Chapter 7
Natural Gas Combustion in Marine Engines: An Operational, Environmental, and Economic Assessment

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Abstract The present study contains a detailed assessment of the state-of-the-art technologies of two-stroke (2-S) and four-stroke (4-S) dual-fuel compression-ignition (CI) engines and four-stroke spark-ignited (SI) natural gas engines from technological, operational, environmental, and economic standpoints. Emphasis will be given to the examination of the effect of natural gas combustion on the performance characteristics and pollutant emissions of marine two-stroke dual-fuel engines and four-stroke dual-fuel and gas SI engines. Also, the CO_2 and CH_4 using EEDI analysis are examined for LNG carriers equipped with three different propulsion systems. The final outcome of the proposed study will be the definition of the parameters that should be taken into account to identify the optimum two-stroke and four-stroke natural gas engine technology frame, which can be used in the near future, as either main propulsion (two-stroke or four-stroke) or auxiliary (four-stroke) engines in marine applications.

Keywords Natural gas · Marine engines · Combustion · Emissions

7.1 Introduction

As known, both sulfur oxides (SO_x) and oxides of nitrogen (NO_x) are two critical gaseous emissions, which have a serious detrimental effect on photochemical smog and through this, on the formation of acid rain (Dieselnet 2017; Cleantech 2017; International Maritime Organization 2017). Also, SO_x and NO_x emissions have a

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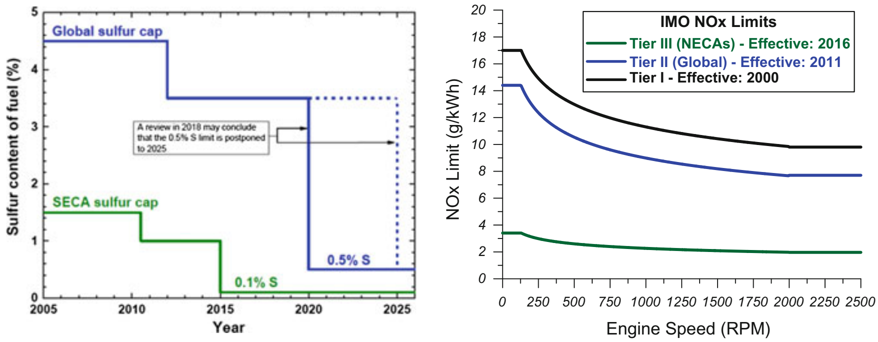
direct hazardous impact on human health since they are causing respiratory problems. Finally, NO_x emissions contribute significantly to eutrophication of lakes and rivers (Dieselnet 2017; Cleantech 2017; International Maritime Organization 2017). Hence, the International Maritime Organization (IMO) having acknowledged all the aforementioned negative influences of NO_x and SO_x emissions both on environment and on human health and having also convinced about the continuous deterioration of greenhouse effect has taken specific actions during recent years. Specifically, aiming to a significant curtailment of shipping-emitted gaseous pollutants, IMO has issued, during recent years, specific standards for SO_x and NO_x concentration values emitted from marine engines (Dieselnet 2017; Cleantech 2017). IMO has also launched specific measures in the form of efficiency design index (Energy Efficiency Design Index—EEDI) for new ships built after January 1, 2013, and in the form of efficiency operational index (Energy Efficiency Operational Index—EEOI) for existing ships for controlling greenhouse gas emissions with a primary interest for CO_2 emissions (Dieselnet 2017; International Maritime Organization 2017).

According to IMO policy (International Maritime Organization 2017), shipping SO_x emissions are primarily controlled by fuel sulfur content. Figure 7.1a illustrates the chronological evolution from 2005 until 2025 of IMO-legislated global fuel sulfur content limit and the pertinent fuel sulfur limit, which is mandated in SO_x Emission Control Areas (SECAs) (Dieselnet 2017; Cleantech 2017; International Maritime Organization 2017). SECAs currently include Baltic Sea, North Sea, English Channel, and a 200 miles zone from North America coastline (International Maritime Organization 2017; Chryssakis et al. 2016; Diesel 2017; Chryssakis and Stahl 2013; Chryssakis and Tvete 2014). Most likely candidates for future inclusion in SECAs are Bosporus Straits and Sea of Marmara, Hong Kong and parts of the coastline of Guangdong in China (Chryssakis et al. 2016; Diesel 2017; Chryssakis and Stahl 2013; Chryssakis and Tvete 2014). As evidenced by Fig. 7.1a, ships operating in SECAs are required to use low-sulfur fuel (0.1% S) or, in case this is not possible, to use after-treatment systems such as SO_x scrubbers. It is worth to mention that the European Union (EU) has imposed the limit of 0.1% fuel sulfur content in ports and inland waterways (Chryssakis et al. 2016; Diesel 2017; Chryssakis and Stahl 2013; Chryssakis and Tvete 2014). Also, EU will probably mandate a limit of 0.5% fuel sulfur cap in EU waters irrespective of IMO potential skepticism (Chryssakis et al. 2015, 2016).

Another category of gaseous pollutants emitted from marine engines are oxides of nitrogen (NO_x). Nowadays, shipping-emitted NO_x is of high concern by IMO, ship owners, classes, and engine manufacturers since their control either inside or/both outside marine engines is not directly facilitated such as in the case of SO_x emissions through fuel sulfur content. Figure 7.1b illustrates older and current IMO NO_x limits as function of engine speed "n" in RPM (Dieselnet 2017; Cleantech 2017; International Maritime Organization 2017). As evidenced by Fig. 7.1b, currently NO_x emission limits designated by standard Tier II are enforced globally, whereas the stricter standard Tier III is legislated by IMO in NO_x Emission Control Areas (NECAs) (Dieselnet 2017; International Maritime Organization 2017; Chryssakis et al. 2016). Currently, NECAs include North American ECA including



(a) IMO fuel sulfur content limits

(b) IMO NOx limits

Fig. 7.1 **a** IMO MARPOL Annex VI fuel sulfur content limits (International Maritime Organization 2017), **b** variation of IMO MARPOL Annex VI NO_x limits from marine engines as function of rated engine speed. It is shown the initial NO_x limits (Tier I) and the NO_x limits that are currently issued globally (Tier II) and in NO_x Emission Control Areas (NECAs) (Tier III) (Dieselnet 2017; Cleantech 2017; International Maritime Organization 2017)

most of USA and Canadian coast and US Caribbean ECA, including Puerto Rico and the US Virgin Islands. According to Fig. 7.1b, the most stringent NO_x emission standard (Tier III) requires an approximate average reduction of shipping-emitted NO_x of 80% compared to initial NO_x emission standard (Tier I).

According to the Chap. 4 of IMO MARPOL Annex VI, there are two mandatory mechanisms intended to ensure an energy efficiency standard for ships: (A) the EEDI for new ships and (B) the Ship Energy Efficiency Management Plan (SEEMP) and the corresponding Energy Efficiency Operational Index (EEOI) for all ships (Dieselnet 2017; International Maritime Organization 2017). The EEDI is a performance-based mechanism that requires specified minimum energy efficiency in new ships. Ship designers and builders are free to choose the technologies to satisfy the EEDI requirements in a specific ship design (Dieselnet 2017; International Maritime Organization 2017). The SEEMP establishes a mechanism for operators to improve the energy efficiency of ships. Aforementioned regulations apply to all ships of and above 400 gross tonnages and enter into force from January 1, 2013.

As can be concluded from the aforementioned analysis, there is a very strong interest in shipping community for the actuation of immediate and drastic measures aiming to the significant curtailment of air pollutants emitted from ships and to their compliance with IMO regulations both in emission control areas and outside of them. Toward this aim, a direct mean for downplaying IMO regulated SO_x, NO_x and GHG emissions is the combustion of natural gas in marine main slow-speed two-stroke compression-ignition engines, in high-speed and medium-speed main and auxiliary dual-fuel CI engines and in marine main and auxiliary high-speed spark-ignition (SI) gas engines (Chryssakis et al. 2016; Diesel 2017; Chryssakis and Stahl 2013; Chryssakis and Tvete 2014; Chryssakis et al. 2015; McGill et al. 2013; Germanischer lloyd 2013; Trauthwein 2012).

Theoretical and experimental studies conducted in the past (Murakami and Baufeld 2013; Mohr and Baufeld 2013; Tozzi et al. 2016; Moriyoshi et al. 2016; Andre 2013; Hiltner et al. 2016; Hiltner 2013; Callahan and Hoag 2013; Brynolf et al. 2014; Li et al. 2015; Wei and Geng 2016; Roecker 2016) in marine two-stroke dual-fuel and marine four-stroke dual-fuel and SI gas engines have demonstrated that natural combustion can:

- Practically eliminate SO_x emissions
- Dramatically downplay NO_x emissions allowing gas engines to directly comply with IMO Tier III regulations
- Significantly reduce CO_2 emissions
- Generate extremely low PM emissions and in the cases of very small pilot diesel quantity and of SI gas engines to attain smokeless engine operation.

However, as reported in the literature (Schlick 2014; Levander 2011; Haraldson 2011; Diesel and Turbo 2017; Wartsila 2-stroke dual fuel technology 2014; Zannis et al. 2017; Yfantis et al. 2017; Kjemtrup 2015; Ott 2015; Hagedorn 2014), natural gas combustion in marine dual-fuel and gas engines is accompanied by the following environmental and operational obstacles:

- Major deterioration of total unburned hydrocarbon (THC), CH_4, and CO emissions compared mainly to conventional CI engines. Especially in the case of CH_4 emissions (phenomenon of "methane slip"), dual-fuel and gas engines may provoke serious amendments of IMO GHG emissions regulations policy if not only CO_2 but also methane is included in future shipping GHG picture (Murakami and Baufeld 2013; Mohr and Baufeld 2013; Hiltner et al. 2016; Brynolf et al. 2014; Schlick 2014; Levander 2011).
- Increasing risk for either knocking or flame quenching in both dual-fuel and gas engines and for misfiring mainly in dual-fuel engines. Potential avoidance of one or more of the aforementioned gas combustion-related phenomena may require the combined incorporation of various combustion chamber design technologies and in-cylinder measures (Murakami and Baufeld 2013; Mohr and Baufeld 2013; Tozzi et al. 2016; Moriyoshi et al. 2016; Schlick 2014; Levander 2011).
- High capital cost and technical complexity for retrofitting existing marine diesel engines in order to be able to operate effectively with natural gas (Murakami and Baufeld 2013; Mohr and Baufeld 2013; Tozzi et al. 2016; Moriyoshi et al. 2016; Schlick 2014; Levander 2011).

Though that in the past there have been reported extensive and elaborative studies in the literature (Murakami and Baufeld 2013; Mohr and Baufeld 2013; Andre 2013; Brynolf et al. 2014; Li et al. 2015; Wei and Geng 2016; Schlick 2014; Levander 2011), which examined thoroughly contemporary natural gas combustion technologies both in marine CI and SI engines (Murakami and Baufeld 2013; Mohr and Baufeld 2013; Andre 2013; Brynolf et al. 2014; Li et al. 2015; Schlick 2014; Levander 2011), and in other engine type (Wei and Geng 2016), there is a lack of consolidated information regarding the technological, environmental, and economic evaluation of modern natural gas combustion technologies in main and auxiliary marine engines. Hence, the main purpose of the present study is to cover the aforementioned gap in the literature by examining thoroughly contemporary natural gas combustion technologies used in marine main and auxiliary engines on a technical, operational, environmental, and economic basis.

7.2 Natural Gas as Maritime Fuel

Before examining the effects of natural gas combustion on marine engines performance characteristics and pollutant emissions, it is essential to understand the main characteristics and the key advantages and disadvantages of natural gas as a maritime fuel. Hence, the advantages of the use of natural gas in marine internal combustion engines are the following (Chryssakis et al. 2016; Diesel 2017; Chryssakis and Stahl 2013; Chryssakis and Tvete 2014; Chryssakis et al. 2015; McGill et al. 2013; Germanischer lloyd 2013; Murakami and Baufeld 2013; Mohr and Baufeld 2013):

- Natural gas indicates high availability from ground reserves: The recent discovery of huge available underwater natural gas reserves in the coasts of Eastern Africa and in the Caspian Sea, for example, guarantees the unobstructed supply of the global markets for many years in the future (McGill et al. 2013).
- The competitive prices of natural gas compared to residual fuel and to distillate fuel: Natural gas is expected to have competitive cost compared to residual fuel and to distillate oil until 2035 (see Fig. 7.2). Nowadays, natural gas indicates almost 70% lower price compared to residual fuel and 85% lower price compared to distillate fuel oil (McGill et al. 2013).

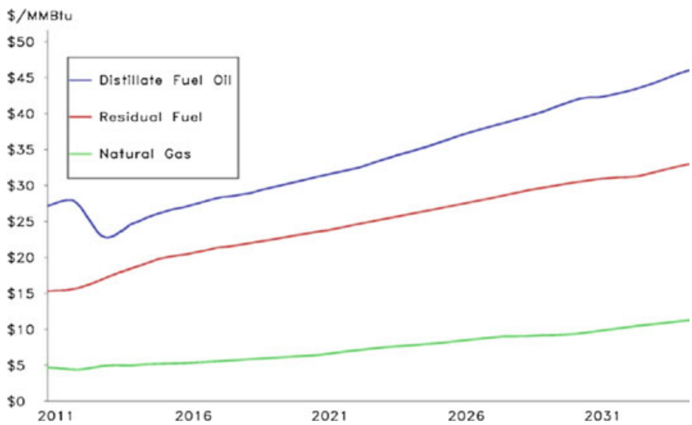


Fig. 7.2 Chronological evolution of the prices of natural gas, residual fuel, and distillate fuel oil (McGill et al. 2013)

- Natural gas combustion can lead to approximately 20% lower CO_2 emissions compared to diesel oil combustion (Schlick 2014) (see Fig. 7.3). Even higher CO_2 reductions have been reported reaching up to 30% when burning natural gas in four-stroke lean-burn marine engines (Levander 2011). The aforementioned CO_2 reductions can be ascribed to the significantly lower carbon to hydrogen mass ratio of natural gas compared to diesel oil (Levander 2011; Haraldson 2011).
- Dual-fuel and natural gas engines can attain 80% lower NO_x emissions compared to conventional diesel engines (see Fig. 7.3). Higher NO_x reductions up to 85% compared to conventional diesel combustion can be accomplished in lean-burn gas engines, which are operating with high air–fuel ratio (Levander 2011). It has experimentally been proved that dual-fuel and spark-ignited gas engines with pre-chamber can directly comply with IMO Tier III NO_x limits without the need for specially designed in-cylinder ("internal") or after-treatment ("external") measures (Chryssakis et al. 2015).
- Natural gas combustion in marine main and auxiliary engines can practically eliminate SO_x emissions since sulfur is removed from fuel when is liquefied (see Fig. 7.3) (Chryssakis et al. 2015; Mohr and Baufeld 2013; Levander 2011).
- The use of natural gas in marine four-stroke engines can result in extremely low PM emissions especially in the case of lean-burn gas engines (Chryssakis et al. 2015). This can be attributed to lower molecular complexity of the natural gas mixture of burning gases, which generate dramatically lower polyaromatic hydrocarbons (PAHs), which are considered as primary particulate matter

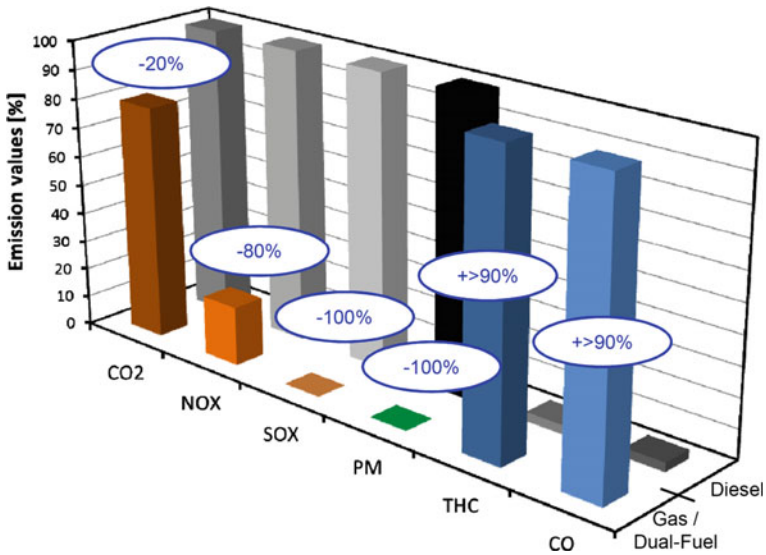


Fig. 7.3 Comparative effect of natural gas combustion in dual-fuel and gas engines on CO_2, NO_x, SO_x, PM, THC, and CO emissions relative to diesel combustion (Schlick 2014)

precursors, compared to conventional diesel combustion. PM emissions during dual-fuel combustion are primarily controlled by pilot diesel quantity. It is worth to mention that, according to detailed experimental studies (Chryssakis et al. 2015; Murakami and Baufeld 2013; Mohr and Baufeld 2013; Levander 2011; Haraldson 2011), natural gas combustion either under dual-fuel mode or under spark-ignited mode results in no visible smoke ("smokeless operation").

- Finally, another virtue of natural gas combustion in marine engines is that it does not leave sludge deposits (Chryssakis et al. 2015; Mohr and Baufeld 2013; Levander 2011).
- It has been established specific rules for the construction of natural gas carriers from various classes such as DNV/GL and Lloyd's Register (McGill et al. 2013).
- There is high availability of marine internal combustion engines operating with natural gas from many different engine manufacturers: Nowadays, there are commercially available various editions of two-stroke and four-stroke marine compression-ignition engines operating with natural gas (dual-fuel diesel/natural gas engines) and four-stroke spark-ignition marine engines operating with natural gas (McGill et al. 2013). Dual-fuel marine compression-ignition engines can be used as main engines (two-stroke and four-stroke) or as electric generators (four-stroke engines), whereas four-stroke gas spark-ignition engines can be used in passenger ships and in ferries but also as electric generators (McGill et al. 2013).
- Finally, natural gas application in shipping is also facilitated by the technical and operational experience, which has been accumulated over recent decades by ship owners, ships' technical staff, classes, shipyards, and engine manufacturers (McGill et al. 2013).

The main disadvantages of the use of natural gas in ships are the following (McGill et al. 2013):

- Natural gas combustion in marine internal combustion engines results in higher emissions of unburned hydrocarbons and of gaseous methane compared to liquid fuel combustion. Specifically, natural gas combustion in either dual-fuel CI engines or in gas SI engines results in the deterioration of unburned hydrocarbon emissions from 70 to 90% compared to conventional engine operation (McGill et al. 2013). In addition, natural gas combustion in marine internal combustion engines results in a serious worsening of methane emissions (methane slip) compared to conventional operation. If we take into account that methane is considered as 20–25 times more detrimental global warming gas compared to carbon dioxide, then natural gas in marine engines provokes great skepticism regarding the phenomenon of methane slip. For this reason, it is expected that methane emissions will be incorporated in the future in the calculation of global warming gases and on that basis is expected the deterioration of the operational cost in ships using natural gas as marine fuel from the potential implementation of carbon tax in maritime sector (McGill et al. 2013).

- Natural gas combustion is not directly compatible with existing marine compression-ignition and spark-ignition engines and with existing marine engines' fuel supply infrastructure: Specifically, the use of natural gas as fuel in ships requires constructional modifications of existing marine internal combustion engines and requires also modifications in the fuel supply networks of main and auxiliary engines (McGill et al. 2013). Major manufacturers of marine four-stroke diesel engines have proposed certain modifications in order their engines to operate with natural gas as dual-fuel engines (Schlick 2014). Specifically, major marine CI engine manufacturers propose the reduction of compression ratio for avoiding excess in-cylinder pressures during dual-fuel operation. It is also proposed the reformulation of piston bowl design for improved fuel-air mixing and the replacement of liner honing in order to be compatible with higher in-cylinder temperatures during dual-fuel operation compared to conventional diesel operation (Schlick 2014). Regarding the cylinder head, it is proposed the installation of individual gas admission valves for each cylinder head, the installation of a pilot fuel injector, the optimization of intake process probably through increased swirl ratio for increasing in-cylinder turbulence levels and improving, thus, fuel–air mixing process. It is also proposed the modification of valve seat geometry for avoiding excessive gas leakages and the installation of a knock sensor in each cylinder for monitoring and processing in-cylinder pressure and thus avoiding pre-ignition or post-combustion knocking phenomena (Schlick 2014). In addition to engine modifications, fuel supply network modifications are required for security reasons also regarding onboard natural gas management. Of special interest are the results of a detailed study, which was performed in 2013 (McGill et al. 2013). In this study it was calculated the retrofit cost of main and auxiliary engines of three different ships in order they have the capacity to operate with natural gas as fuel. It was also calculated the pertinent retrofit cost of fuel supply networks of the main and auxiliary engines for being compatible with natural gas. The main findings of this study are summarized in Table 7.1 (McGill et al. 2013).

Table 7.1 Retrofit costs of certain ship types for operation of their engines with natural gas (McGill et al. 2013)

Ship type	Size (tons)	Engine	Engine retrofit cost (million $)	Engine fuel supply system retrofit cost (million $)	Total engine retrofit cost (million $)
Tug	150	2 × 1500 HP	1.2	6.0	7.2
Ferry	1000	2 × 3000 HP	1.8	9.0	10.8
Great lakes bulk carrier	19,000	2 × 5000 HP	4.0	20	24

- The use of natural gas as marine fuel in new buildings introduces a construction premium compared to newly constructed ships burning liquid fossil fuels in their engines. Specifically, the building cost of a gas-powered ship is considerably higher than the pertinent cost of a fossil fuel-powered ship since there is a cost increase for gas engines and for gaseous fuel system and also for LNG storage tanks (McGill et al. 2013). According to a detailed study performed by Germanischer Lloyd (Germanischer lloyd 2013), natural gas utilization as marine fuel introduces an additional investment cost of 25% over that of the cost for constructing a typical new container ship. Also according to a recent DNV study (Schlick 2014) if a ship spends more than 30% of its operating time in ECAs the cost of gas-fueled engines can be justified (McGill et al. 2013; Zannis et al. 2017; Yfantis et al. 2017) drawback of natural gas as maritime fuel is the increased onboard space requirements for natural gas storage (McGill et al. 2013; Zannis et al. 2017; Yfantis et al. 2017). Specifically, increased onboard space requirements are necessary for the storage of the required natural gas volume for ship autonomy equal to of a conventional ship with onboard storage of liquid fuels (McGill et al. 2013; Zannis et al. 2017; Yfantis et al. 2017). A specific natural gas weight stored as liquefied natural gas (LNG) requires almost only 40% of the corresponding natural gas weight stored as compressed natural gas (CNG) at 3600 psi (≈250 bar). Hence, LNG requires much less storage space compared to CNG, and for this reason, natural gas carriage as LNG is the preferred onboard storage and carriage technique. When natural gas is stored as LNG requires two times bigger space compared to the corresponding storage space of a liquid fuel whereas when natural gas is stored on board as CNG requires five times bigger space compared to the corresponding storage space of a liquid fuel (McGill et al. 2013; Zannis et al. 2017; Yfantis et al. 2017). Table 7.2 shows comparative examples of the minimum storage capacity and of the onboard storage volume for the carriage of residual fuel, distillate fuel, LNG, and CNG with three different ship types. In Fig. 7.3 is given the comparison of the fuel weight per equivalent gallon of distillate fuel for distillate fuel, for residual fuel, for LNG, and for CNG@3600 psi. In Fig. 7.4 is also depicted a comparison of the storage volume per equivalent gallon distillate fuel for distillate fuel, residual fuel, LNG, and CNG@3600 psi (McGill et al. 2013; Zannis et al. 2017; Yfantis et al. 2017).
- The increased ship supply time with natural gas is another serious drawback of natural gas as marine fuel. Specifically, the required supply time of a ship with natural gas is higher compared to the pertinent time for ship supply with a liquid fuel. In particular regarding the onboard storage of natural gas as CNG, it can be stated that this solution cannot be considered as more or less viable due to (McGill et al. 2013; Sarigianidis 2016):

– Higher ship supply times compared to ship supply time with LNG.
– Additional onboard space requirements for the CNG storage tanks.
– Limited carriage volume, which leads to limited CNG ship autonomy.

Table 7.2 Comparative examples of the minimum onboard storage capacity and of the onboard storage volume for the transportation of distillate or heavy fuel oil, LNG, and CNG with three different ship types (McGill et al. 2013)

				Typical minimum onboard storage capacity		Onboard storage volume		
Ship	Fuel type	HP	Daily fuel consumption (gal)	(days)	(gal)	Distillate oil (ft^3)	LNG (ft^3)	CNG (ft^3)
Towing tug	Distillate oil	3000	1417	14	20,000	2674	4830	12,178
100-car ferry	Distillate oil	6000	2268	7	16,000	2139	3864	9742
Great lakes ore carrier	Heavy fuel oil	10,000	6934	21	145,000	19,385	38,183	92,264

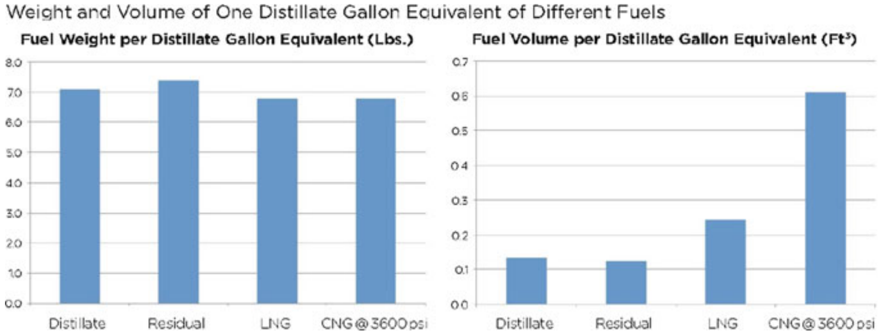


Fig. 7.4 (Left figure) Comparison of fuel weight per equivalent gallon of distillate fuel oil for distillate fuel oil, for residual fuel, for LNG, and for CNG@3600psi. (Right figure) Comparison of fuel volume per distillate gallon equivalent for distillate fuel, for residual fuel, for LNG, and for CNG@3600 psi (McGill et al. 2013)

For the aforementioned reasons, CNG is considered as viable solution for ships doing limited-distance trips and they also have sufficient return time for ship resupply with CNG. On the other hand, the main drawbacks of the LNG are the following (McGill et al. 2013; Sarigianidis 2016):

- The liquefaction of natural gas requires the deep freezing of the gas at a temperature almost equal to −160 °C. Nowadays, this deep freezing is attainable through technically complicated and expensive shored installations.
- LNG carriers should be equipped with technical systems of deep freezing for keeping LNG at temperatures close to −160 °C.

- The additional onboard safety requirements of natural gas carriage ship is another obstacle to the broad use of natural gas in maritime sector. Specifically, natural gas carriage from a ship requires additional installations and safety settings, and it results in onboard constructional interventions, which lead to a higher overall constructional cost of a natural gas carriage ship compared to a conventional one (McGill et al. 2013).
- Finally, the last disadvantage of natural gas as marine fuel is the global limited number of shored infrastructures for the bunkering of ships with natural gas. Specifically, an international network of shored LNG bunkering stations should be constructed in order the LNG ship carriage and the use of natural gas as maritime fuel to be attractive for the majority of commercial ships. Today, LNG bunkering from shored LNG stations is more expensive and technically more complicated compared to liquid fuel bunkering. In addition, LNG bunkering can be accomplished by a limited number of corresponding LNG shored stations in the world (McGill et al. 2013).

In the following sections are examined the contemporary natural gas combustion technologies in marine two-stroke dual-fuel engines and in marine four-stroke dual-fuel and gas SI engines. It is also evaluated the performance of the aforementioned gas engines from technological, operational, environmental, and economic standpoints.

7.3 Dual-Fuel and Gas Spark-Ignition Engines: Contemporary Combustion Technologies

7.3.1 Two-Stroke Dual-Fuel Engines Technologies

Nowadays, there are commercially available two main types of marine main two-stroke CI engines burning natural gas: The one operating according to diesel cycle and the one operating according to Otto cycle (Wartsila 2-stroke dual fuel technology 2014; Zannis et al. 2017; Yfantis et al. 2017; Kjemtrup 2015; Ott 2015). The operational principles of the two aforementioned types of two-stroke natural gas engines are shown in Fig. 7.5a and b. Regarding the "Diesel-cycle" two-stroke dual-fuel engine, it should be mentioned that the specific engine type operates according to diesel-cycle principle, which means that natural gas–diesel combustion is primarily controlled by diffusion-controlled combustion as in conventional diesel engines (Wartsila 2-stroke dual fuel technology 2014; Kjemtrup 2015; Ott 2015). Specifically, in "diesel-cycle" dual-fuel engine, a pilot diesel oil quantity of about 3%m/m is injected before natural gas injection both immediately before top dead center (TDC). The compression heat creates in-cylinder temperatures high enough to auto-ignite liquid fuel oil. After diesel oil ignition, a flame front is created inside cylinder, which is expanded inside the entire combustion chamber being fed by natural gas injection and combustion (Wartsila 2-stroke dual fuel technology 2014;

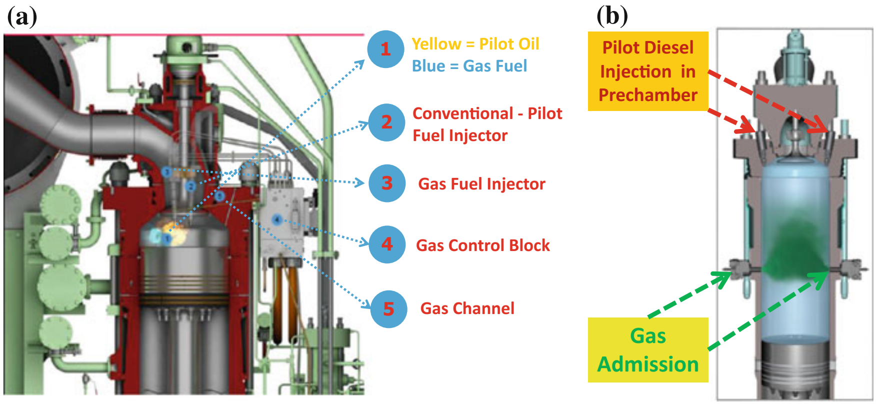


Fig. 7.5 Operational principles of **a** "diesel-cycle" two-stroke dual-fuel engine (Kjemtrup 2015) and **b** "Otto-cycle" two-stroke natural gas engine (Wartsila 2-stroke dual fuel technology 2014)

Kjemtrup 2015; Ott 2015). Natural gas is injected in "diesel-cycle" engine at a pressure of 300barg and at a temperature of 45 °C. "Diesel-cycle" two-stroke gas engine meets SO_x requirements with LNG or low-sulfur diesel oil, and its most important virtue is that indicates a high fuel efficiency close to 50% depending on engine load (Wartsila 2-stroke dual fuel technology 2014; Kjemtrup 2015; Ott 2015). However, the "diesel-cycle" two-stroke gas engine cannot meet directly IMO Tier III NO_x limits, and thus, there is a potential need for exhaust gas recirculation (EGR) or selective catalytic reduction (SCR) (Wartsila 2-stroke dual fuel technology 2014; Kjemtrup 2015; Ott 2015).

On the other hand, the operational principle of the "Otto-cycle" two-stroke natural gas engine is based primarily on the premixed lean-burn natural gas combustion with pilot diesel ignition (Wartsila 2-stroke dual fuel technology 2014; Kjemtrup 2015; Ott 2015). Specifically, natural gas is injected at low pressure (<10 bar) at mid-stroke after scavenging. A pilot fuel oil quantity of 1%m/m is injected before TDC in pre-chamber, and then, it auto-ignites due to compression heat of the premixed air/gas mixture (Wartsila 2-stroke dual fuel technology 2014; Kjemtrup 2015; Ott 2015). The "Otto-cycle" two-stroke natural gas engine indicates lower fuel efficiency compared to the "diesel-cycle" engine, which is close to 47% depending on engine load (Wartsila 2-stroke dual fuel technology 2014; Kjemtrup 2015; Ott 2015). Also, the predominantly premixed-controlled natural gas combustion of the "Otto-cycle" two-stroke gas engine creates serious considerations regarding increased THC and CO emissions. It also creates skepticism regarding a significant deterioration of methane slip compared to "diesel-cycle" engine and also regarding potential undesirable pre-ignition or post-combustion knocking phenomena (Wartsila 2-stroke dual fuel technology 2014; Kjemtrup 2015; Ott 2015). The most important environmental virtues of the "Otto-cycle" two-stroke gas engine is that fulfills ECA SO_x requirements with LNG or low-sulfur fuel oil and also meets directly Tier III NO_x requirements without in-cylinder measures or after-treatment (Wartsila 2-stroke dual fuel technology 2014; Kjemtrup 2015; Ott 2015).

It is of high importance to understand the main difference in engine operational characteristics between the "diesel-cycle" and the "Otto-cycle" dual-fuel marine two-stroke engines, which are responsible for their distinctive variations in engine efficiency and NO_x emissions. For this reason, in Fig. 7.6 are shown indicative predictions of mean bulk gas in-cylinder temperature obtained from the literature (Wartsila 2-stroke dual fuel technology 2014) for the "diesel-cycle" and the "Otto-cycle" dual-fuel two-stroke marine CI engines. The black color curve corresponds to the in-cylinder temperature of the "Diesel-cycle" engine, whereas the orange color curve corresponds to the cylinder temperature of the "Otto-cycle" engine. As observed from Fig. 7.6, the "Otto-cycle" indicates an earlier initiation of cylinder temperature rise during the early stages of combustion, which can be ascribed to the more homogeneous nature of natural gas combustion in this type of two-stroke marine engine. However, the "diesel-cycle" two-stroke engine indicates significantly higher peak cylinder temperatures compared to the corresponding ones of the "Otto-cycle" engine. Specifically, the "diesel-cycle" two-stroke engine demonstrates 300–500 °C higher maximum flame temperatures compared to the "Otto-cycle" engine. Also during expansion stroke, the "diesel-cycle" engine indicates higher cylinder temperatures compared to the "Otto-cycle" engine, which can be attributed to the diffusion-controlled of the fuel–air mixture, which is the dominating combustion mode in the "diesel-cycle" engine. The higher in-cylinder

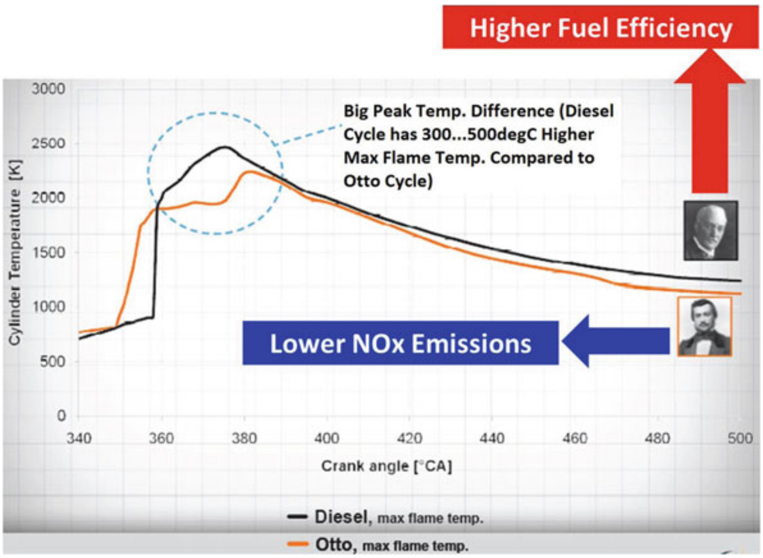


Fig. 7.6 Comparison of in-cylinder temperature–crank angle curves between "Diesel-cycle" and "Otto-cycle" two-stroke marine engines (Ott 2015)

temperatures of the “diesel-cycle” engine, which are evidenced during combustion and expansion stroke, are responsible for the higher fuel efficiency of this type of engine compared to the “Otto-cycle” engine. On the other hand, the lower peak flame temperatures and mainly the lower cylinder temperatures during expansion stroke, which are observed in the case of the “Otto-cycle” engine, are primarily responsible for the lower NO_x emissions of the “Otto-cycle” engine compared to the ones of the “diesel-cycle” engine.

Having examined the main constructional and operational differences between the “diesel-cycle” and the “Otto-cycle” dual-fuel two-stroke marine CI engines, a question raised regarding the operational modes of the “diesel-cycle” engine and whether or not the transition from conventional diesel engine operation to dual-fuel operation is smooth and unobstructed in “diesel-cycle” engine. Toward providing answers to these questions, Fig. 7.7 is given, which provides an illustrative overview of the different operational modes of the “diesel-cycle” dual-fuel two-stroke marine CI engine (Kjemtrup 2015). As evidenced by Fig. 7.7, the “diesel-cycle” two-stroke engine can operate under three different operational modes: (a) fuel-only mode, (b) minimum-fuel oil mode, and (c) specified gas mode. Under fuel-only mode, the “diesel-cycle” engine operates as conventional two-stroke diesel engine with heavy or light diesel oil. Under minimum-fuel oil mode, the “diesel-cycle” two-stroke engine uses 97%m/m natural gas with 3%m/m pilot diesel oil for ignition. Under this mode, the “diesel-cycle” two-stroke marine CI engine can operate between 10 and 100% of full engine load (Kjemtrup 2015). Finally, under

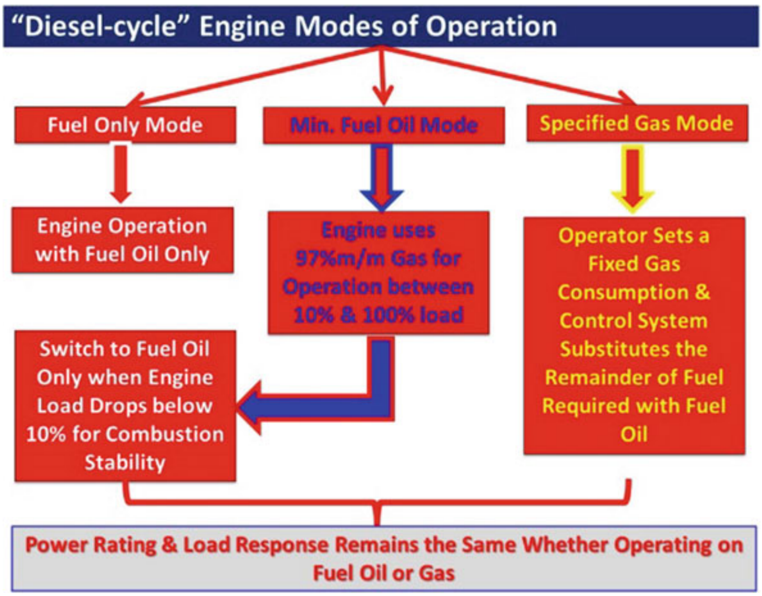


Fig. 7.7 “Diesel-cycle” two-stroke dual-fuel marine engine modes of operation (Zannis et al. 2017; Kjemtrup 2015)

the specified gas mode, the operator of the engine specifies desirable natural gas consumption and the control system of the "diesel-cycle" engine substitutes the remained of the fuel required with heavy or light diesel oil.

Figure 7.8a and b show the variation of fuel percentage with engine load during minimum-fuel mode (Fig. 7.8a) and during specified dual fuel mode (Fig. 7.8b) of the "diesel-cycle" two-stroke marine CI engine. In both Fig. 7.8a and b with red color is showed the total fuel consumption and with green color the pilot fuel consumption. According to red area of Fig. 7.8a, the "diesel-cycle" engine under minimum-fuel mode can operate smoothly from 10 to 100% of full load (SMCR) with 97%m/m natural gas and with only 3%m/m diesel oil. As observed from Fig. 7.8b, the "diesel-cycle" two-stroke marine CI engine can also smoothly operate from 10 to 100% of full load (SMCR) with different analogies of predefined natural gas consumption and pilot diesel oil quantities. According to the literature (Kjemtrup 2015) under specified dual-mode operation, the "diesel-cycle" two-stroke dual-fuel CI marine engine experienced no fuel slip, no knocking problems and its operation was insensitive to natural gas methane number (i.e., natural gas quality). Also during specified dual-mode operation, the specific engine experienced unchanged load response, which is an important virtue of this type of engine. For this reason, the "diesel-cycle" engine managed to reduce significantly its gas mode operational load up to 10%, which is of significant importance for sustaining its high efficiency and operational smoothness under significantly low engine loads. The most important conclusion regarding "diesel-cycle" two-stroke dual-fuel CI engine is that power rating and load response remain the same whether the engine operates on diesel oil or on gas, which is extremely important for the power delivery to ship propellers.

7.3.2 Four-Stroke Dual-Fuel and Gas Spark-Ignition Engines Technologies

After examining the two types of commercially available two-stroke natural gas engines, it is of particular importance to examine the contemporary natural gas combustion technologies currently used in four-stroke dual-fuel engines and in gas SI engines, which are summarized in Fig. 7.9 (Murakami and Baufeld 2013; Mohr and Baufeld 2013). The four-stroke natural gas technologies are divided into main categories according to the constructional design of the combustion chamber and their operational principle: the open-chamber four-stroke dual-fuel and gas SI engines and the corresponding natural gas engines equipped with a divided chamber (i.e., pre-chamber and main combustion chamber) (Murakami and Baufeld 2013). The open-chamber natural gas engines are divided into two categories: The open-chamber dual-fuel engines with micro-pilot injection (OCMP) and the open-chamber spark-ignition (OCSI) gas engines (Murakami and Baufeld 2013; Mohr and Baufeld 2013). Similarly, the natural gas engines with divided chamber are divided into two categories: the pre-chamber with micro-pilot (PCMP) engines

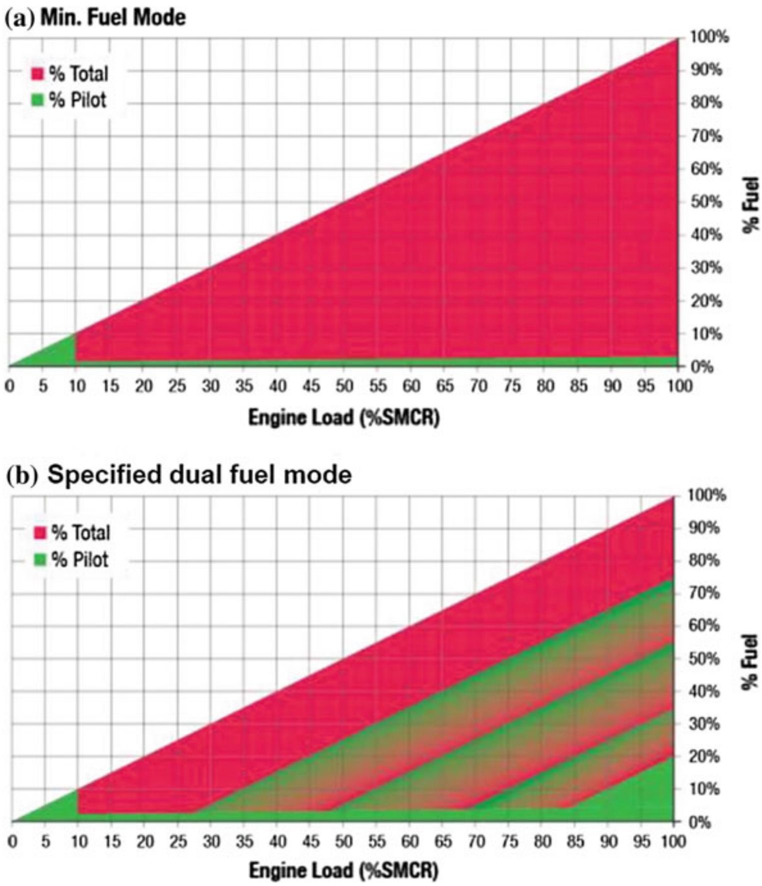


Fig. 7.8 Variations of fuel percentage versus engine load (%SMCR) during **a** minimum-fuel mode and **b** specified dual-fuel mode of the “Diesel-cycle” dual-fuel two-stroke CI marine engine (Kjemtrup 2015)

and the pre-chamber spark-ignition (PCSI) engines (Murakami and Baufeld 2013; Mohr and Baufeld 2013; Zannis et al. 2017; Yfantis et al. 2017).

The operation principle of OCSI engines is based on the induction of a natural gas/air mixture inside the cylinder through the intake valves, which is ignited through a spark plug. The key optimization factor of OCSI engines is the piston bowl design and, specifically, the attainment of high swirl ratio and squish flow for achieving of high turbulence, which has a direct and immediate positive impact on high flame velocity (Murakami and Baufeld 2013; Mohr and Baufeld 2013; Zannis et al. 2017; Yfantis et al. 2017). However, the application of this combustion technology in large bore engines is challenging due to the increased flame travel distance, which curtails brake thermal efficiency (BTE) and worsens knocking

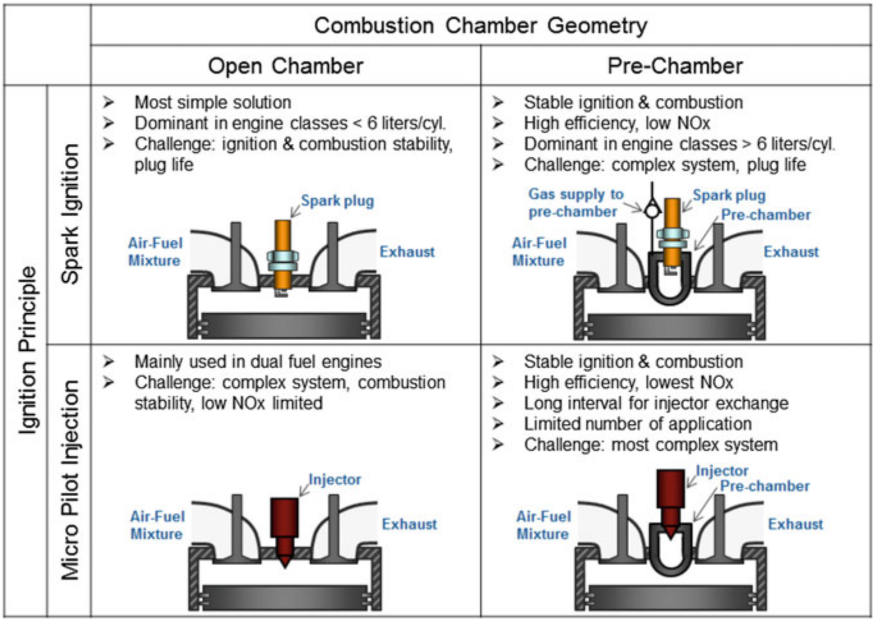


Fig. 7.9 Four-stroke marine dual-fuel CI and natural gas SI engines combustion technologies (Murakami and Baufeld 2013; Mohr and Baufeld 2013)

resistance (Murakami and Baufeld 2013; Mohr and Baufeld 2013; Zannis et al. 2017; Yfantis et al. 2017).

Regarding OCMP operational principle, this is based also in the induction of a natural gas/air mixture through intake valves, which is burned after injection and ignition of micro-pilot fuel oil quantity (Murakami and Baufeld 2013; Mohr and Baufeld 2013; Zannis et al. 2017; Yfantis et al. 2017). In this case, it is observed a slightly faster combustion compared to the OCSI concept due to stronger ignition source (i.e., micro-pilot fuel injection) (Murakami and Baufeld 2013; Mohr and Baufeld 2013). OCMP combustion concept is mainly applied for gas mode operation of four-stroke dual-fuel engines (Murakami and Baufeld 2013; Mohr and Baufeld 2013).

In the PCSI combustion concept, air/natural gas mixture is pushed into the pre-chamber during compression stroke with additional gas directly supplied to the pre-chamber (Murakami and Baufeld 2013; Mohr and Baufeld 2013). This provides a rich mixture close to stoichiometric, which ensures strong and stable ignition (Murakami and Baufeld 2013; Mohr and Baufeld 2013). Hence, this technology is capable of combusting very lean gas mixtures improving, thus, BTE/NO_x trade-off (Murakami and Baufeld 2013; Mohr and Baufeld 2013). According to a large engine manufacturer of four-stroke dual-fuel and gas SI engines (Murakami and Baufeld 2013; Mohr and Baufeld 2013), the key optimization factors in this type of natural gas combustion technology is the pre-chamber geometry, the mixture formation in the pre-chamber for ensuring stable combustion, and the piston bowl optimization for free flame propagation (Murakami and Baufeld 2013; Mohr and Baufeld 2013). The technical challenges of PCSI concept are the increased system complexity and the serious effort required for optimized combustion. It is noteworthy that the application of PCSI concept in smaller engines is limited due to limited installation space for pre-chamber and due to increased engine development cost (Murakami and Baufeld 2013; Mohr and Baufeld 2013).

According to PCMP concept, the liquid fuel pilot injector replaces spark plug in the pre-chamber. This combustion concept offers improved ignition stability at low NO_x levels since liquid fuel stability is not seriously affected by the high air/fuel ratio in the pre-chamber (Murakami and Baufeld 2013; Mohr and Baufeld 2013). This is an advantage of PCMP concept compared to PCSI concept which requires lean air/gas mixtures in the pre-chamber for attaining low NO_x levels worsening, thus, ignition stability (Murakami and Baufeld 2013; Mohr and Baufeld 2013). Another advantage of PCMP compared to PCSI concept is that pilot fuel injector requires longer maintenance intervals compared to the ones required for spark plug (Murakami and Baufeld 2013; Mohr and Baufeld 2013).

7.4 Operational, Environmental, and Economic Performance of Dual-Fuel and Gas Spark-Ignition Engines

7.4.1 Two-Stroke Dual-Fuel Engines

This section will be devoted to the examination of the effect of natural gas combustion in the “diesel-cycle” and in the “Otto-cycle” marine two-stroke CI engines performance characteristics and exhaust emissions. Initiating with “diesel-cycle” two-stroke gas engines, an experimental investigation was performed by the manufacturer of this type of engines to evaluate the effect of natural gas combustion on performance characteristics and exhaust emissions (Kjemtrup 2015). Specifically, engine tests were performed in a “diesel-cycle” two-stroke marine CI engine under conventional diesel operation considering directly injected (DI) diesel oil and under specified dual-fuel mode with 70%m/m natural gas injection (GI) and 30%m/m diesel oil at 25, 50, 75, and 100% of full engine load (Kjemtrup 2015). In Fig. 7.10a–d are shown heat release rate profiles as mean values of all cylinders for conventional diesel operation (DI) with black color curves and for specified dual-fuel operation with natural gas injection (GI), which generated from measurements of in-cylinder pressure during the aforementioned engine tests in a “diesel-cycle” two-stroke marine CI engine. Results for heat release rates at given at 25% (Fig. 7.10a), 50% (Fig. 7.10b), 75% (Fig. 7.10c), and 100% (Fig. 7.10d) of full engine load (Kjemtrup 2015). As evidenced by the observation of Fig. 7.10a and b, natural gas combustion under specified dual-mode operation (GI) results in more intense premixed combustion phase leading to higher peak heat release rate values for this mode compared to conventional diesel operation (GI). Especially, in the case of 50% load (Fig. 7.10b), a slightly earlier initiation of combustion is observed in the case of specified dual-mode operation with natural gas (GI). According to Figs. 7.10b and c, the combustion of natural gas in the "diesel-cycle" CI marine engine under specified dual-mode results in more intense premixed combustion, which leads to higher peak heat release rate values compared to conventional diesel engine operation (DI) and to less intense diffusion-controlled compared again to DI engine operation. As evidenced by Fig. 7.10d, there are no substantial differences in ignition point between specified dual-mode operation (GI) and conventional diesel operation (DI) of the examined "diesel-cycle" two-stroke marine CI engine. Only slightly more higher peak heat release rate values and less intense diffusion-controlled combustion phase is observed in the case of GI operation compared to DI operation at 100% of full engine load. Hence, the combustion of fuel mixture of 70%m/m natural gas/30%m/m diesel oil compared to 100% diesel oil consumption under the same engine operating conditions results in the intensification of premixed combustion phase at all engine loads with the effects to be more pronounced at low engine loads. The variations in premixed combustion phase and peak heat release rate values between GI and DI operation can be attributed to variations in the lower heating value of natural gas/diesel oil mixture and to the natural gas quality (i.e., methane number), which probably make the diesel/natural gas mixture to be more "explosive" (i.e., more intense premixed fuel burning phase with higher peak burning rate values) compared to conventional diesel engine operation.

In Fig. 7.11a and b are given experimental results of the specific fuel oil consumption (SFOC) (Fig. 7.11a) and of the specific NO_x emissions (Fig. 7.11b) from the aforementioned experimental investigation conducted in a "diesel-cycle" two-stroke marine CI engine under two different operational modes: (a) conventional diesel engine operation with directly injected fuel (DI mode) and (b) specified dual-fuel mode with 70%m/m natural gas and 30%m/m diesel oil (GI mode). In both Fig. 7.11a and b, measured SFOC and NO_x emissions are presented as functions of engine load. Also, both SFOC and NO_x values are corrected considering ISO conditions and reference peak cylinder pressure. As evidenced by Fig. 7.11a, natural gas combustion under GI mode results in significantly lower values of relative SFOC at 50, 75, and 100% of full engine load compared to conventional DI mode indicating, thus, a considerable improvement of engine efficiency in the case of dual-fuel combustion compared to conventional diesel operation. This efficiency improvement is related to the aforementioned heat release rates comparison between GI and DI modes and more specifically can be ascribed to the intensification of premixed-controlled combustion observed in the case of GI mode. In other words, as evidenced in the case of GI mode, the higher proportion of fuel mixture, which is burned under premixed conditions (i.e., homogeneously), results in the substantial improvement of engine efficiency in the case of GI mode compared to conventional DI mode. As observed from Fig. 7.11b, dual-fuel combustion under GI mode results in lower NO_x emissions at 50, 75, and 100% of full engine load compared to

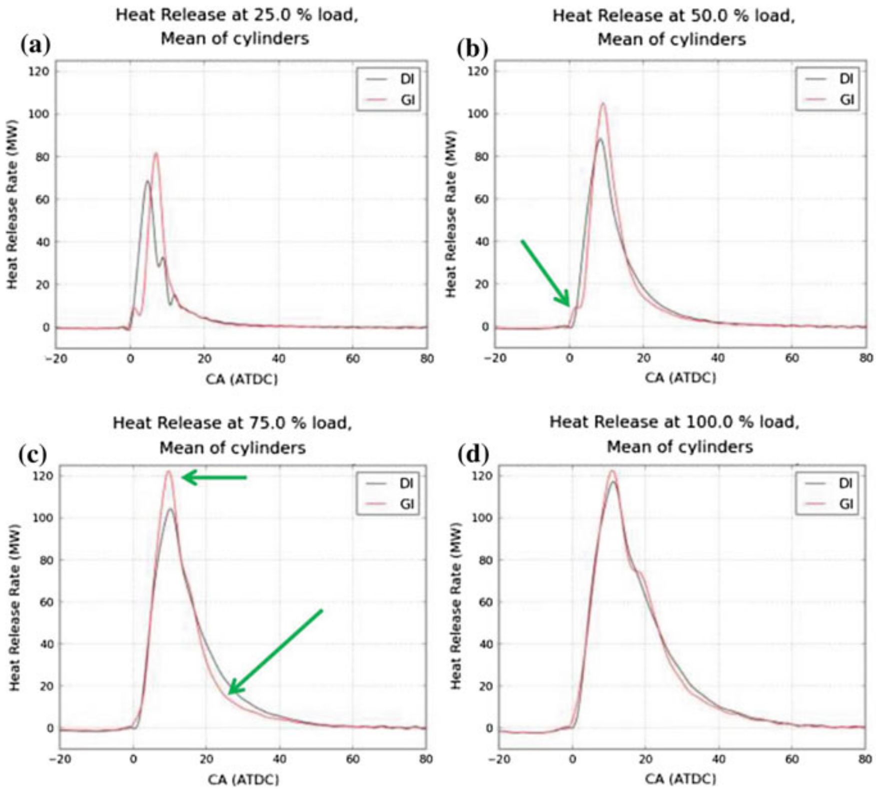


Fig. 7.10 Comparison of heat release rates between conventional diesel operation (DI) and gas injection (GI) of a "diesel-cycle" two-stroke engine at 100% of full load (**a**), 75% of full load (**b**), 50% of full load (**c**), and 25% of full load (**d**). Data were provided from shop tests of a "diesel-cycle" two-stroke engine (Kjemtrup 2015)

conventional DI mode, whereas at 30% load, higher NO_x emissions are evidenced for GI mode compared to conventional DI mode. Also, the lower NO_x emissions observed in the case of GI mode compared to DI mode at 50, 75, and 100% load are directly related to the less intense diffusion-controlled combustion phase witnessed in the case of GI mode compared to DI mode. The less intense diffusion-controlled combustion phase during late combustion and expansion stroke resulted in lower in-cylinder temperatures in the case of GI mode compared to DI mode. According to the well-known Zeldovich thermal NO_x formation mechanism, in-cylinder temperature reduction during late combustion and expansion stroke resulted in lower specific NO_x emissions in the case of GI mode compared to DI mode. It is noteworthy to mention that despite the reduction of specific NO_x emissions, which is observed in the case of GI mode compared to DI mode, the measured NO_x values remained higher than the corresponding values dictated by IMO Tier III NO_x emission standards. Hence, the specific type of dual-fuel engines (i.e., "diesel-cycle"

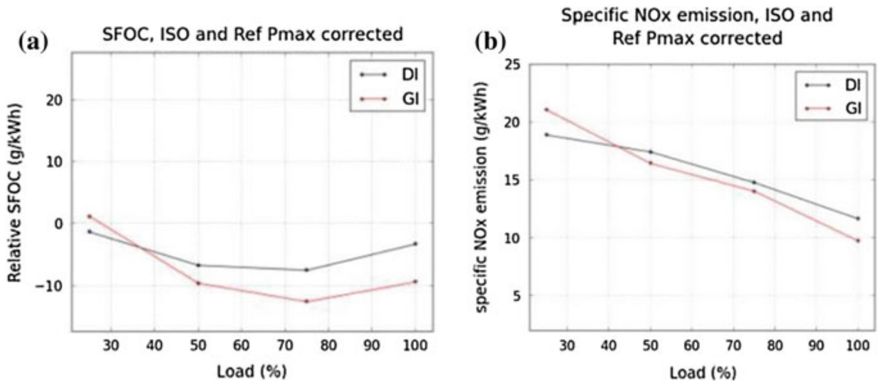


Fig. 7.11 Comparison of relative SFOC (**a**) and specific NO_x emissions (**b**) between conventional diesel operation (DI) and gas injection (GI) operation of a "diesel-cycle" two-stroke engine. Relative SFOC and specific NO_x emissions are given as function of engines load. Data were provided from the shop test of a "diesel-cycle" two-stroke engine (Kjemtrup 2015)

two-stroke dual-fuel marine CI engine) cannot be directly compliant with the stringent IMO NO_x standards without the implementation of either in-cylinder measures (e.g., exhaust gas recirculation—EGR) or exhaust after-treatment measures (e.g., selective catalytic reduction—SCR).

In Fig. 7.12a and b are given experimental results of the specific unburned hydrocarbon (HC) emissions (Fig. 7.12a) and of the specific carbon monoxide (CO) emissions (Fig. 7.12b) from the aforementioned experimental investigation conducted in a "diesel-cycle" two-stroke marine CI engine under two different

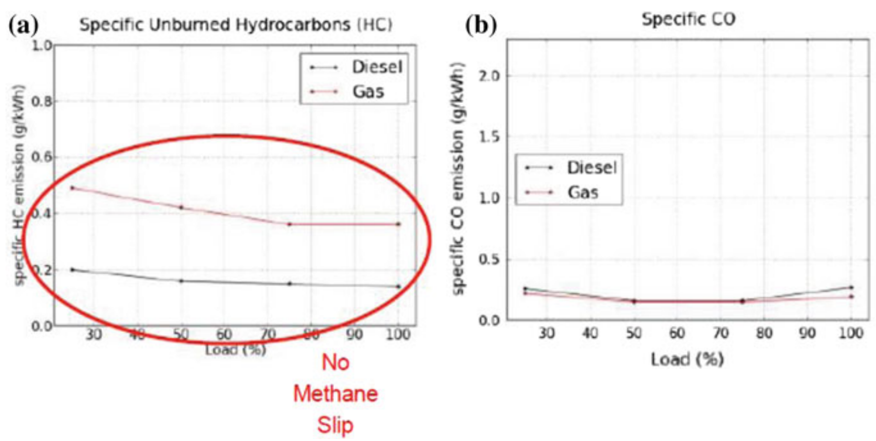


Fig. 7.12 Comparison of specific HC emissions (**a**) and specific CO emissions (**b**) between conventional diesel operation (DI) and gas injection (GI) operation of a "diesel-cycle" two-stroke engine. Relative SFOC and specific NO_x emissions are given as function of engines load. Data were provided from the shop test of a "diesel-cycle" two-stroke engine (Kjemtrup 2015)

operational modes: (a) conventional diesel engine operation with directly injected fuel (DI mode) and (b) specified dual-fuel mode with 70%m/m natural gas and 30% m/m diesel oil (GI mode). In both Fig. 7.12a and b, measured HC and CO emissions are presented as functions of engine load. As evidenced by Fig. 7.12a, the operation of the two-stroke "diesel-cycle" engine under specified dual-fuel mode with gas/diesel 70/30 resulted in the significant increase of HC emissions compared to conventional diesel engine operation (i.e., DI mode) at all loads examined. The deterioration of HC emissions with natural gas/diesel combustion compared to diesel-only combustion can be attributed as already evidenced to the intensification of the premixed combustion phase, which means that higher fuel mixture proportion is burned under homogeneous-like conditions. As expected, shift from diffusion to homogeneous (i.e., premixed) combustion results in deterioration of HC emissions. However, it is quite encouraging that the worsening of HC emissions observed in the case of GI mode compared to DI mode did not accompany by noticeable methane slip. The proven no methane slip observed in "diesel-cycle" engines is an important virtue of this type of dual-fuel two-stroke engines considering that methane emissions are almost 25 times worst greenhouse gas (GHG) compared to carbon dioxide (CO_2). According to Fig. 7.12b, transition from conventional DI engine operating mode to specified dual-fuel mode (i.e., GI mode) did not accompany by substantial variation of measured CO emissions, which is another encouraging environmental finding of the aforementioned experimental investigation.

Consolidating the aforementioned observations from the experimental study performed in a "diesel-cycle" two-stroke marine CI engine, it can be concluded that the transition from conventional two-stroke diesel operation (i.e., DI mode) to specified dual-fuel mode gas/diesel 70/30 operation (i.e., GI mode) resulted in:

- More intense premixed-controlled combustion phase and less intense diffusion-controlled combustion phase, which resulted in lower in-cylinder temperatures during expansion stroke and thus to lower exhaust gas temperature values.
- Lower SFOC values (i.e., higher efficiency values) and smaller NO_x emissions, which, however, remained higher than IMO Tier III limits.
- Higher HC emission values without, however, noticeable methane slip and same CO emission values.

In a second experimental investigation performed in a "diesel-cycle" two-stroke dual-fuel marine CI engine by the corresponding engine manufacturer, engine tests were performed at 25, 50, 75, and 100% of full load considering four gas/diesel (G/D) analogies: 0/100, 30/70, 70/30, and 95/5 (Zannis et al. 2017; Kjemtrup 2015). In Fig. 7.13a and b are presented heat release rate profiles, which were generated from measured cylinder pressure data for all examined gas/diesel analogies (i.e., G/D = 0/100, G/D = 30/70, G/D = 70/30, and G/D = 95/5) at 75% load (Fig. 7.13a) and at 100% of full engine load (Fig. 7.13b). As evidenced by both Fig. 7.13a and b, the increase of gas to diesel proportion results in more intense premised

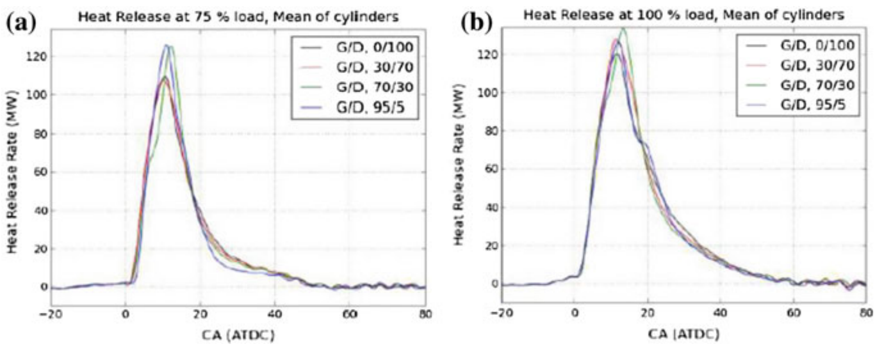


Fig. 7.13 Comparison of heat release rates for gas/diesel (G/D) = 0/100, G/D = 30/70, G/D = 70/30, and G/D = 95/5 of a "diesel-cycle" two-stroke marine engine at 75% of full load (**a**) and at 100% of full load (**b**) (Kjemtrup 2015)

combustion phase and less intense diffusion-controlled combustion phase. Hence, the increase of natural gas quantity to diesel quantity the premixed combustion phase becomes more reactive resulting thus in higher peak heat release rate values (Kjemtrup 2015). This means that more gaseous to liquid fuel mass is physically and chemically prepared during ignition delay period leading, thus, to more reactive premixed combustion phase. The less intense diffusion-controlled combustion phase with increasing gas to diesel analogy as evidenced by Fig. 7.13a and b is expected to result in reduction of in-cylinder temperature during expansion stroke and thus to lower exhaust gas temperature values (Kjemtrup 2015).

In Fig. 7.14a–e are shown experimental results from the aforementioned investigation in a "diesel-cycle" two-stroke marine CI engine for specific NO_x emissions (Fig. 7.14a), specific HC emissions (Fig. 7.14b), and specific CO emissions (Fig. 7.14c), particulate emissions (Fig. 7.14d), and relative SFOC (Fig. 7.14e). At all Fig. 7.14a–e, measured values are given as function of engine load for four different gas/diesel (G/D) analogies, i.e., G/D = 0/100, G/D = 30/70, G/D = 70/30, and G/D = 95/5 (Kjemtrup 2015). According to Fig. 7.14a, the increase of G/D ratio resulted in a substantial reduction of specific NO_x emissions at engine loads higher than 50%. Higher reductions of NO_x emissions are observed in the case of G/D = 70/30. NO_x emission reduction with increasing G/D ratio can be ascribed with the aforementioned less intense diffusion-controlled combustion phase, which as expected resulted in reduction of in-cylinder temperature during expansion stroke (Kjemtrup 2015). An important observation here is that despite significant reduction of NO_x emissions with increasing G/D ratio compared to conventional diesel operation (i.e. G/D = 0/100), the lowest absolute NO_x values remained higher than the corresponding IMO Tier III limits indicating, thus, the necessity for additional measures implementation in order this type of engine to be IMO Tier III compliant (Kjemtrup 2015).

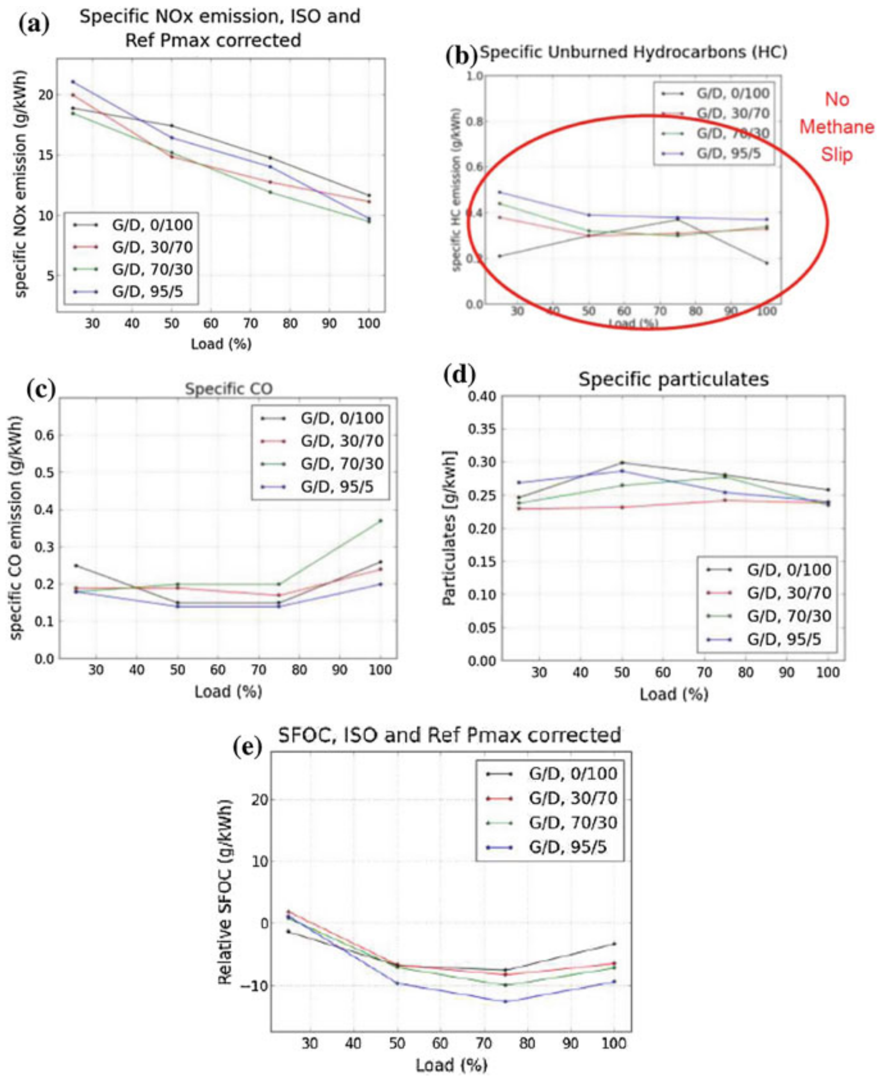


Fig. 7.14 Comparison of specific NO_x–engine load curves (**a**), specific HC emissions–engine load curves (**b**), specific CO emissions–engine load curves (**c**), particulates–engine load curves (**d**), and relative SFOC–engine load curves (**e**) for gas/diesel = 0/100, gas/diesel = 30/70, gas/diesel = 70/30, and gas/diesel = 95/5 of a "diesel-cycle" two-stroke marine engine (Kjemtrup 2015)

As observed from Fig. 7.14b, the increase of G/D ratio resulted in a substantial deterioration of HC emissions compared to conventional diesel operation (i.e., G/D = 0/100) at all engine loads examined. The worsening of HC emissions with increasing gas to diesel proportion is correlated with the intensification of premixed combustion phase under which fuel mixture is burned homogenously. It is noteworthy here to mention that according to the literature (Kjemtrup 2015) the increase of HC emissions with increasing G/D ratio did not accompany by noticeable methane slip, which is a quite encouraging environmental evident for this type of two-stroke dual-fuel marine CI engine (Kjemtrup 2015).

According to Fig. 7.14c, higher values of specific CO emissions are observed in the case of G/D = 30/70 and G/D = 70/30 compared to conventional diesel operation (i.e., G/D = 0/100) at loads higher than 50%. Surprisingly, at the case of G/D = 95/5, there were not observed substantial variations in CO emissions compared to G/D = 0/100. In this case also, CO emission deterioration with increasing G/D ratio compared to conventional diesel operation is related to the intensification of the premixed combustion phase (Kjemtrup 2015).

Observing Fig. 7.14d, lower particulate emission values are witnessed for G/D ratios 30/70, 70/30, and 95/5 compared to G/D = 0/100 at engine loads examined higher than 50%. The lowest particulate values are observed in the case of G/D = 30/70. The reduction of particulate emissions with dual-fuel operation compared to diesel-only operation can be attributed to the less intense diffusion-controlled combustion phase observed with increasing G/D ratios (Kjemtrup 2015).

According to Fig. 7.14e, a clear decrease of relative SFOC (i.e., increase of engine efficiency) is observed in all cases of dual-fuel operation (i.e., G/D = 30/70, 70/30 and 95/5) compared to conventional diesel-only operation (i.e., G/D = 0/100) at engine loads higher than 50%, which is again related to the shift of combustion toward homogeneous premixed combustion (Kjemtrup 2015).

The following similarities were observed during the aforementioned investigation between diesel-only operation (G/D = 0/100) and gas/diesel operation: Same power density, same T/C, scavenging and exhaust gas temperatures, good part load SFOC, good transient engine response, gas/diesel operation was robust to gas quality changes (i.e. MN was irrelevant), simple cylinder lubrication system and use of known lube oil types and finally, regarding safety issues it was not occurred risk of misfiring or knocking and also there was no risk for explosion in scavenge receiver (Kjemtrup 2015).

As already mentioned, gas/diesel operation in “diesel-cycle” two-stroke dual-fuel engine indicated NO_x values lower than diesel-only operation but higher than IMO Tier III limits. For this reason, a second experimental investigation was performed in this type of engine considering the following operational modes: DI = conventional diesel operation, GI = gas/diesel engine operation with EGR, DI + EGR = diesel operation with EGR and GI + EGR = CNG/diesel operation with EGR (Kjemtrup 2015). In Fig. 7.15a and b are shown experimental results from this particular investigation for relative SFOC (Fig. 7.15a) and for specific NO_x emissions (Fig. 7.15b) (Kjemtrup 2015). As evidenced by Fig. 7.15a, there is a clear increase of RSFOC with EGR both for DI and GI operational modes as expected since EGR, as known, reduces in-cylinder temperature (Kjemtrup 2015). According to Fig. 7.15b, the use of EGR in gas/diesel combustion (GI + EGR) results in a substantial reduction of NO_x emissions compared to GI and DI modes

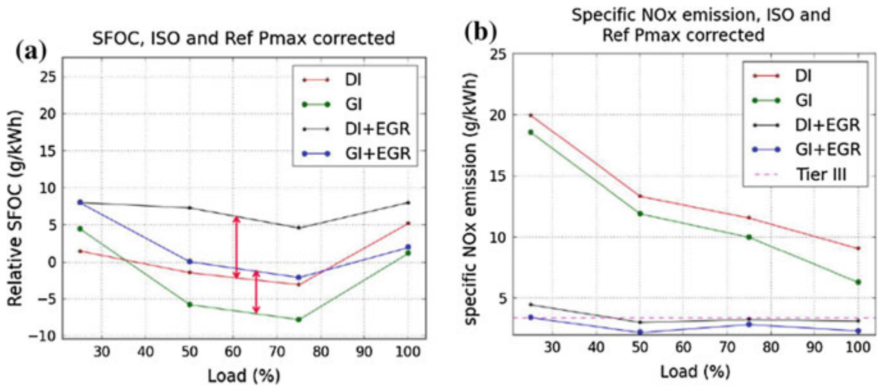


Fig. 7.15 Experimental results for **a** relative SFOC and **b** specific NO_x emissions both as functions of engine load. Results shown in both figures refer to four operational modes: DI (red curves), GI (green curves), DI + EGR (black curves), and GI + EGR (blue curves) (Kjemtrup 2015)

and most importantly leads to engine compliance with IMO Tier III limits at all engine loads examined (Kjemtrup 2015).

Having examined the operational and environmental performance of "diesel-cycle" two-stroke gas/diesel engines, it is essential to investigate the operational performance of "Otto-cycle" two-stroke dual-fuel engines. For this reason, in Fig. 7.16a is shown the effect of natural gas quality in terms of methane number (MN) on the engine power of "Otto-cycle" dual-fuel engines (Ott 2015). As evidenced by Fig. 7.16a, the maximum engine output of this type of engines may be limited by MN if engine power rating point is close to R1-R3 curve (Ott 2015). However, there is no power output limitation from MN if rating point is close to R2-R4 line (Ott 2015). It is noteworthy to mention that MN of LNG is typically between 70 and 90 and natural gases with MN lower than 70 can be burned in "Otto-cycle" dual-fuel engines by reducing engine power output (derating) (Ott 2015). However, it should be kept in mind that the operating area for low-speed two-stroke engines is typically <85% of maximum continuous rating (MCR) (Ott 2015). In Fig. 7.16b are shown two power layout diagrams: one with black line covering an orange color area, which corresponds to conventional two-stroke diesel operation and one with green color curves, which corresponds to "Otto-cycle" two-stroke dual-fuel engine operation (Ott 2015). As observed from Fig. 7.16b, the maximum rating of "Otto-cycle" dual-fuel engine is lower than conventional diesel engine due to knocking/pre-ignition limitations (Ott 2015). Green dotted lines represent selected rating point of a two-stroke diesel engine in standard ship designs, and as observed from Fig. 7.16b, "Otto-cycle" dual-fuel engine operation is covering more than 90% of these rating points (Ott 2015). According to the engine manufacturer of the "Otto-cycle" two-stroke dual-fuel engine only on exceptional cases, an additional cylinder will be needed to meet engine power output requirements (Ott 2015). Regarding an important issue of "Otto-cycle" gas engines, which is combustion stability, Fig. 7.16c shows experimental results for the variation of indicated mean effective pressure (IMEP) with engine cycles number (300 cycles in total) for conventional two-stroke diesel engine and two-stroke "Otto-cycle" dual-fuel engine (Ott 2015). As evidenced by Fig. 7.16c, in the case of "Otto-cycle" dual-fuel engine, a stable combustion was observed since cycle-to-cycle variation of IMEP of "Otto-cycle" gas engine was comparable to pertinent IMEP variation of conventional two-stroke diesel engine (Ott 2015). Hence, in terms of combustion stability, "Otto-cycle" dual-fuel engine operation does not bring any serious problem compared to conventional diesel operation.

Regarding the environmental performance of "Otto-cycle" two-stroke dual-fuel engines, Fig. 7.17a shows measured values of NO_x emissions of this type engine in contrast with IMO Tier limits (Ott 2015; Hagedorn 2014). As evidenced by Fig. 7.17a, "Otto-cycle" two-stroke dual-fuel engine clearly emits lower values of NO_x emissions compared to the most stringent IMO NO_x regulations (Tier III) revealing direct compliance of these engines with Tier III without any need for in-cylinder measures or after-treatment technologies. In Fig. 7.17b, EEDI value of

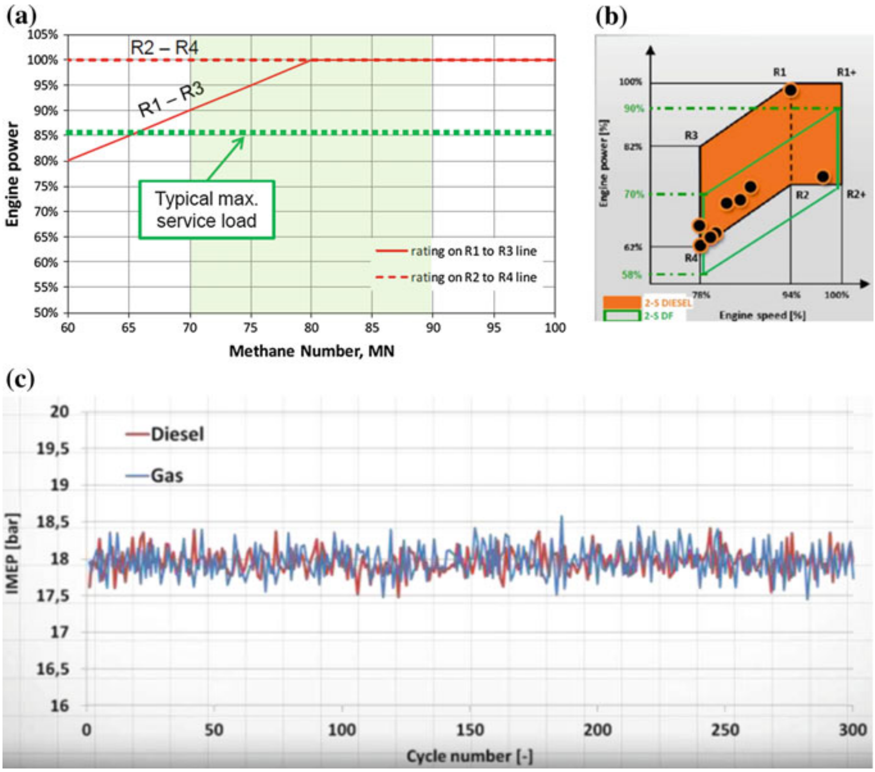


Fig. 7.16 **a** Effect of natural gas MN on two-stroke "Otto-cycle" dual-fuel engine power (Ott 2015), **b** Power layout diagrams for two-stroke diesel engine and two-stroke "Otto-cycle" dual-fuel engine (Stiefel 2015), **c** IMEP variation with engine cycles number for diesel-only and "Otto-cycle" gas engine operation (Ott 2015)

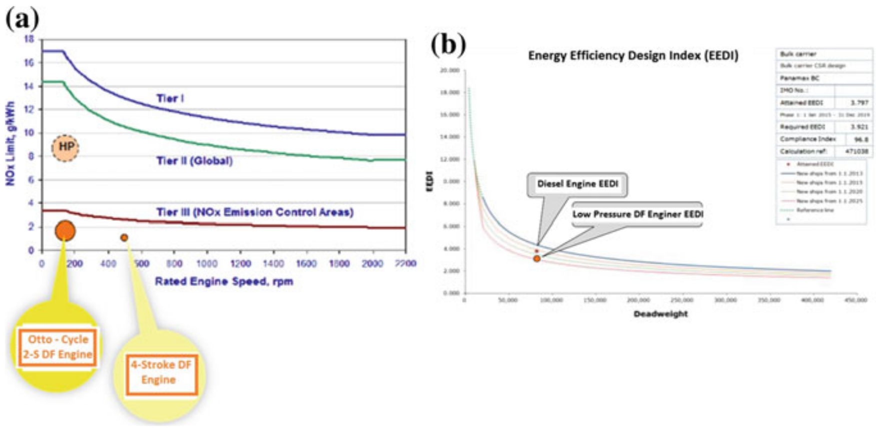


Fig. 7.17 **a** Compliance of "Otto-cycle" two-stroke dual-fuel engine with IMO Tier III limits (Hagedorn 2014), **b** EEDI values of a conventional two-stroke diesel engine and of an "Otto-cycle" two-stroke dual-fuel engine (Stiefel 2015)

low-pressure dual-fuel engine is compared with pertinent value of a conventional two-stroke diesel engine, and as evidenced, EEDI value of the "Otto-cycle" gas engine is below conventional diesel engine EEDI point due to lower overall CO_2 emissions (Ott 2015; Hagedorn 2014).

7.4.2 *Four-Stroke Dual-Fuel and Gas SI Engines*

To examine the effect of natural gas combustion in the performance characteristics and pollutant emissions of four-stroke marine dual-fuel and gas engines is described a detailed experimental investigation performed by an engine research and development company (Schlick 2014). The specific experimental investigation (Schlick 2014) was conducted in the single cylinder turbocharged (T/C) CI engine FM250. The specific engine may operate with diesel fuel only or burning almost only natural gas with micro-pilot fuel injection or as a dual-fuel engine. The maximum allowable value of peak cylinder pressure is 250 bar (Schlick 2014). AVL FM250 engine uses a high-pressure common rail fuel injection system, whereas natural gas supply can be attained either with central mixing or with gas injection in engine intake (Schlick 2014). The engine research and development company performed engine tests in FM250 engine considering the following:

- FM250 is operating according to E3 cycle. In this cycle tests, the engine is operating as main marine dual-fuel engine, whereas engine loading by the brake is following the propeller curve.

- FM250 is operating according to E2 cycle. In this cycle tests, the engine is operating as a main gas engine under constant engine rotating speed with natural gas supply in the pre-chamber.
- FM250 is operating according to D2 cycle. In this cycle tests, the engine is operating as auxiliary gas engine generating electric power under constant engine speed with natural gas supply in the pre-chamber.

In Fig. 7.18 are shown the experimental results from the CI engine FM250 for the variation of brake mean effective pressure (BMEP), brake efficiency, peak cylinder pressure, pilot fuel quantity, and 50% mass fraction burned (Fig. 7.18a). Also are shown experimental results from FM250 for air excess ratio, specific NO_x, specific THC, specific CO, and smoke number (Fig. 7.18b). Experimental results shown in Fig. 7.18a and b have been obtained for E3 cycle (Schlick 2014).

From the examination of Fig. 7.18a is observed that the highest brake efficiency (almost 42.5%) of dual-fuel engine FM250 is attained at maximum engine speed and maximum power of FM250 with only 1% pilot fuel quantity (Schlick 2014). In addition, it is observed that when the engine speed is decreased the pilot fuel injection timing and the pertinent 50% mass fraction burned (MFB50%) should be decreased in order the covariance (COV) of indicated mean effective pressure (IMEP) should be lower than 2% at all examined cases. Also when engine speed is decreased, the pilot fuel quantity should be increased for effective commencement of combustion (i.e., constant NO_x emissions). From the examination of the results shown in Fig. 7.18a is can be observed that the peak cylinder pressure did not

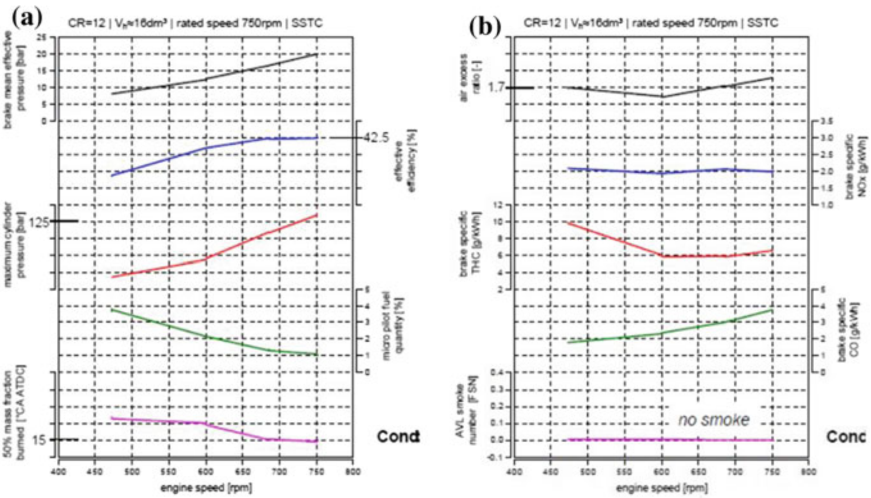


Fig. 7.18 Experimental results from the CI engine FM250 for the variation of brake mean effective pressure, brake efficiency, peak cylinder pressure, pilot fuel quantity, and 50% mass fraction burned (**a**). Also are shown experimental results from FM250 for air excess ratio, specific NO_x, specific THC, specific CO, and smoke number (**b**). Experimental results shown in figures **a** and **b** have been obtained for E3 cycle: dual-fuel operation on propeller curve (Schlick 2014)

exceed the limit of 250 bar, and thus, there was no problem with the mechanical strength of FM250 engine (Schlick 2014).

As evidenced by the examination of the experimental results shown in Fig. 7.18b, the air excess ration remains almost the same with variation of engine speed and load. This proves that FM 250 engine had the same air–fuel analogy during E3 cycle tests (Schlick 2014). According to NO_x results in Fig. 7.18b, FM 250 attained an extremely low level of NO_x emissions at all four points of E3 cycle. On the other hand, it is noteworthy to mention the high values of specific THC and CO emissions during E3 cycle tests, which are noticeably higher, compared to the ones of conventional diesel operation (Schlick 2014). Finally, it is quite encouraging that the smoke emissions are not a significant issue during FM250 operation as a main dual-fuel engine since its absolute values were almost zero (i.e., smokeless operation) (Schlick 2014).

In Fig. 7.19 are shown the experimental results from the CI engine FM250 for the variation of brake mean effective pressure (BMEP), brake efficiency, peak cylinder pressure, pilot fuel quantity, and 50% mass fraction burned (Fig. 7.19a). Also are shown experimental results from FM250 for air excess ratio, specific NO_x, specific THC, specific CO and smoke number (Fig. 7.19b). Experimental results shown in Fig. 7.19a and b have been obtained for E2/D2 cycle (Schlick 2014).

As evidenced by Fig. 7.19a, the brake efficiency is increased with engine load and its highest value is observed at highest engine load and it is equal to 46%

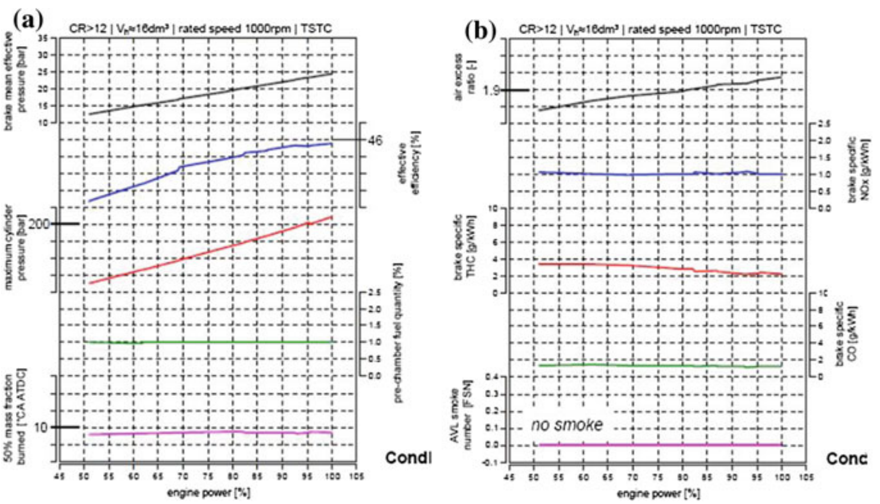


Fig. 7.19 Experimental results from the CI engine FM250 for the variation of brake mean effective pressure, brake efficiency, peak cylinder pressure, pilot fuel quantity, and 50% mass fraction burned (**a**). Also are shown experimental results from FM250 for air excess ratio, specific NO_x, specific THC, specific CO, and smoke number (**b**). Experimental results shown in figures **a** and **b** have been obtained for E2/D2 cycle: gas engine operation with gas-fueled pre-chamber (Schlick 2014)

(Schlick 2014). Peak cylinder pressure remained at high levels at all engine loads and for load higher than 95% exceeded the limit of 200 bar (Schlick 2014). Hence, FM250 engine operation with micro-pilot injection in the pre-chamber leads to higher values of peak cylinder pressure compared to dual-fuel operation of the same engine under E3 cycle. As evidenced by the variation of 50% mass fraction burned engine operation was stable at all engine loads with IMEP covariance less than 1% at all cases (Schlick 2014). As observed from Fig. 7.19a, 1% micro-pilot fuel quantity in the pre-chamber is enough for strong ignition, which leads to fast flame propagation in the main combustion chamber (Schlick 2014). According to Fig. 7.19b, the air excess ration should be increased with engine load increase starting from 1.9 and reaching up to 2.0 at maximum engine load (Schlick 2014). During increase of air excess ration with increasing engine load, specific NO_x emissions remained constant. It is of utmost importance that when FM250 engine operated as gas engine with micro-pilot injection in the pre-chamber was compliant with the most stringent IMO regulations for NO_x emissions in NECAs (Tier III). In addition, it is quite important that specific THC and specific CO emissions are significantly lower compared to the values measured during FM 250 engine operation as a dual-fuel engine under E3 cycle (Schlick 2014). Finally, according to Fig. 7.19b, at all examined cases of FM250 engine operation as gas engine with micro-pilot injection in the pre-chamber, zero smoke emissions were observed (Schlick 2014).

In Fig. 7.20a is shown the variation of measured specific NO_x emissions with engine speed for four-stroke main diesel engines, four-stroke pre-chamber spark-ignition (SI) engines, four-stroke main dual-fuel engines and four-stroke diesel main, and auxiliary engines (Schlick 2014). As evidenced by Fig. 7.20a, four-stroke main and auxiliary diesel engines are compliant only with IMO Tier II, which is issued worldwide outside of NECAs. On the other hand, four-stroke main dual-fuel engines and auxiliary pre-chamber SI engines are compliant with the most

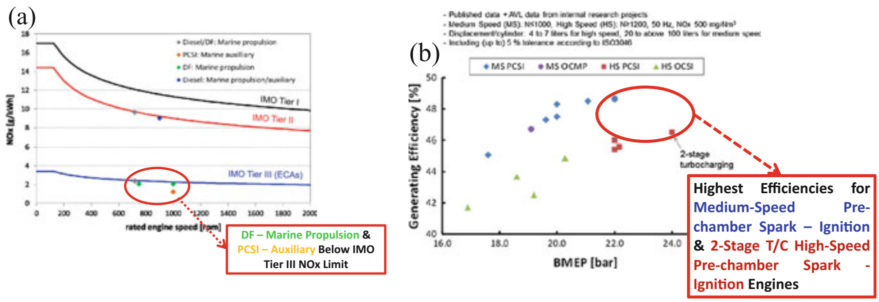


Fig. 7.20 **a** Experimental results for specific NO_x emissions for four-stroke main diesel engines, four-stroke pre-chamber spark-ignition (SI) engines, four-stroke main dual-fuel engines and four-stroke diesel main and auxiliary engines (Schlick 2014), **b** experimental results for the variation of generating efficiency with BMEP for four-stroke medium-speed pre-chamber SI engines, medium-speed open-chamber gas engines with micro-pilot injection, high-speed pre-chamber SI engines, and high-speed open-chamber SI engines (Murakami and Baufeld 2013; Mohr and Baufeld 2013)

stringent IMO NO_x limits (Tier III), which are issued in NECAs (Schlick 2014). In fact, auxiliary pre-chamber SI engines are below Tier III NO_x limits, which reveal the superiority of this type of marine auxiliary engines regarding their NO_x performance compared to other types of marine main and auxiliary engines.

In Fig. 7.20b is shown experimental results from previous studies (Murakami and Baufeld 2013; Mohr and Baufeld 2013) for the variation of generating efficiency with BMEP for four-stroke medium-speed pre-chamber SI engines, medium-speed open-chamber gas engines with micro-pilot injection, high-speed pre-chamber SI engines, and high-speed open-chamber SI engines. As evidenced, the highest generating efficiencies are observed for medium-speed pre-chamber SI engines and for two-stage T/C high-speed pre-chamber SI engines. Specifically, the highest possible generating efficiency is close to 49% and is observed for medium-speed pre-chamber SI engines (Murakami and Baufeld 2013; Mohr and Baufeld 2013). These results reveal the superiority of pre-chamber technology in terms of generating efficiency compared to open-chamber technology (Murakami and Baufeld 2013; Mohr and Baufeld 2013).

Except the operational and environmental characteristics of four-stroke dual-fuel engines and four-stroke lean-burned gas engines, is essential to compare the usage of LNG in four-stroke marine engines compared to other available fuel types and IMO SO_x and NO_x limits compliant solutions. For this reason, in Fig. 7.21a–h are shown results from a detailed economic and environmental analysis performed for a Ro-Ro vessel, which was scheduled to operate in ECAs (Levander 2011; Yfantis et al. 2017). Specifically, in this particular analysis were examined three available solutions in order the specific Ro-Ro vessel to be compliant with SO_x and NO_x limits in ECAs: (a) combustion of marine gas oil (MGO) in main and auxiliary four-stroke engines in conjunction with SCR for reducing NO_x emissions to IMO Tier III limit, (b) combustion of heavy fuel oil (HFO) in main and auxiliary engines in conjunction with SO_x scrubber and SCR for reducing NO_x emissions to IMO Tier III limit, and (c) use of dual-fuel main and auxiliary engines without exhaust after-treatment devices (Levander 2011). Figure 7.21a shows the relative annual fuel consumption and the corresponding relative cost with respect to MGO operation and as evidenced LNG operation indicates the lowest relative annual fuel cost compared to MGO and HFO operation (Levander 2011). Figure 7.21b shows the annual fuel, lubricant oil, and consumables (i.e., NaOH—fresh water for scrubbers, chemicals, and urea for SCR) cost (Levander 2011). As evidenced by Fig. 7.21b, the LNG solution indicates the lowest annual fuel cost compared to MGO and HFO operation. Also, LNG solution does not indicate any consumable cost since there is no need for NaOH, urea or chemicals (Levander 2011). According to Fig. 7.21c, which illustrates the machinery investment cost, LNG solution has the same total capital expenses (CAPEX) with HFO/scrubber solution whereas both LNG solution and HFO/scrubber solution have higher total CAPEX compared to MGO solution (Levander 2011). As evidenced by Fig. 7.21d, which depicts annual machinery cost for the three ECAs solutions, the LNG solution has noticeably lower operational expenses (OPEX) compared to MGO solution and HFO/scrubber solution (Levander 2011). According to Fig. 7.21e, which presents the relative payback times (PT) of each ECA solution with respect to MGO operation, the LNG solution has higher PT compared to MGO solution and significantly lower PT compared to HFO/scrubber solution (Levander 2011). Figure 7.21f demonstrates the Net Present Values (NPV) of each ECA solution considering a 10-year operation and as evidenced, LNG solution has the highest NPV compared to MGO solution and to HFO/scrubber solution (Levander 2011). Figure 7.21g shows the relative CO_2, NO_x and SO_x emissions of HFO/scrubber and LNG solutions compared to MGO solution. As observed from Fig. 7.21g, the LNG solution indicates the lowest CO_2 emissions compared to other two solutions, slightly higher NO_x emissions compared to MGO/SCR solution and almost same NO_x emissions with HFO/Scrubber/SCR solution. Also LNG solution does not generate any SO_x emissions (Levander 2011). Finally, according to Fig. 7.21h, which depicts the EEDI values of the three examined ECA solutions, LNG alternative indicates the lowest EEDI value compared to the other two ECA solutions (Levander 2011).

7.4.3 Evaluation of Different Propulsion Systems of LNG Carriers Using Energy Efficiency Design Index (EEDI)

Ekanem Attah and Rucknall (Ekanem Attah and Bucknall 2015) performed a detailed evaluation of different propulsion systems of LNG carriers using the Energy Efficiency Design Index (EEDI). In their analysis considered the impact of methane slip emissions on the calculation of overall GHG emissions. In the following sections, the alternative natural gas ship propulsion systems considered in the study of Ekanem Attah and Rucknall (Ekanem Attah and Bucknall 2015) will be described. It will be described also the EEDI analysis methodology and for each propulsion system will be discussed the main findings of the EEDI analysis.

7.5 Description of Alternative LNG Carriers Propulsion Systems

Steam Turbines Propulsion System (*STPS*). STPS ships obtain the 71% of the existing LNG carriers' fleet (Ekanem Attah and Bucknall 2015; Chang et al. 2008a; Clarksons Shipping Intelligence Network 2014a; Wayne and Hogson 2006; American Bureau of Shipping 2014; Bureau Veritas 2014), and this high percentage is attributed to the easy handling of boil-off gas (BOG) in these ships, the simple operation, and their internal safety. When the LNG tank pressure is elevated, then the steam generators are burning BOG for high-pressure steam generation, which expands in the steam turbines. Steam turbines are connected to ship propellers. When the engine load is not sufficient for burning all BOG mass, the remainder natural gas is directed to the condensers for being liquefied again. This simple

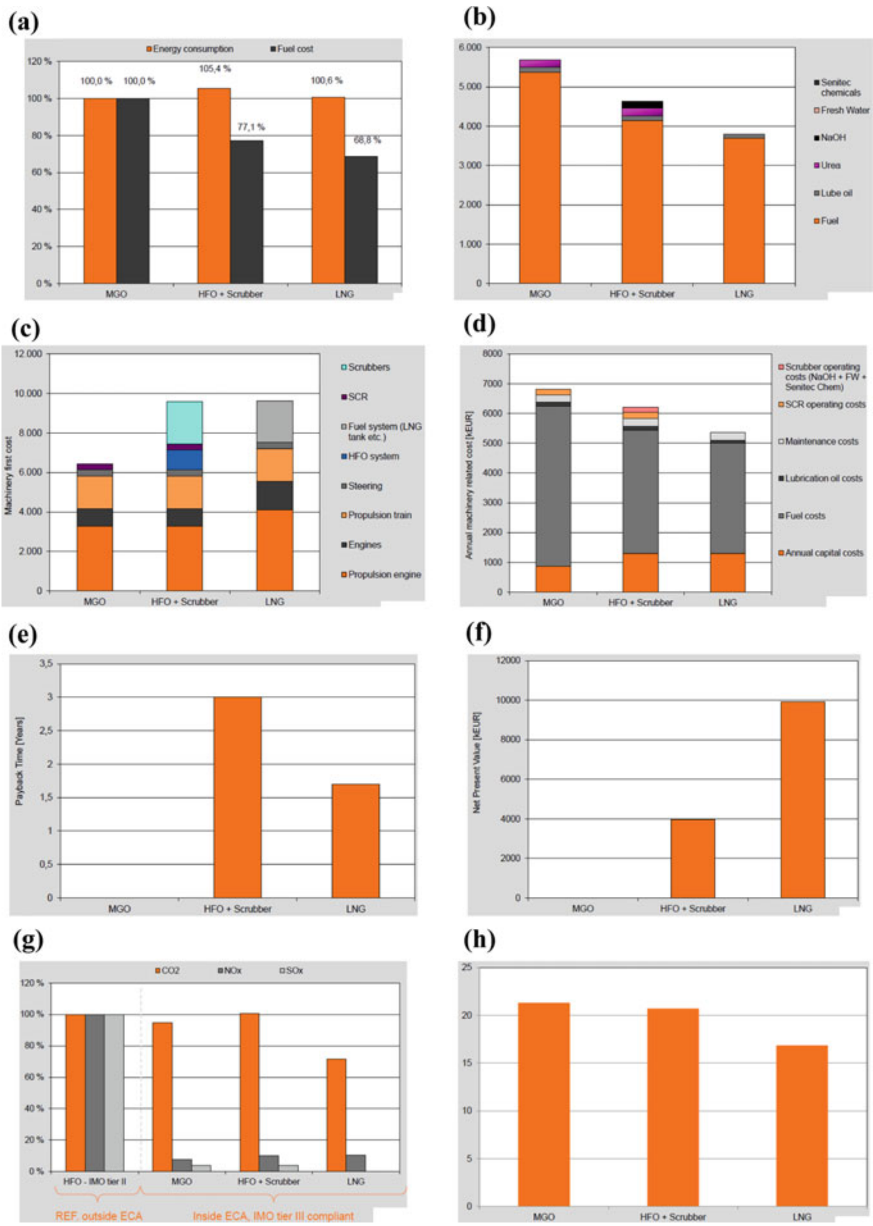


Fig. 7.21 Results of the environmental and economic analysis of a Ro-Ro vessel operation in ECAs **a** annual fuel consumption and relative cost versus MGO operation, **b** annual fuel, lube oil, and consumables cost, **c** machinery investment cost, **d** annual machinery cost, **e** payback time (expressed as relative value with respect to MGO operation), **f** net present value assuming 10 years operation, **g** CO_2, NO_x, and SO_x emissions relative values compared to the ones of MGO operation and **h** EEDI values (Levander 2011)

strategy eliminates the need for one gas combustion unit, which is demanded for the other two gas propulsion systems considered in the study of Ekanem Attah and Bucknall (Ekanem Attah and Bucknall 2015): the traditional slow-speed two-stroke diesel engine with re-liquefaction plant (SSDR) and the dual-fuel diesel–electric (DFDE) propulsion system. A schematic view of the LNG carrier propulsion system based on steam turbines (STPS) is given in Fig. 7.22. The specific installation is comprised of two steam generators using natural gas/heavy fuel oil as fuel. The generated steam besides expansion in steam turbines is used in auxiliary systems including electric generators and pumps (Ekanem Attah and Bucknall 2015; Chang et al. 2008a; Clarksons Shipping Intelligence Network 2014a; Wayne and Hogson 2006; American Bureau of Shipping 2014; Bureau Veritas 2014).

Slow-speed two-stroke diesel engines with re-liquefaction of Boil-Off Gas (BOG). The slow-speed two-stroke diesel engines installation is an integrated propulsion system with re-liquefaction installation of boil-off gas, where the BOG is liquefied and returns to storage tanks instead of being burned in the engines. The layout of the slow-speed diesel engines in such systems is usually based on the twin screw layout with two slow-speed diesel engines directly connected to two propellers as evidenced by Fig. 7.23 (Ekanem Attah and Bucknall 2015; Chang et al. 2008a; Clarksons Shipping Intelligence Network 2014a; Wayne and Hogson 2006; American Bureau of Shipping 2014; Bureau Veritas 2014; Chang et al. 2008b). This propulsion unit is also equipped with a gas control unit for selecting BOG in cases where the BOG capacity is higher than the re-liquefaction installation capacity. The BOG liquefaction installation is based on a closed nitrogen cycle

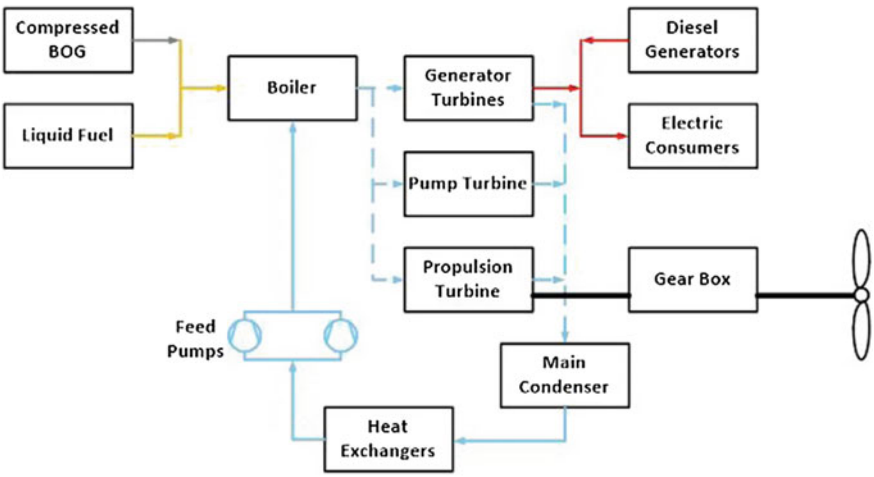


Fig. 7.22 Schematic view of the installation of a steam turbine propulsion system (STPS) with natural gas (Ekanem Attah and Bucknall 2015; Chang et al. 2008b) "*Figure reproduced from 35. Ekanem Attah, E. and Bucknall, R.: An Analysis of the Energy Efficiency of LNG Ships Powering Options using the EEDI. Ocean Engineering, Vol. 110, pp. 62–74, 2015 with permission from Elsevier*"

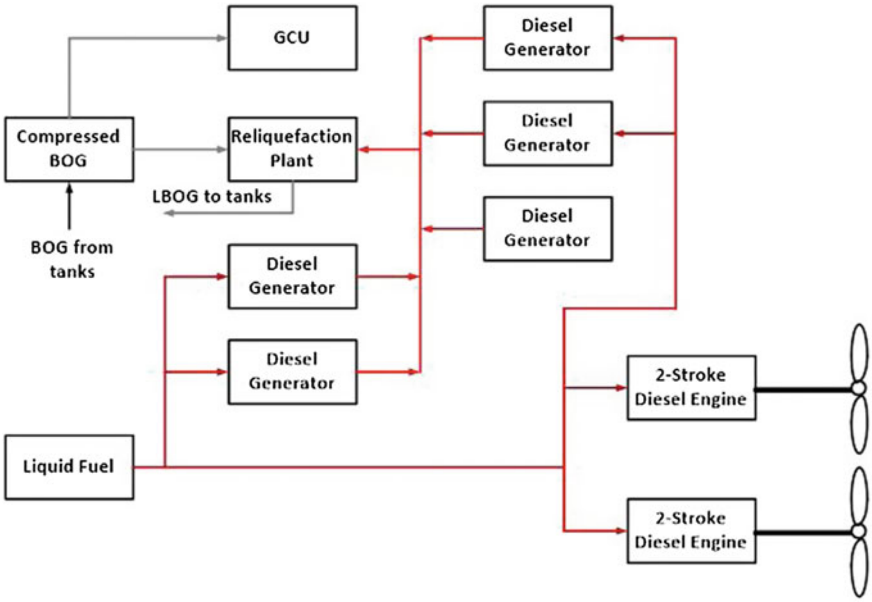


Fig. 7.23 Schematic view of ship propulsion system with slow-speed two-stroke dual-fuel diesel engines and re-liquefaction devices of the boil-off gas (BOG) (Ekanem Attah and Bucknall 2015; Chang et al. 2008b) "*Figure reproduced from 35. Ekanem Attah, E. and Bucknall, R.: An Analysis of the Energy Efficiency of LNG Ships Powering Options using the EEDI. Ocean Engineering, Vol. 110, pp. 62–74, 2015 with permission from Elsevier*"

aiming to the heat extraction from the BOG. This ensures condensation of all hydrocarbons contained in the natural gas in a way that they all transformed again in LNG, whereas nitrogen and the other non-condensed gases remain as gas bubbles inside LNG. However, these gas bubbles are separated using a liquid separator where LNG is separated and is pumped back to the storage tanks and the non-condensed gases indicating high nitrogen concentration are rejected to the atmosphere or they burned in the gas combustion unit (GCU) (Ekanem Attah and Bucknall 2015). For LNG carriers, this additional re-liquefaction system will induce an additional electric power load between 3 and 4 MW though that some current LNGC propulsion installations with CI engines have gas capacities between 216,000 and 260,000 m^3 demanding a parasitic electric power ranging from 4.5 to 5.5 MW (Ekanem Attah and Bucknall 2015).

Dual-Fuel Diesel–Electric Propulsion—DFDE. This propulsion system contains retrofitted diesel engines in order they be able to burn BOG besides diesel oil as evidenced by Fig. 7.24 (Ekanem Attah and Bucknall 2015; Chang et al. 2008a; Clarksons Shipping Intelligence Network 2014a; Wayne and Hogson 2006; American Bureau of Shipping 2014; Bureau Veritas 2014; Chang et al. 2008b). This propulsion installation uses multiple diesel generators, usually four, in order all powering needs of the ship to be covered. The ship in this case contains the main

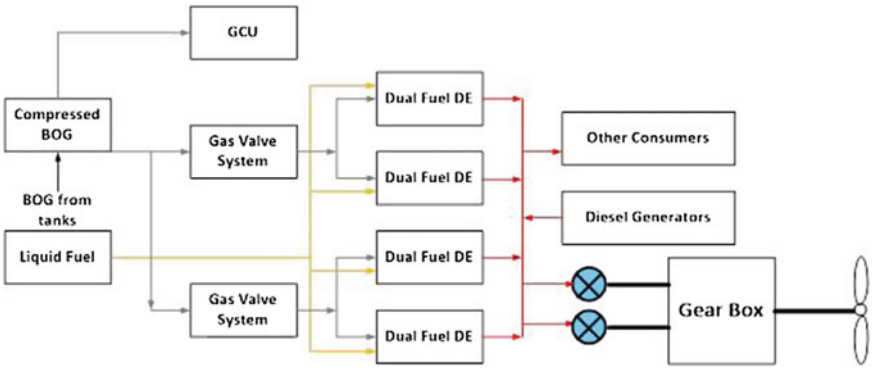


Fig. 7.24 Schematic view of the propulsion system installation with dual-fuel diesel–electric propulsion (DFDE) (Ekanem Attah and Bucknall 2015; Chang et al. 2008b) "*Figure reproduced from 35. Ekanem Attah, E. and Bucknall, R.: An Analysis of the Energy Efficiency of LNG Ships Powering Options using the EEDI. Ocean Engineering, Vol. 110, pp. 62–74, 2015 with permission from Elsevier*"

system since the diesel generators produce electric energy and the electric motors use this electric energy for ship propelling (Ekanem Attah and Bucknall 2015). However, the diesel engine operates with natural gas, which is inducted in the engine through inlet valves gas, whereas ignition in each cylinder is induced using pilot diesel injection. The diesel engines of these systems can also operate under diesel-only operation. However, they cannot operate with both natural gas and diesel oil. This is a deficiency of the DFDE system compared to the STPS system, where steam generators can burn efficiently different proportions of gas and diesel simultaneously (Ekanem Attah and Bucknall 2015). Multiple diesel–electric generators operation provides operational flexibility and increased autonomy (Ekanem Attah and Bucknall 2015).

Methodology of Comparative Evaluation of Natural Gas Propulsion Systems based on EEDI Though that most of the energy efficiency indices are based on the simple principle of using less energy for the production of the same amount of useful energy, there is a current trend for the quantification of the energy efficiency in relevance with the corresponding environmental merits such as the reduction of anthropogenic emissions (Ekanem Attah and Bucknall 2015). This is the case with International Maritime Organization (IMO), which has defined and issued the Energy Efficiency Design Index (EEDI), which is based on mathematical formula used for the calculation of CO_2 mass emitted per transportation work (metric tons per nautical mile) of a specific energy production unit. Actual EEDI is calculated using mathematical formulas and directives published by IMO, and the calculated EEDI should be lower than a predefined initial value, which is progressively be reduced during a period of five years. The compliance of actual EEDI with the continuous reduced predefined limits during a period of five years is expected to trigger more effective CO_2 reduction methods than the initial ones considered. In the study of Ekanem Attah and Bucknall (Ekanem Attah and Bucknall 2015), EEDI is used an analytical tool for the evaluation of the efficiency of the current fleet of LNG carriers.

Though that the EEDI is valid only for new buildings, many studies have tried to use EEDI as a tool for the evaluation of the efficiency of existing ship projects aiming to the prediction of the impact this IMO regulation has in conjunction with specific propulsion technology or to improve EEDI values for the future energy consumption plans of the examined ship. Due to the fact that the majority of LNG carriers use unconventional propulsion systems, and thus, they were excluded from the initial EEDI regulations, and most of the studies are focused to an EEDI calculation method for these unconventional propulsion systems (Ekanem Attah and Bucknall 2015). However, there is any study analyzing EEDI of ships using steam turbines as propulsion system, which was corresponded to 80% of the ships at the moment (Ekanem Attah and Bucknall 2015). However, in 2011 were published more detailed studies covering more propulsion types (Ekanem Attah and Bucknall 2015). These studies contained publications describing an EEDI calculation method for STPS and DFDE technologies (Ekanem Attah and Bucknall 2015). Hence, these studies approved by IMO as a standard, one which the current EEDI regulations for LNG carriers are based (Ekanem Attah and Bucknall 2015).

One major advantage of the EEDI analysis is that it is based on well-established regulations, and thus, its importance for future LNG carriers (LNGC) plans should be taken into consideration. It is expected that all new LNGC EEDI regulations, which have approved in 2014, will be issued from September 2015 for all newly accepted LNGCs (Ekanem Attah and Bucknall 2015). Another virtue of LNGC EEDI analysis is that this efficiency analysis is very detailed, and it can be used effectively for assessing the efficiency and the emitted GHG values of different propulsion systems such as steam turbines (STPS), dual-fuel diesel–electric propulsion (DFDE), and slow-speed direct drive (SSDR) propulsion. EEDI analysis does not take into account other emissions except CO_2 (Ekanem Attah and Bucknall 2015). Though that the majority of other maritime emissions such as NO_x and SO_x are covered by other IMO regulations, there are GHG emissions such as methane emissions (i.e., methane slip), which at the moment are not covered by any IMO regulation (Ekanem Attah and Bucknall 2015). This observation creates the problem that there is possibility the operation of a specific propulsion system to result in reduction of CO_2 emissions and simultaneously to result in the deterioration of uncontrolled emissions such as methane slip. In addition, the use of a common EEDI reference value for three different propulsion technologies and their pertinent efficiencies is possible to cause a high data dispersion as we moving from the less efficient STPS to more efficient configurations such as DFDE (Ekanem Attah and Bucknall 2015).

Analysis of EEDI Methodology The main objectives of the Ekanem Attah and Bucknall study (Ekanem Attah and Bucknall 2015) are the quantification of the CO_2 values emitted from ships, and through this process, the definition of CO_2 reference curves for new buildings. EEDI regulations have issued for many different ship types such as tankers, container ships, and cargo ships from January 2013, whereas LNG carriers have been excluded from these regulations due to the initial difficulties in the calculation of EEDI values for steam turbines propulsion system and diesel–electric propulsion system, which both correspond to 90% of the current LNGC fleet and of the corresponding future predictions (Ekanem Attah and Bucknall 2015). However, after an overall re-evaluation, IMO has approved amendments in MARPOL Annex VI to expand EEDI application in LNG ships and this amendment was published in April 2014, whereas its application has initiated from September 2015 (Ekanem Attah and Bucknall 2015). The baseline adopted for LNGCs is shown in Eq. (7.1), whereas details for the corresponding phase distribution can be found in Ekanem Attah and Bucknall (2015).

$$\text{Baseline value } = 2253.7 \times \text{ deadweight}^{-0.474} \tag{7.1}$$

EEDI Analysis of Current LNGC Fleet A statistical analysis of current LNGC fleet was performed by (Ekanem Attah and Bucknall 2015) to predict effectively the repercussions of EEDI basic limits for LNGCs regarding the design of future LNG carriers. Current LNGC fleet data were obtained from Clarkson's World Fleet Register taken into account only ships built in 2000 or later. Estimation for these ships is calculated using EEDI, which takes only into consideration the generated powers of the main and auxiliary engines, standardized fuel consumptions (based on IMO MEPC 65) (International Maritime Organization 2017; Ekanem Attah and Bucknall 2015; Clarksons Shipping Intelligence Network 2014a; Clarksons Shipping Intelligence Network 2014b), with capacity in tons deadweight and the speed of the ship, both obtained from the Clarkson's World Fleet Register and verified from the corresponding class societies registers (American Bureau of Shipping 2014; Bureau Veritas 2014). More details about the EEDI analysis of current LNGCs fleet can be retrieved from Ekanem Attah and Bucknall study (Ekanem Attah and Bucknall 2015).

EEDI Calculation Considering Methane Slip. According to Ekenam Attah and Bucknall (Ekanem Attah and Bucknall 2015) though that with dual-fuel diesel–electric propulsion (DFDE) are attained lower EEDI values compared to propulsion with steam turbines and propulsion with slow-speed two-stroke dual-fuel engines, DFDE suffers from increased methane slip. The term "methane slip" corresponds to unburned methane emitted to the atmosphere from internal combustion engines. Methane (CH_4) has severe negative impact on greenhouse phenomenon, and thus, this hazardous environmental impact of methane compromises seriously the environmental benefits from reduced EEDI values in CO_2 emissions (Ekanem Attah and Bucknall 2015). This effect is considerably worrying since methane has 20–25 times worst impact on greenhouse phenomenon compared to CO_2 if the effects are calculated on a life cycle of 100 years whereas the corresponding effects calculated on a life cycle of 20 years the negative greenhouse effect of methane compared to CO_2 is 72 times worst (Ekanem Attah and Bucknall 2015). This means that the release of even small gas volumes to the atmosphere will counterbalance all the merits from the reduction of CO_2 emissions caused by the improved engine efficiency. Hence, during calculation of CO_2 emissions effects based on EEDI mathematical formulas, methane emissions should be taken into account as equivalent CO_2 emissions (Ekanem Attah and Bucknall 2015).

Methane slip issue is more intense in four-stroke engines used in dual-fuel diesel–electric propulsion compared to gas injection and diesel engines mainly because in four-stroke DFDEs the unburnt methane is trapped in combustion chamber crevices such as piston rings and valve seats. In these crevices, the fuel–air mixture equivalence ration has such a value that it cannot be totally burnt during combustion leading to methane emission with other exhaust gases through the exhaust valves during expansion stroke (Ekanem Attah and Bucknall 2015). On the other hand, natural gas injection engines operate with natural gas direct injection as in conventional diesel engines ensuring that no gas is present during compression stroke or during scavenging process reducing, thus, methane emissions to levels, which are comparable with the ones of conventional liquid fuels (Ekanem Attah and Bucknall 2015).

Considerable research has been performed aiming to the reduction of methane slip, which focuses on the use of combustion pre-chamber and on the improvement of combustion technology (optimization of injection timing, increase of injection pressure, and increase of inlet air temperature) (Ekanem Attah and Bucknall 2015). In the case that the reduction of methane slip inside the engine combustion chamber is not feasible, it has been examined the use of oxidation catalyst for methane capture in engine's exhaust. However, for most DFDE engine technologies, methane slip has been reduced to 3–4 g/kWh compared to 8–15 g/kWh of existing DFDEs (Ekanem Attah and Bucknall 2015). Having given that the mean specific fuel consumption (SFC) of DFDEs is approximately 175 g/kWh, the following formulas are used for the transformation of methane slip to equivalent CO_2 emissions (Ekanem Attah and Bucknall 2015):

$$\begin{aligned} &\text{Methane slip } = 8\text{ g/kWh} \\ &\text{SFC } = 175\text{ g/kWh} \\ &\text{Methane equivalent of SFCme } = 8/175 = 4.57\% \end{aligned}$$

Assuming 1 ton of gas fuel containing methane is burning in DFDEs, then 2.75 tons of CO_2 are produced assuming 4.57% methane slip. Hence, 1 ton of fuel gas containing methane generates 2.624 tons of CO_2 and 0.0457 tons of CH_4. Taking into account that in a life cycle of 100 years, the methane effect is 21 times higher than of this of CO_2 it is concluded that the total equivalent CO_2 emissions are (Ekanem Attah and Bucknall 2015):

$$\text{Total CO}_2\text{equiv } = 2.624 + (0.0457 \times 21) = 3.5837\text{ tCO}_2\text{equiv}$$

Taking into consideration that in a life cycle of 20 years, the negative impact of methane slip to greenhouse phenomenon is 72 times higher than that of CO_2

emissions is concluded that the total equivalent CO_2 emissions are (Ekanem Attah and Bucknall 2015):

$$\text{Total } CO_2\text{equiv} = 2.624 + (0.0457 \times 72) = 5.9144 \text{ t}CO_2\text{equiv}$$

Hence, the equivalent CO_2 emissions taking into account 1-ton methane slip to atmosphere are 3.5837 t CO_2 equivalent with 100 years life cycle analysis and 5.9144 t CO_2 equivalent with 20 years life cycle analysis (Ekanem Attah and Bucknall 2015).

Main Conclusions of the Theoretical Analysis of Three Different Propulsion Systems of LNGCS based on EEDI Ekanem Attah and Bucknall (Ekanem Attah and Bucknall 2015) have performed a detailed evaluation of effect of LNG transportation using EEDI. More specifically, they examined the energetic and the environmental performance of three different propulsion systems, which are used in LNG carriers in terms of CO_2 and CH_4 emissions. The three propulsion systems investigated were a steam turbines' propulsion system (STPS), a propulsion system equipped with slow-speed two-stroke dual-fuel engines (SSDR), and a propulsion system equipped with four-stroke dual-fuel CI engines (DFDE). The main conclusions of this elaborative study are the following (Ekanem Attah and Bucknall 2015):

- In terms of specific CO_2 emissions (in g per metric ton of ship capacity), DFDE indicates the lowest specific CO_2 emissions, whereas STPS indicates the highest specific CO_2 emissions (Ekanem Attah and Bucknall 2015).
- The current IMO EEDI limits are satisfied more or less from the 23% of the current LNG carriers' fleet. In this percentage of LNGCs, various propulsion systems are used. It is worth to mention that EEDI values corresponding to diesel–electric propulsion systems of LNGCs are considerably lower compared to the current EEDI baseline. Taking into account that 72% of future orders (reference year: 2015) for LNG carriers will have dual-fuel diesel–electric propulsion system from 2025 and onwards according to IMO dictations, they will demand additional energy efficiency improvement measures of the existing dual-fuel diesel–electric propulsion systems since they already comply with these limits (Ekanem Attah and Bucknall 2015).
- Two-stroke CI engines with natural gas injection, which correspond to the 60% of LNG carriers being ordered for construction in the near future and they will be used in direct drive propulsion systems it appears that they will offer almost 30% EEDI improvement compared to current two-stroke engines direct drive systems due to the reduction of specific fuel consumption, the reduction of CO_2 emissions and the elimination of the need for boil-off gas re-liquefaction installation. Such EEDI improvements of direct drive systems bring them to the same level (in terms of EEDI values) with four-stroke dual-fuel diesel–electric propulsion systems. Both two-stroke slow-speed direct drive systems and dual-fuel diesel–electric propulsion systems are fully complying with EEDI requirements from 2025 and onwards (Ekanem Attah and Bucknall 2015).

- When methane slip is taken into account in EEDI picture as equivalent CO_2 emissions, the energy efficiency of diesel–electric propulsion systems with dual-fuel engines is reduced. Taking into consideration a 100 years life cycle, total CO_2 emissions are increased by 30% compared to the corresponding EEDI value to be close or relatively lower to the current EEDI baseline curve. Oppositely considering a life cycle of 20 years, total CO_2 emissions are increased by 115% with the corresponding EEDI value to be considerably higher from the current EEDI baseline surpassing even the STPS, which has the lowest efficiency from all three systems examined (Ekanem Attah and Bucknall 2015).

7.6 Conclusions

In the present study, a detailed technological, environmental, and economic survey regarding natural gas combustion in marine two-stroke dual-fuel engines, marine four-stroke dual-fuel engines, and gas spark-ignition engines was performed. Starting from marine main two-stroke dual-fuel engines, the comparison of the "diesel-cycle" and the "Otto-cycle" dual-fuel engine types resulted in the derivation of the following conclusions:

- "Diesel-cycle" dual-fuel engine has higher fuel efficiency compared to the "Otto-cycle" dual-fuel engine type.
- "Diesel-cycle" dual-fuel engine has smoother load response and easier transition to diesel-only and back to gas/diesel operation compared to the "Otto-cycle" dual-fuel engine type.
- Unlike "Otto-cycle" dual-fuel engine, two-stroke diesel engine can be retrofitted for operating as two-stroke "diesel-cycle" dual-fuel engine.
- "Diesel-cycle" dual-fuel operation in contrast to "Otto-cycle" dual-fuel operation is not seriously affected by natural gas quality (i.e., MN).
- Both two-stroke dual-fuel engine types meet directly ECA SO_x requirements
- "Otto-cycle" engine type meets directly IMO NO_x Tier III limits, whereas "diesel-cycle" engine type requires EGR or SCR.
- Unlike "diesel-cycle" dual-fuel engine type, "Otto-cycle" engine type indicates a knocking risk and noticeable methane slip.

The detailed assessment of the four-stroke dual-fuel and gas SI engines on a technological, environmental, and economic basis resulted in the determination of the following conclusions:

- Modern four-stroke dual-fuel and gas SI engines have increased fuel efficiency (lower than two-stroke ones) and increased power density (higher than two-stroke ones) compared to the recent past.

- Pre-chamber technology ensures stable ignition and more complete combustion and also ensures improved combustion stability and reduced knocking risk.
- Dual-fuel marine engines and PCSI auxiliary engines meet directly ECA SO_x requirements and IMO NO_x Tier III limits, whereas indicate higher THC, methane slip, and CO emissions compared to conventional marine diesel engines.
- Conventional four-stroke marine CI engines can be retrofitted to dual-fuel engines
- Dual-fuel engines compared to other SO_x and NO_x ECA solutions indicate highest NPV and also SO_x free operation.

The most promising future propulsion system solutions for LNG carriers in terms of increased energy efficiency and IMO compliance for GHG emissions are the four-stroke dual-fuel CI engines and the slow-speed two-stroke dual-fuel CI engines, whereas there is an immediate requirement for inclusion of methane emissions (i.e., methane slip) in EEDI picture.

The increasing market share of dual-fuel diesel–electric propulsion and two-stroke main dual-fuel engines in current and future LNG carriers (Stiefel 2015) is anticipated to lead to the development of future two-stroke and four-stroke dual-fuel engines with superior performance characteristics and reduced gaseous and particulate emissions compared to recent past. It should be clearly underlined that the main purpose of the present study was to consolidate and critically evaluate existing knowledge in the field of marine natural gas internal combustion engines by demonstrating the advantages and the disadvantages of each engine-type solution. Consequently, the adaptation of natural gas as marine fuel in a specific vessel is a complex and multi-variable decision, which will be determined by many technical, operational, environmental, and economic aspects on a case-by-case basis.

References

American Bureau of Shipping (2014) Vessel's Database. [Online] Available at: https://www.eagle.org/safenet/record/record_vesselsearch. Accessed 30 on July 2014

Andre R (2013) Dual—fuel for maritime application. CIMAC paper no. 204, CIMAC Congress 2013, Shanghai, China

Brynolf S, Fridell E, Andersson K (2014) Environmental assessment of marine fuels: liquefied natural gas, liquefied biogas, methanol and bio-methanol. J Clean Prod 74:86–95

Bureau Veritas (2014) Online register of ships. [Online] Available at: http://www.veristar.com/portal/veristarinfo/generalinfo/registers/seaGoingShips. Accessed 01 Aug 2014

Callahan TJ, Hoag K (2013) An updated survey of gas engine performance development. CIMAC paper 277, CIMAC congress 2013, Shanghai, China

Chang D, Rhee T, Nam K, Lee S (2008a) Economic evaluation of propulsion systems for LNG carriers a comparative life cycle cost approach. [Online] Available at: http://osewiki.kaist.ac.kr/images/6/68/Week_3_Comparison_of_LNG_propulsion_options.pdf. Accessed 26 on June 2014

Chang D, Rhee T, Nam K, Lee S (2008b) A study on availability and safety of new propulsion systems for LNG carriers. [Online] Available at: http://ac.els-cdn.com/S0951832008001105/1-s2.0-S0951832008001105-main.pdf?_tid¼62906dd2-26f8-11e3-a0be-00000aacb35f&acdnat¼1380233579_df20deabaa3077eeeef5702c6cc239c2. Accessed 26on July 2014
Chryssakis C, Stahl S (2013) Well-to-propeller analysis of alternative fuels for maritime applications. CIMAC paper no. 265, CIMAC congress, Shanghai, China
Chryssakis C, Tvete H-A (2014) Alternative fuels for shipping. DNV GL strategic research and innovation, position paper 1–2014
Chryssakis C, Brinks H, King T (2015) The fuel trilemma: next generation of marine fuels. DNV GL strategic research and innovation, position paper 03–2015
Chryssakis C, Brinks H, Sames P, Morch C, Clausen N (2016) Feasibility and environmental impact of alternative fuels for shipping. CIMAC Paper 2016/132, 28th CIMAC world congress, Helsinki, Finland
Clarksons Shipping Intelligence Network (2014a) LNG carrier fleet [Online]. Available at: http://www.clarksons.net/sin2010/register/Default.aspx?sValues¼rOpt%3dregister%7crSel%3d3%7cZS0_SHIP_TYPE%3dVG9%7ctitle%3dLNGþGasþCarrierþFleet%7c. Accessed 18 on July 2014
Clarksons Shipping Intelligence Network (2014b) LNG carrier order book [Online] Available at: http://www.clarksons.net/sin2010/register/Default.aspx?sValues¼rOpt%3dorderbook%7crSel%3d3%7cZS0_SHIP_TYPE%3dVG9%7ctitle%3dLNGþGasþCarrierþOrderbook%7c. Accessed 18 on Aug 2014
Cleantech (2017) http://cleantech.cnss.no/policies-and-instruments/nox-emissions/
MAN Diesel (2017) http://hhpinsight.com/marine
MAN Diesel and Turbo (2017) Maritime costs and benefits of alternative fuels: key results from a DNV GL and MAN diesel and turbo joint study for an LR1 product tanker 2017
Dieselnet (2017) Emission standards, International: IMO marine engine regulations. https://www.dieselnet.com/standards/inter/imo.php#ghg
Ekanem Attah E, Bucknall R (2015) An analysis of the energy efficiency of lng ships powering options using the EEDI. Ocean Eng 110:62–74
Germanischer lloyd (2013) Will it be cost effective? Available at http://www.glgroup.com. Accessed August 8, 2013
Hagedorn M (2014) LNG engines—specifications and economics. LNG Shipping Rostock, Wärtsilä, 13/10/2014. http://www.golng.eu/files/Main/20141017/Rostock/LNG%20Shipping%20Session%20II%20-%20LNG%20Engines-Specifications%20and%20Economics-%20W%C3%A4rtsil%C3%A4,Ship%20Power%20-%20Hagedorn.pdf
Haraldson L (2011) LNG as a fuel for environmentally friendly shipping: retrofit perspective. In: 33rd motorship propulsion & emissions conference, Copenhagen 11–12 May 2011
Hiltner J (2013) Understanding the influence of heat transfer and combustion Behavior on end gas knock in heavy duty lean burn engines. CIMAC paper no. 36, CIMAC congress 2013, Shanghai, China
Hiltner J, Loetz A, Fiveland S (2016) Unburned hydrocarbon emissions from lean burn natural gas engines—Sources and solutions. CIMAC paper 2016/032, 28th CIMAC world congress 2016, Helsinki, Finland
International Maritime Organization (2017) Emission standards. http://www.imo.org
Kjemtrup N (2015) Gas 2-stroke marine engine design and operation. GCA Seminar Jan 2015. http://www.lme.ntua.gr:8080/whats-new/news-1/G2X_Presentations.pdf
Levander O (2011) Dual—fuel engines: latest developments. Wartsila, 27.9.2011, Hamburg, Germany http://www.ship-efficiency.org/onTEAM/pdf/PPTLevander.pdf
Li J, Wu B, Mao G (2015) Research on the performance and emission characteristics of the lng—diesel marine engine. J Nat Gas Sci Eng 27:945–954
McGill R, Remley W, Winther K (2013) Alternative fuels for marine applications. A report from the IEA advanced motor fuels implementing agreement report. http://www.iea-amf.org/app/webroot/files/file/Annex%20Reports/AMF_Annex_41.pdf

Mohr H, Baufeld T (2013) Improvement of dual—fuel engine technology for current and future applications. CIMAC paper no. 412, CIMAC congress, Shanghai, China

Moriyoshi Y, Xiong Q, Takahashi Y, Kuboyama T, Morikawa K, Yamada T, Suzuki M, Tanoue K, Hashimoto J (2016) Combustion analysis in a natural gas engine with pre-chamber to improve thermal efficiency. CIMAC paper 2016/262, 28th CIMAC congress, Helsinki, Finland, June 6–10, 2016

Murakami S, Baufeld T (2013) Current status and future strategies of gas engine development. CIMAC paper no. 413, CIMAC congress 2013, Shanghai, China

Ott M (2015) Low Pressure gas engines. The industry standard. CIMAC discussion Athens 22, Jan 2015 http://www.lme.ntua.gr:8080/whats-new/news-1/03_MO.pdf/

Roecker R et al (2016) State-based diesel fueling for improved transient response in a dual—fuel engine. CIMAC paper no. 161, 28th CIMAC congress, Helsinki, Finland, June 6–10, 2016

Sarigianidis C (2016) Innovative method for natural gas sea transport in compressed form (Autonomous CNG Transport Ship). (in Greek), 2016 Annual Meeting of Marine Technology, Marine Institute of Marine Technology, Athens, Greece, 2016

Schlick H (2014) Potentials and challenges of gas and dual—fuel engines for marine application. 5th CIMAC CASCADES, Busan, Korea, 23/10/2014. https://www.cimac.com/cms/upload/events/cascades/cascades_2014_busan/presentations/Presentation_Session2_AVL_CASCADES_Busan_Oct2014_Harald_Schlick.pdf

Stiefel R (2015) Low pressure gas engines, "The Industry Standard", CIMAC discussion, Wartsila, Athens 22, January 2015

Tozzi L, Sotiropoulou E, Beshouri G, Lepley D (2016) Novel pre-combustion chamber technology for large bore natural gas engines. CIMAC paper 2016/259. 28th CIMAC congress, Helsinki, Finland, June 6–10, 2016

Trauthwein G (2012) Fuel for thought, maritime reporter and engineering news

Wartsila 2-stroke dual fuel technology (2014) CIMAC NMA norge annual meeting, 22.01.2014. https://www.sintef.no/globalassets/upload/marintek/cimac2014/6—2-s-df-technology-cimac-norway-jan-22-2014.pdf

Wayne WS, Hogson M (2006) The options and evaluation of propulsion systems for the next generation of lng carriers. [Online] Available at: http://igu.dgc.dk/html/wgc2006/pdf/paper/add10055.pdf. Accessed 26 on Sep 2013

Wei L, Geng P (2016) A review on natural gas/diesel dual fuel combustion, emissions and performance. Fuel Process Technol 142:264–278

Yfantis EA, Zannis TC, Katsanis JS, Pariotis EG, Papagiannakis RG, Mohr H (2017) Four—stroke marine natural gas diesel and spark—ignited engines: a state-of-the-art technologies evaluation. In: 3rd scientific conference SEA-CONF 2017, May 18–19 2017, Constanta, Romania

Zannis TC, Yfantis EA Katsanis JS, Pariotis EG (2017) Natural gas combustion in marine internal combustion engines: a technological and environmental assessment. invited lecture, Hellenic Institute of Marine Technology, January 30th 2017, Piraeus, Greece