesel engines, the use of systems such ORC leads to the possibility of recovering different engine wasted heat sources, some of them usually considered at too low temperature for effective use in traditional steam power generation (e.g. engine coolant or SAC heat).

Moreover, due to new IMO Tier legislations, these systems must be developed considering flexibility of operation (for example outside and inside ECAs), reliability and cost effectiveness, while fuel reduction operational strategies, such as slow steaming, impose new constraints in the design of the recovery systems, which must be capable of giving real cost and performance benefits also when the boundary conditions are

not the optimum over all ships' operational regimes.

Some manufactures are starting to introduce prototypes and systems for marine applications, however ORC technology is not yet spread into the marine market, as it is for land-based power generation, thus high potential for architectures, working fluids and performance optimisation is still available in order to further increase overall system efficiency. However, cost effectiveness and system reliability must still be completely proven.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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