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Control of the combustion process and emission formation in marine gas engines - a review

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Abstract A smooth transition to the use of gas engines instead of conventional engines in marine shipping is a logical pathway for compliance with tightening environmental regulations. Currently, five major gas engine concepts are applied in maritime sector. In this paper, a review of the marine gas engine concepts was performed with a focus on the control of combustion and emission. To assess all the contributors to combustion and the emission formation process, three main factors were outlined: design, operational parameters and fuel. The assessment of gas engines was conducted based on these factors. The present paper helps to provide an understanding of the current progress in development of marine gas engines towards improving of combustion efficiency and reducing the emissions. Moreover the knowledge gaps, particularly in four-stroke marine high-pressure gas engines, were identified.

Keywords Combustion · Marine gas engines · Natural gas · Dual-fuel · Emission

1 Introduction

Environmental concerns and the corresponding regulations to reduce emissions from ships are creating a strong impulse for the development of new technological and operational solutions. The main attention is focused on measures to reduce combustion-born gases and particles that have negative effects on human health [1] and climate [2]. The oxidation of hydrocarbon-type fuels creates CO_2, and reduced emissions are directly associated with improvement of the energy conversion efficiency and hence reduction of fuel consumption. There are three major air pollutants other than CO_2, namely, nitric oxides (NOx), sulphur and particulate matters (PM), which are currently subjected to continuously tightening regulations. Although sulfur, which has been controlled since 2005 by the International Maritime Organization (IMO) [3], is no longer treated as an issue for ship-owners when low-sulphur fuel is used, reduction of PM and NOx emissions is related either to improvement of the combustion process or to the application of abatement technologies. Although the stringent regulations of IMO and Environmental Protection Agency (EPA) for NOx and PM (see Figure 1) apply globally and in coastal ECA's, designers of marine propulsion systems use Tier III (IMO) and Tier IV (EPA) requirements as some of the main guidances for their design concepts.

Technical solutions, such as selective catalytic reduction (SCR), dual-fuel (DF)/pure gas engines, exhaust gas recirculation (EGR), batteries/hybrids and fuel cell/hybrids, are often proposed to satisfy the NOx environmental regulations [4]. These technological pathways have their advantages and disadvantages, which deserve separate discussion. However, in context of lifetime cycle [5, 6] and the readiness of the marine engine market to adapt to environmental regulations, gas engines and the active use of EGR and SCR systems, at least in the near future, will be the top priorities of shipowners and ship designers.

The employment of an SCR system in combination with a diesel engine operated on marine gas oil can satisfy the requirements for NOx emissions; however, this symptomatic solution is considered to be limited due to the high cost of the reacting agent and operational is-

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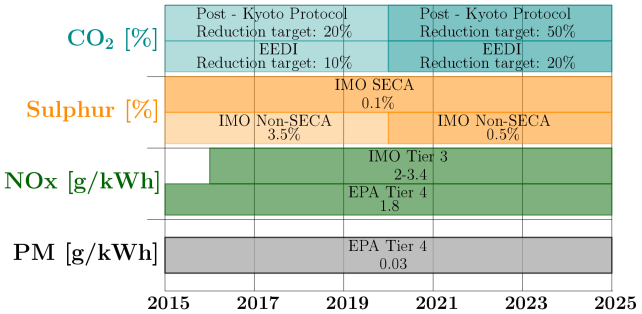

Fig. 1: Environmental regulations for emissions from ships (Adapted from Ohashi [7])

sues at low exhaust gas temperatures [8, 9]. Moreover, when shipping in the Arctic region, where strong attention is paid to emitted black carbon due to its effect on ice melting [10], SCR cannot be considered to be a panacea. Thus, the utilization of gas engines operating with relatively short carbon chain natural gas (NG), which reduces emission formation during the combustion process to prescribed standards, appears to the most promising alternative [11].

Another advantage of gas engines, along with potential emission reduction, is that the operation and maintenance of them and their auxiliaries, based on collected operational experience by DNV GL, is not more challenging than that of diesel engines [12]. In addition to the environmental benefits and the relative simplicity of the power unit maintenance, the utilization of NG is also lucrative in terms of fuel prices [13]. These and other arguments were used to predict marine fuel trends towards 2030, which forecast a significant increase in the NG fraction of fuels consumed by marine power plants [14]. This forecast was made despite the following challenges associated with the utilization of gaseous fuels in ship gas engines: increased fuel storage volume, additional regulations regarding the safety of operation and engine modification due to diesel engine retrofitting.

2 Marine gas engines

The development of marine gas engines originated in Wärtsilä (Sulzer) and brought benefits in the form of the first low-speed DF engine in the early 1970s [15]. More thorough gas engine development, based on Sulzer's experience, started in late 1980s in IHI (Japan) and MARINTEK (Norway), which led to the birth of the high-pressure dual-fuel (HPDF) engine concept. Later, in 1990s, researcher's focus shifted to low-pressure dual-fuel (LPDF) and lean-burn spark-ignited (LBSI) engines to comply with IMO NOx Tier II. Today, the market of gas engines for marine applications is composed of more than ten major manufacturers with a large variety of products that can roughly be divided into two main groups: pure gas engine and dual-fuel (DF) engines. The pure gas engines are lean-burn gas internal combustion engines (ICE) that operate on the Otto cycle, whereas DF engines offer flexibility in terms of used fuel and can implement either the diesel or Otto cycle, depending on the engine's configuration. The principal marine gas engine classification in shown in Figure 2.

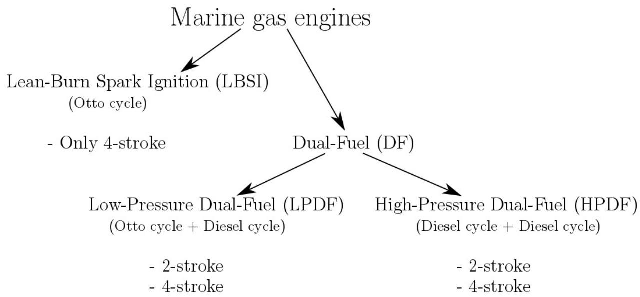

Fig. 2: Classification of marine gas engines

The choice of gas engine type for marine application is highly dependent on many factors, namely, type of vessel, its operational profile, both operational and capital costs, availability of bunkering infrastructure etc. However, in the present paper only combustion and emission aspects of the abovementioned types of marine gas engines are discussed in order to highlight challenges and gaps in research that aiming to improve combustion efficiency in marine gas engines.

Today the majority of producers report that modern LBSI and LPDF engines satisfy IMO NOx Tier III regulations without additional exhaust gas treatment [16–20], however a number of issues remain to be addressed:

- Unburned hydrocarbons (UHC) from lean-burn gas engines. As methane has a high GWP factor, its emission from ships should be minimized.
- Operational stability for LBSI and gas mode LPDF engines. The engine control system is not improved to avoid knock and misfiring events that can lead to engine damage and release unoxidised gaseous fuel to the atmosphere.
- Operation of lean-burn gas engines at low load [21].

When it comes to HPDF, the amounts of emitted NOx and PM are significantly higher than those from lean-burn engines [22].

3 Parametrization of combustion phenomenon in a marine gas engine

The combustion process in marine gas engine barely differs from that used in engines for cogeneration and the heavy truck industry. However, due to the difference in operation requirements imposed by DNV [23] and the abovementioned IMO regulations, the control of the combustion process in marine gas-powered units have unique features. The main factors influencing combustion and emission formation in marine gas engines are outlined in Figure 3. Although these features are similar to those in engines used for other applications, it is clear that some of these factors are unique to marine gas engines. For example, HPDF engines, which operate in a diesel cycle and use high-pressure gas jet combustion, have a number of particular factors because pure diffusion combustion of NG occurs in such an engine. Clearly, most of the defined factors not only direct influence the combustion process but also affect each other. In this study, to obtain a more intuitive understanding of the causal relationships, a simplified form of the diagram was selected. The authors have

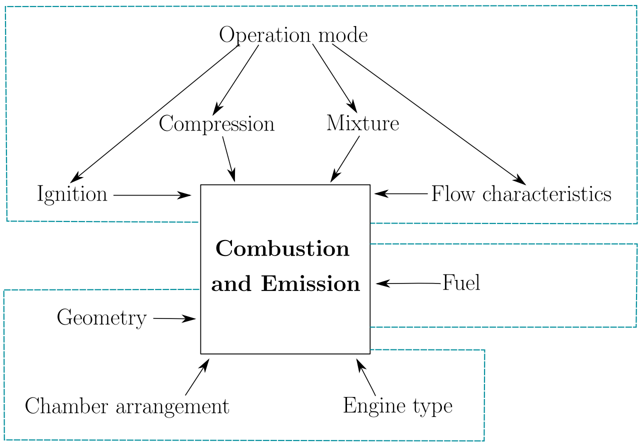

Fig. 3: Main parameters affecting combustion and emission formation in a marine gas engine

divided the abovementioned factors into three groups: design factors, operational factors and fuel factors. The design factor group includes the combustion chamber geometry and the arrangement features of an engine that cannot be varied during the operation process, whereas the operational parameters provide functional flexibility and are used in the active combustion control system of an engine.

Since the factors influencing the combustion process are unique for different types of engines and also vary from product to product for the same engine type depending on producer, the current review aims to provide as detailed context information about the findings as possible.

4 Design factors influencing combustion

This section is divided according to the marine gas engine concepts with emphasis on the particular *design factors*. Moreover, because the gas mode of a four-stroke DF engine differs from that of an LBSI only by ignition source and operational control, the design of some components are alike. Therefore, some information regarding these engines applies to both. Some NG combustion concepts that are tested and/or used in marine engines but cannot be directly attributed to the engines mentioned in the classification (Figure 2) are combined in a separate subsection called *Alternative combustion concepts*.

4.1 Lean-burn spark-ignited engines

Continuous research on the optimization of the combustion chamber geometry to improve in-cylinder process stability and efficiency resulted in two main chamber configurations for LBSI marine engines: open chamber and enriched prechamber. The open-chamber lay-

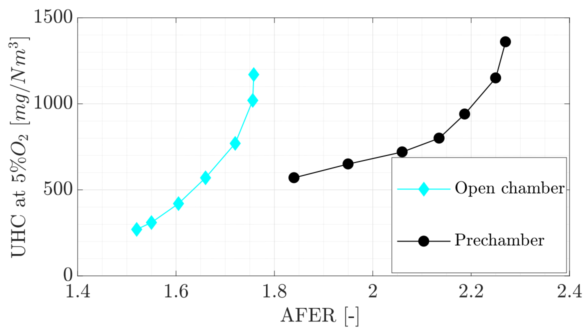

Fig. 4: UHC measured at the middle of combustion (MFB 50%) in an open chamber and prechamber engine at different AFER (Adapted from Hiltner et al. [24])

out (where the spark plug is located in the main engine chamber) provides a simple and cost-competitive chamber arrangement [25]. On the other hand, due to the increased NOx formation and slower transient load response compared to the prechamber type, the application of this concept in marine engines is limited. The greater complexity, from both a design and combustion perspective, of a cylinder with a prechamber ensures an extended lean limit of the air-fuel mixture. The experimental study of Hiltner et al. [24] on medium-speed

LBSI one-cylinder engines showed that with an enriched prechamber, the lean air-fuel equivalence ratio (AFER) limit could reach 2.2, whereas in a conventional open chamber, the AFER cannot exceed 1.78 (see Figure 4). This approach leads to reduced flame temperature and corresponding low NOx emission compared to that of the open-chamber arrangement. According to Zuo and Zhao [26], the prechamber intensifies the combustion process and maintains a high burning rate by jetted energy.

To ensure ignition and combustion stability in the entire engine's combustion chamber, the prechamber's air-fuel ratio is controlled separately and within an approximately stoichiometric value. Moreover, to promote rapid combustion and to reduce NOx formation in the prechamber, the turbulent intensity, which is dependent on the prechamber geometry, is kept high [27]. By contrast, a compromise between a high turbulence level and flame extinction should be maintained because a very high turbulence intensity can extinguish the flame [28].

Another important concern is NOx formation in the prechamber, which is considered to be a significant (could be greater than 10%) contributor to the total emitted nitric oxides [29], which is why it is important to keep the prechamber volume containing the enriched mixture as small as possible. Based on the previously mentioned requirements for combustion in a prechamber, Tozzi et al. [30] presented a prechamber design that provides an essential reduction in cycle-to-cycle combustion variability and decreased NOx emission with respect to a typical prechamber. One of the main reasons for the combustion stability improvement with the new prechamber design is the weakened effect of lubrication oil droplet auto-ignition on the combustion instability in gas engines with brake mean effective pressure (BMEP) greater than 18 bar [31, 32].

Another way to intensify the combustion in a prechamber to improve global engine performance is to properly arrange the prechamber injector and spark plug. Liyan et al. [33] studied the impact of different injection angles on the flow field and speed of combustion in the prechamber. The research was based on combined 2D PIV experiments in a stand-alone prechamber model of a marine LBSI engine with 3D CFD analysis, and the results revealed a trade-off between the strength of turbulence and mixture homogeneity, resulting in different NOx formation rates in the prechamber as the fuel injection angle was altered. Neither of the tested conditions produced symmetrical flame jet propagation into the main combustion chamber, which caused enhanced bulk gas temperature gradients.

As a virtual continuation of this research, Disch et al. [34] investigated the effect of prechamber hole dimensions on the flame jet penetration length and combustion intensity in a multi-cylinder medium-speed engine combustion chamber via OH^* chemiluminescence. The results showed that a reduction in the prechamber orifice diameter decreased the flame jet penetration length (see Figure 5), which was eventually reflected by a difference in the rate of combustion in the main chamber.

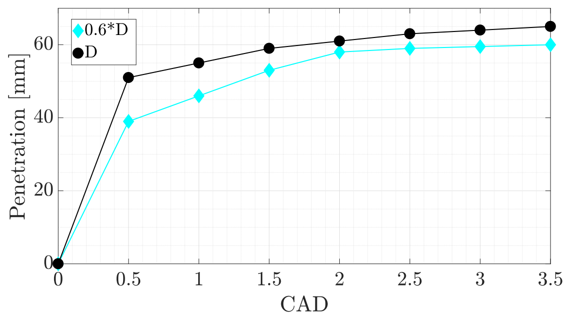

Fig. 5: Penetration length of the flame jet as a function of CAD (D is the initial diameter of the prechamber orifice) (Adapted from Disch et al. [34])

In addition to the prechamber design, the mixture properties in the main chamber also play a major role in combustion. The gas flow behavior and mixture formation were studied for different gas admission and intake valve seat ring configurations for a DF medium-speed engine by Waldenmaier et al. [35]. The mixture homogeneity in the combustion chamber and consequently the NOx formation were shown to be insensitive to changes in the arrangement of the intake system. However, for relatively rich air-fuel mixtures, the significance of the applied mixing device increased. Song et al. [36] and Yang et al. [37] studied the effect of gas nozzle design for LBSI engines on the gas mixture uniformity prior to combustion. The conclusion of this study based was that the utilization of a cross multi-hole gas nozzle is associated with less mixture stratification, higher combustion intensity and lower NOx emission than those of a single-hole gas nozzle.

Evaluation of combustion chamber design is also conducted from the perspective of total hydrocarbons (THC) emission. Comprehensive analysis of THC emission sources in NG engines emphasized the significance of the design factors on combustion completeness and UHC emission [24]. Improper piston design and crevices forming dead volumes and causing local flame quenching are considered to be the most crucial contributors to so-called methane slip [38]. According to Lee et al. [39], a 50% reduction in the top land height can increase the thermal efficiency by 0.3% and reduce methane slip by 21%. In addition to the modification shown in Figure 6, it is important to reduce the volumes on the top land and between the liner and cylinder head [25, 40, 41]. Stenersen and Thonstad [22] also included the gasket area between the cylinder head and cylinder liner, the volume behind the anti-polishing ring, the volume in the piston ring pack crevices and the crevices around the valve seats in a list of potential dead volumes.

In addition to a reduction in emitted THC, minimization of the top clearance (see Figure 6) results in intensified squished motion in engine clearance when the piston approaches the TDC. When it occurs during the compression stroke, the increased turbulence results in faster combustion and hence lower fuel consumption [42, 43]. Specially designed intake ports and intake valves can also contribute to in-chamber turbulence by creating swirl motion [43]; however, no study on marine gas engine applications has been published.

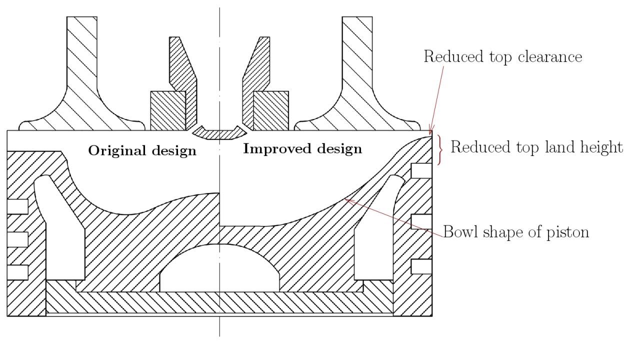

Fig. 6: Schematics of combustion chamber optimization to reduce flame quenching

4.2 Low-pressure dual-fuel engines

The work of Karim [44] is beneficial to the understanding of the fundamentals of LPDF engines. However, in this chapter, the authors focus on the design features of marine DF engines.

As in an LBSI engine, the operation of a DF engine in gas mode is conducted with a very lean air-fuel mixture, but instead of a spark plug, the cylinder charge ignition is provided by a flame generated from the combustion of a micro-pilot diesel injection. This approach overcomes the short lifetime of the spark plug and misfire issues due to insufficient ignition energy [45] and requires a corresponding combustion chamber arrangement. The combustion chamber layout varies greatly depending on the type of engine (two-stroke or four-stroke).

The prechamber-type arrangement is used in two-stroke LPDF engines, analogously to the prechamber-type LBSI. The choice of this solution is based on a comparative analysis of the operation and emission performance of prechamber-type and open-chamber two-stroke DF engines [41]. A reduction in NOx emission, increased knock margin, and improved thermal efficiency were observed when for the prechamber-type gas engine. In contrast to the LBSI, a DF engine is equipped with two diametrically placed prechambers because an adequately large exhaust valve required for sufficient uniflow scavenging is located in the center of the cylinder head, restricting the space for both the main and micro-pilot injectors [46]. Wärtsilä, as the only developer of this type of engine for marine applications, adapted their two-stroke diesel engine piston crown for DF engines. Despite the fact that the geometry of these pistons appears to be not be optimized with respect to UHC due to the relatively large top land height, the manufacturer claims reduced methane slip relative to that of LBSI engines due to the lower engine speed and longer time for oxidation [41].

The positions of the scavenging ports must be optimized in consideration of the two-stoke engine efficiency. The higher the scavenging ports in the engine's cylinder are, the lower the effective compression ratio and the higher the level of mixture stratification. The higher the position of the scavenging ports is, the less time for gas mixing will be available.

In addition to the compression ratio, Nylund and Ott [47] identified the following design parameters that have a strong impact on mixture homogeneity and methane slip: location of the gas admission ports, number of admission ports and gas admission nozzle geometry. The position of the gas ports is a compromise between the "too stratified" mixture obtained when the gas admission ports are located close to the cylinder head and "gaseous fuel escape" that occurs when the gas admission nozzles are placed near the scavenging ports. Therefore, the mid-stroke position of the gas admission nozzles is the default choice [48].

Hirose et al. [49] studied the feasibility of injection nozzle installation in the scavenging ports. The results showed improved cylinder charge homogeneity, which resulted in increased indicated mean effective pressure (IMEP) compared to that of the mid-stroke natural gas supply approach. Furthermore, the THC emission was relatively low for gas admission in the scavenging ports. Moreover, at high mean effective pressure (81%), the pilot diesel ignition was unstable, and pre-ignition occurred.

Although some attempts to apply the prechamber concept to four-stroke DF engines have been made [50], mainly the open-chamber approach is employed in the maritime sector. The main combustion chamber is equipped with at least one diesel injector that can both deliver a sufficient amount of fuel for full load in diesel mode and provide proper atomization as a pilot injector corresponding to $\approx 1\%$ of the total supplied chemical energy. This functionality is maintained by either using a well-designed single-needle injector [51, 52] or integrating two needles into one injection valve unit for different engine operational modes [53, 54]. By contrast, Kawase et al. [55] reported the use of two injectors, each for a specific purpose. The main injector, as a rule, is installed in the middle of the cylinder head to ensure a symmetric liquid spray distribution in diesel mode, leaving only the eccentric position available for the micro-pilot injector [56]. This injector placement requires tuning of the piston crown geometry in both engine modes and is considered to be one of the main challenges in the design of a medium-speed DF engine, along with the mutual arrangement of the injectors and the necessity of cylinder head permutation while retrofitting a pure diesel engine to a DF engine [56].

Similarly to LBSI engines, the piston crown of a four-stroke LPDF usually has a bowl form, but its concavity is balanced with respect to the different modes. Menage et al. [56] suggest a deeper bowl for operation with liquid fuel and a shallower bowl for gaseous fuels. The mutual arrangement of the injectors refers to placement of the injectors in such a way as to avoid difficulties in air-fuel mixing and hence to sustain high combustion efficiency when two injectors are in service. However, the operation of a pilot injector is required in diesel engine mode to provide the injector's cooling.

4.3 High-pressure dual-fuel engines

Another name for this concept is gas-diesel since the combustion of gas is conducted via a diesel cycle. For diffusion combustion similar to diesel fuel, the high density of the gas jet is facilitated by NG compressed to 250-350 bar. For gas ignition assistance, a so-called liquid spark (a flame from the pilot injection of high cetane number fuel) is required. Despite some economical challenges due to NG pressurization and high NOx and PM emission compared to fully premixed lean combustion [22], the HPDF approach allows a gas engine to be run analogously to the well-known conventional diesel engine with its various operational advantages [57]. Moreover, although the HPDF concept inherited the emission problems of diesel combustion, this concept attenuates two main issues associated with lean-burn combustion: knocking and methane slip. If the former can similarly occur in a conventional CI engine [58], the latter may occur due to failure of pilot fuel self-ignition, bulk flame quenching or piston wetting. In addition, HPDF engines operate with similar compression ratios and mean effective pressures as those of conventional diesel engines.

Currently, both two-stroke and four-stroke HPDF engines are available on the ship power plant market. Among the most well-known engines are auxiliary four-stroke Wärtsilä 32 and Wärtsilä 46, which are products of diesel engine retrofitting [59], and the two-stroke MAN Diesel & Turbo SE engine family. Modification of the combustion chamber of Wärtsilä engines was conducted by replacing the liquid fuel injector with an injector that can handle both diesel (100% energy and 4-5% of the total energy at pilot injection) fuel and high-pressure gas. The holes in this injector are established to guarantee sprays and jet overlap for gas mode and good mixing formation in diesel mode.

Unreported experiments performed by Sintef Ocean revealed a very long combustion duration of a gas jet compressed to 250 bar compared to that of diesel spray combustion in Wärtsilä 32 at >85% engine load. This result is clearly due to the difference in momentum of jets, the fluid compressibility, the injection duration and jet spreading in the combustion chamber. Increasing the injection pressure to 350 bar solved this issue. A similar trend was reported for two-stroke MAN Diesel & Turbo SE engines, where the heat release rates of diesel fuel and CNG follow the same pattern [60] at full engine load. These discrepancies highlight the effect of gas admission system design on the combustion efficiency.

Similar to four-stroke HPDF engines, the two-stroke MAN Diesel & Turbo SE units are retrofitted diesel engines where a gas admission system was additionally implemented without changing the combustion chamber shape [60]. In contrast to Wärtsilä power plants, the two-stroke engines are equipped with 2 pairs of injectors per cylinder: a diesel injector for different modes and a standing gas injection valve.

The effects of the introduced design variables, such as the gas injector geometry, on the combustion process were studied by Aesoy and Pedersen [61]. By using a 0D/1D model, the authors investigated the impacts of the cavity size and cross-section area in high-pressure gas injectors on the gas injection profile. A detailed description of the injector can be found in the report of MAN Diesel & Turbo SE [62].

4.4 Alternative combustion concepts

In contrast to LBSI engines, Niigata Power System developed a pure gas engine for marine application with pilot injection as the source of ignition [45, 63]. The replacement of a spark plug with a micro-pilot injector in the prechamber cavity overcame the known problems associated with relatively weak spark ignition but resulted in additional design and operational complexity. Ultimately, the producer could not completely avoid the use of a spark plug, which is employed to start the engine. From an operational perspective in marine applications, the engine is associated with extensive knocking and misfiring during transient load, which prevent the manufacturer from meeting the environmental requirements.

The hot surface of a glow plug was tested as an ignition aid for a CI direct-injection NG engine [64, 65]. The main conclusion was that surface temperatures greater than 930^{o}C are required to ignite the high-pressure gas jet. Moreover, the ignition stability is highly dependent on the gas composition and position of the glow plug with respect to the geometry of the fuel jet. The immaturity of this technology was illustrated by unstable gas jet ignition due to the relatively high engine cyclic variability. The short lifetime of the glow plug and the substantial energy loss through the hot surface further complicated this concept. In general, CI as an alternative approach that theoretically allows a reduction in emission formation and fuel consumption, has number of issues, including control at low loads, to be solved in practice.

5 Operational factors influencing combustion

This section is structured in the same manner as *Design factors*; however, because of the number of operational similarities in LBSI and LPDF engines, a separate subsection was created to highlight the commonalities of the combustion control alternatives of these engines.

5.1 Lean-burn spark-ignited engines

The operation of lean homogeneous charge gas engines at high BMEP is always challenging. Figure 7 shows that to achieve a stable combustion process that is free from knocking and misfiring, the gas engine must be precisely tuned across the global AFER of the mixture. Therefore, a major focus of engine designers is controlling the air and fuel mass fractions and the cylinder charge pressure and temperature conditions. In LBSI engines employing prechambers, the AFER adjustment is even more crucial.

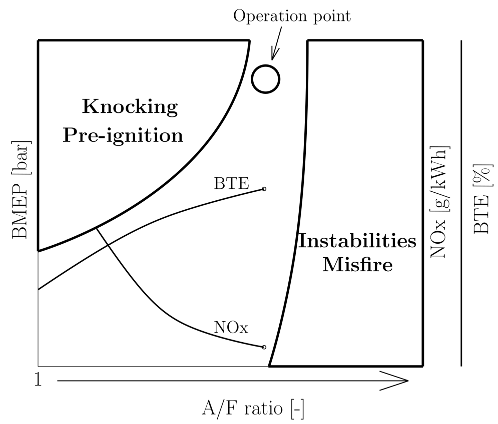

Fig. 7: Operation window for homogeneous charge LBSI engines

As mentioned earlier, the local AFER in the prechamber cavity is kept relatively enriched to achieve stable ignition. Simultaneously, it is important to reduce the "prechamber-main chamber" ignition delay, which can be achieved by slight tuning of the AFER in the prechamber clearance. Disch et al. [34] experimented with different AFERs in the prechamber and studied the flame jet area and flame penetration length by using OH^*-chemiluminescence. The results shown in Figure 8 reveal that the richer the mixture is, the longer the flame penetration length, decreasing the between-chambers ignition delay.

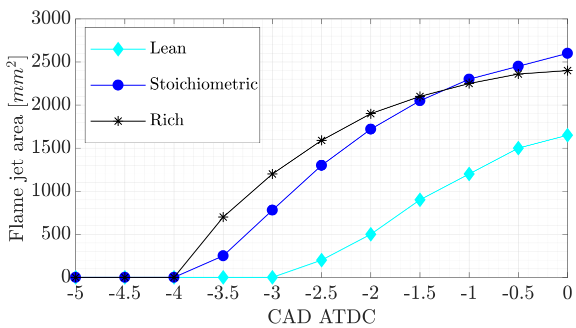

Fig. 8: Flame jet area for different air-fuel mixtures in the prechamber (Adapted from Disch et al. [34])

In addition to studying the effect of mixture enrichment, the same authors tested the influence of cylinder pressure on the main flame shape and the ignition delay by changing the spark-ignition timing. The results of the flame jet area across the crank angle degrees (CAD) are displayed in Figure 9. As expected, later ignition led to shorter flame penetration. Pure gas engines can

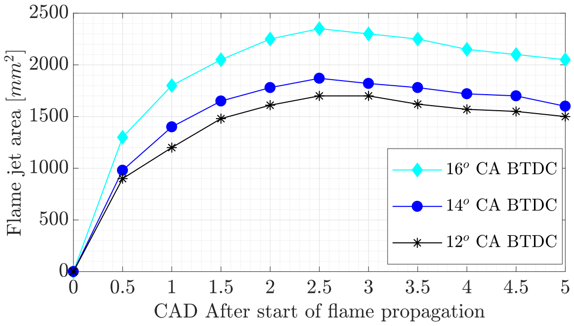

Fig. 9: Flame jet area with different ignition timing (Adapted from Disch et al. [34])

operate both as propeller drivers [66] or power generators [67]. However, historically, the utilization of gas engines was for land-based co-generation, and the first marine gas engines were operated as part of gas-electric power plant installations [68]. Therefore, the operation of the gas engines is conducted according to the IMO E2/D2 cycle, where the engine's speed is held constant. Figure 10 shows an example of how the main SI gas engine parameters vary when changing the loading conditions without exceeding the IMO NOx Tier III limit. The curves indicating BMEP and the maximum combustion pressure (MCP) behave according to general logic, where less fuel delivered leads to less energy withdrawn, both in the combustion chamber in the form of combustion pressure and on the crankshaft in the form of torque. The declining characteristics of brake thermal efficiency (BTE) state only that the engine has the most optimal conditions at high loads, where the highest combustion completeness is achieved. The lowest THC and the constant CO emission along the load range also indicate improved combustion completeness. The observed trend of reduced AFER at low loads is explained by the requirements to maintain high cycle-to-cycle stability.

5.2 Low-pressure dual fuel engines

Based on an understanding of the main idea behind the DF engine, one can imagine a number of parameters that might be varied during DF engine operation, where the parameters related to pilot injection and their impact on DF engine performance are considered to be the most important, in contrast to LBSI engines.

The effect of pilot injection quantity on NOx emission from heavy-duty marine engines was studied by a number of research groups [69–71]. The results show-

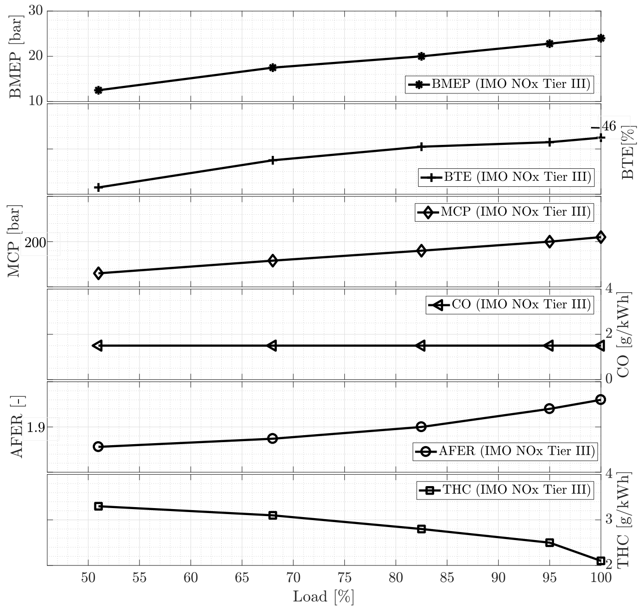

Fig. 10: Operation of an AVL LBSI engine in an E2/D2 generator configuration to comply with IMO NOx Tier III (MCP is the maximum cylinder pressure) (Adapted from Schlick [67])

ing direct proportionality between the amount of energy supplied by pilot injection and NOx emissions are in good agreement with those published by Karim and Burn [72](see Figure 11). On other hand, a reduction in pilot injection is not always the best solution in terms of engine performance. Tagai et al. [73] showed that there is a trade-off between BTE and NOx under varying quantities of diesel fuel in the pilot injection. In other words, under extreme minimization of pilot injection, the combustion speed slows, reducing the engine's efficiency.

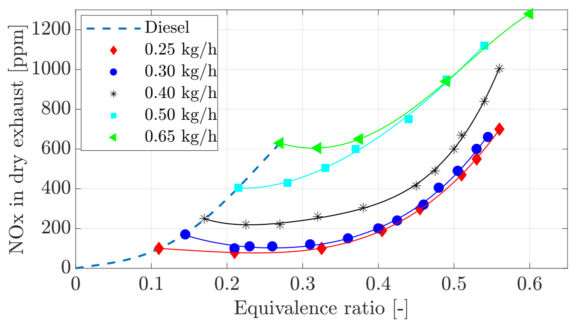

Fig. 11: Effect of pilot quantity on NOx emission in dual-fuel operation (Adapted from Karim and Burn [72])

When considering the effect of pilot injection timing and observing the trends shown in Figure 12, the conclusion is more unambiguous than that about the effects of pilot amount. Increased BTE and reduced NOx emissions can be achieved by advancing the injection timing. The only explanation behind the change in AFER at different CAD is the difference between the feed rates of engine air and NG.

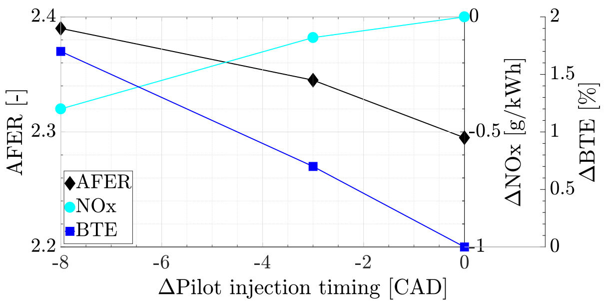

Fig. 12: Effect of pilot injection timing on NOx emission, AFER and BTE under partial loading conditions (Adapted from Tagai et al. [73])

Another important parameter defining the physical contribution to cylinder charge ignition rates and engine emissions is the injection pressure of pilot diesel. In this regard, the authors of this paper refer to the studies performed for pure diesel engines [74, 75] because no studies on marine DF power plants are available. However, the known trade-off between improved combustion characteristics and energy losses due to diesel fuel compression is considered to be universal and could be applied for any type of high-pressure injection system [76].

Referring to the statements in Schlick [67], DF installations are primarily used as a component of mechanical propeller-drive systems. Therefore, a DF engine runs in accordance with the E3 cycle, in which the engine output power follows the propeller characteristics. An example of such operation is shown in Figure 13. The engine was preliminarily tuned to emit less NOx than the maximum limit defined in IMO Tier III. Based on the indicated parameters, the most convenient operational conditions are in the range of 75-100% load (91-100% speed). Only the CO emission tends to increase with increasing engine power, whereas BMEP, MCP, and BTE appeared to have their highest values. With decreasing torque, the share of energy delivered by pilot injection increases at an almost constant global AFER because further reduction in the amount of pilot fuel is associated with delivery instability. As a result, during operation at low load, the AFER of the cylinder charge is higher than that of the full load. Methane slip, in the form of THC emission, also increases from 75% load, demonstrating the tendency of NG under-burning due to enhanced flame quenching. The correlation between the AFER of homogeneous charge and flame extinguishing is discussed in the next section since it is considered to be a common feature for LBSI and LPDF engines.

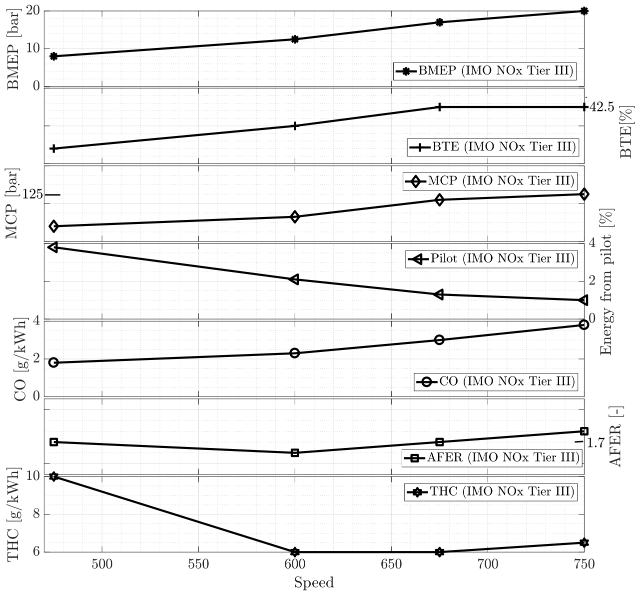

Fig. 13: Operation of an AVL DF engine in E3 propeller configuration to comply with IMO NOx Tier III (MCP is the maximum cylinder pressure) (Adapted from Schlick [67])

5.3 Commonalities of the operational factors of lean-burn engines

Both LBSI and LPDF engines operating under lean gas mixture conditions require advanced control systems, which are mainly applied to match the loading states with the cylinder charge conditioning. To a great extent, engine control is related to facilitating the mixture AFER, which in turn has a direct influence on methane slip. Figure 14 shows that an increase in homogeneous charge AFER leads to thickening of the quenching layer and hence methane slip. This correlation is behind the problematic methane emission at low load for LPDF engines (see Section 5.2).

In addition to the quenching layer, excess air in the cylinder charge plays an important role in combustion in crevices. Figure 15 shows that the probability of high THC emission is higher for rich gas blends than it is for leaner mixtures.

When virtually decomposing the mixture formation, the air supply and the fuel supply can be treated separately. Here, the control of fuel delivery is considered to

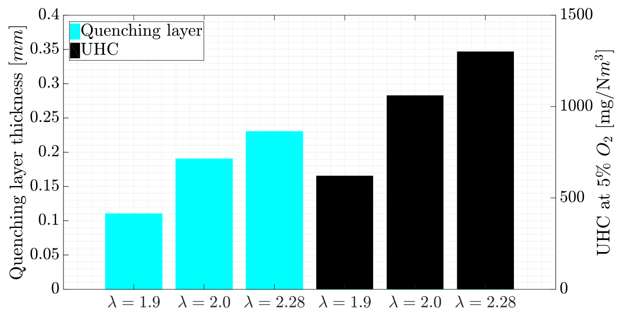

Fig. 14: Quenching layer thickness and UHC as a function of the AFER at TDC (Adapted from Hiltner et al. [24])

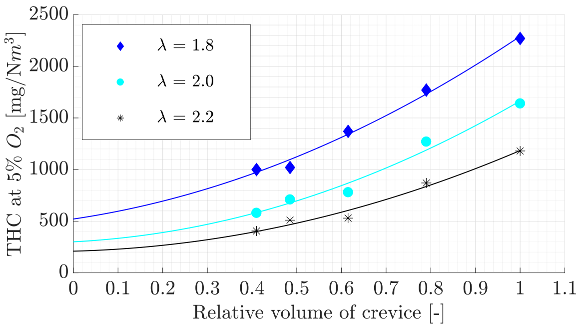

Fig. 15: Effect of the relative volume of crevices (with respect to the tall top land HCR piston) on THC for different AFER (Adapted from Hiltner et al. [24])

be no less challenging than that of air; however, few reports are available to draw clear conclusions. Christiner et al. [77] designated several requirements for port fuel injection: absence of leakage and precise mass delivery regardless of backpressure and speed. Advanced control of the injection timing, injection pressure and duration are applied to satisfy these requirements. In contrast to port injection systems, Mitsubishi S16 engines employ a charge mixer installed before the turbocharger. However, no reports are available to evaluate and compare this approach to conventional port fuel injection.

In terms of air admission and the air conditioning system, which usually involve a turbocharger and possibly and EGR system, the utilization of a variety of technological, sometimes complex, solutions is required.

5.3.1 Valve overlap

In a four-stroke diesel engine, valve overlap contributes to proper scavenging, but for port gas admission and homogeneous charge combustion of NG, this feature can be challenging. Driven by the difference in pressure between the intake air and exhaust manifolds, purging can lead to methane escape. Hiltner et al. [24] quantitatively proved that a positive difference between the pressure in the manifolds can triple the emission of UHC compared to that in a scenario with a negative pressure difference (see Figure 16). Therefore, valve overlap is eliminated for some gas engines (mainly for spark-ignited engines) under all conditions.

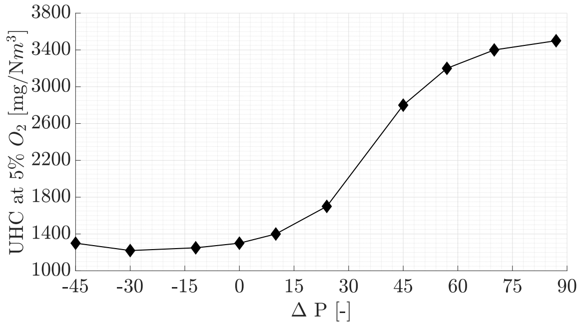

Fig. 16: Effect of a pressure difference (intake manifold and exhaust manifold) on UHC formation (Adapted from Hiltner et al. [24])

5.3.2 Throttle valve

The throttle valve, which is widely applied in relatively small lean-burn gas engines to control the AFER at low load by regulating the air charge pressures to lower than atmospheric pressure, is not commonly employed in marine gas engines due to the complexity of throttle control and the relatively high cost [78]. Engines equipped with a throttle valve are EYG26L [79] and Rolls-Royce B- and C-type power units. Figure 17 quantitatively illustrates the positive effect of throttle valve regulation on mixture AFER control, which results in reduced THC emission and fuel consumption.

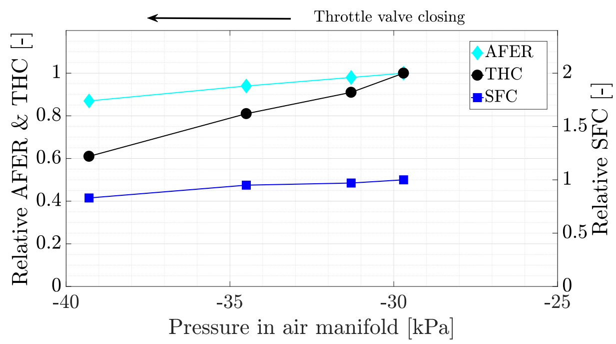

Fig. 17: Influence of throttle valve closing on AFER, THC and specific fuel consumption (SFC) at low engine load (Adapted from Hagiwara and Ohashi [79])

To ensure AFER control at high engine load, when the throttle valve stands fully opened, this engine is equipped with an air-bypass valve (ABV).

5.3.3 Boost pressure control

In addition to the ABV as a tool to control the boost pressure and consequently AFER at high engine loads [80, 81], some marine lean-burn engines are equipped with a turbocharger with variable turbine geometry (VTG) or an exhaust wastegate valve (EWV). The variable effective aspect ratio of the turbocharger provides flexibility to set the required pressure conditions in the intake port [82]. The functionality of the EWV is similar to that of the ABV, but instead of bypassing the compressor, the valve bypasses the turbine [83]. The importance of controlling the boost pressure was studied by Järvi [84] and is shown in Figure 18, where a slight change in boost pressure produces bias towards a significant increase or reduction in NOx and THC.

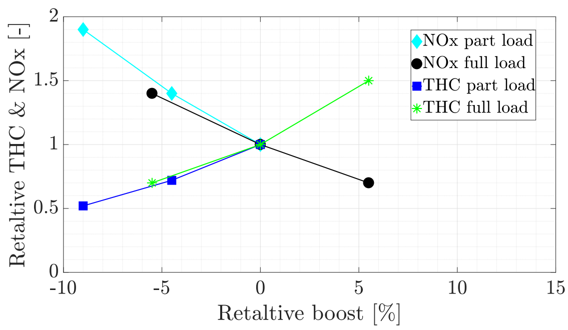

Fig. 18: Effect of a change in relative boost pressure on THC and NOx emission (Adapted from Järvi [84])

5.3.4 Blow off valve

A blow off valve (BOV) allows the air to drain from high-pressure air to high-pressure exhaust line. This valve helps to overcome the problem of high exhaust gas temperature at turbine low expansion ratios [84]. Without the BOV, the engine might produce increased THC emission due to excessively high AFER, as shown in Figure 19. In turn, to control air mass flow at high engine loads, the engine might additionally include one of the systems described in subsection 5.3.3[85, 84].

5.3.5 Charge air preheater vs intercooler

The charge air heater was considered in Ritscher and Greve [86]. Although there is no information about the advantages of using such a device, the authors assumed

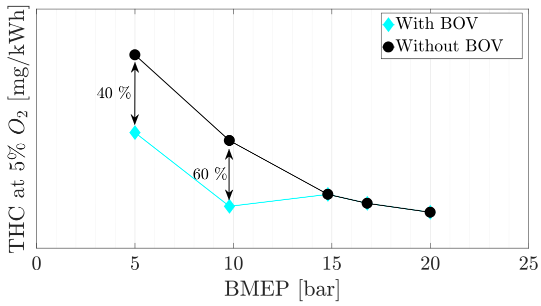

Fig. 19: Influence of BOV on THC emission from a DF engine operating at constant speed (Adapted from Järvi [84])

that operation of the air preheater is limited to engine start. In general, the operation of an engine with an excessively high air temperature in intake port might, as discussed earlier, lead to combustion instability and even knocking. Therefore, air intercoolers are installed before the intake valve to control the charge pressure and temperature conditions. Additionally, as shown in Figure 20, excessively cold air can result in heightened THC emission.

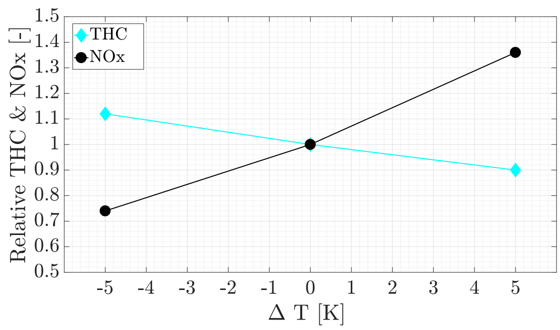

Fig. 20: Effect of air temperature after the intercooler on NOx and THC emission from lean-burn medium-speed engines (Adapted from Järvi [84])

5.3.6 Variable valve timing and compression ratio

Another widely applied system that enables regulation of charge properties is variable valve timing (VVT) [87]. Adjustment of the inlet and exhaust valve opening phases (in a four-stroke engine), regardless of the camshaft position, enables both control of the amount of air entering the cylinder and regulation of the air-fuel mixture pressure and temperature conditions at different engine loads. So-called Miller timing, which reduces the effective compression ratio by early or late inlet valve closing, is the most widely known VVT system [88]. The impact of the compression ratio on combustion stability is shown in Figure 21. Miyai et al. [89] numerically showed how an effective compression ratio could be tuned in conjunction with Miller timing to achieve a high level of BTE under different loading conditions[89].

Järvi [84] also studied the influence of the compression ratio on THC emission and concluded that an increase in the compression ratio increases the emission of unburned hydrocarbons. To compensate for the air charge shortage at the end of compression stroke due to Miller timing, the engine is usually equipped with a turbocharger. Depending on the engine's BMEP and the Miller strategy, a one- or two-stage turbocharger is used. Two-stage turbocharger are required to ensure high load sustainability and to achieve the required NOx emission reduction under strong Miller strategies and BMEP 21 [90].

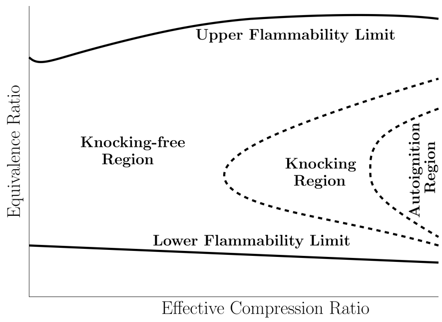

Fig. 21: Dependence of combustion stability on the fuel equivalence ratio and compression ratio[44]

In a two-stroke DF engine, the VVT system originally designed for two-stroke diesel ICE is used. Here, the advanced exhaust valve closing (EVC) ensures internal EGR, decreasing peak flame temperature, and adjusts scavenging at low-load conditions[91].

5.3.7 Exhaust gas recirculation

The increase in the cylinder charge heat capacity to reduce the flame temperature and NOx formation is the main reason to dilute the charge air with exhaust gas. As in any other ICE, the EGR system could be internal or external. In internal EGR, the products of the reaction are retained in the combustion chamber or returned through the exhaust valve [80]. Although this approach is the easiest, in terms of the required facilities, the internal EGR demands readjustment of the valve timing, which could be challenging if the VVT is already tuned to achieve high BTE. Poor cylinder scavenging could be achieved by advanced closure of the exhaust valve, similar to that in Millo et al. [92], where the authors studied the effect of VVT on marine diesel engine performance.

In general, the application of the EGR system is not mandatory if NOx emission is below the regulation limits. However, according to Järvi [84], an external low-pressure EGR can reduce methane slip. An EGR system composed of an EGR cooler and EGR valve between the exhaust gas after the turbine and an air line before the compressor reduced THC on 50 mg/Nm^3 by increasing EGR to approximately 14%. In addition, Nitta et al. [93] proposed a new combined EGR (C-EGR) capable of simultaneously reducing both methane slip and NOx emission from engines. The development of different EGR configurations for land-based gas engines[94, 95] could be an inspiration for future improvement of marine gas installations to achieve emission goals.

5.3.8 Knock and misfire control

Despite the fact that when running a gas engine one is theoretically aware of the required operation conditions to avoid undesirable events, fast load transitions, changing gas composition and deterioration of cylinder charge conditioning might result in misfire, knocking or pre-ignition. To return the engine to smooth operation, some control measures are required [96, 97]. The most common way to avoid unwanted events is to use advanced or retarded ignition timing. Therefore, spark ignition [98] or pilot injection timing [53] must be adjusted accordingly.

A typical control system measures combustion inconsistencies, compares them with predefined thresholds and executes a compensating act if necessary. Modern knocking-detection systems are abundant [99], but in the marine industry two main approaches are applied: vibration + exhaust gas temperature measurement and measurement of the combustion pressure. For example, in the DF engine of Wärtsilä, both systems are used; however, measurements from the accelerometer and exhaust gas thermometer are considered as a back-up [97]. A pressure sensor installed in the combustion chamber could be considered to be an additional source of UHC (referring to Section 4.1).

Sometimes, successful achievement of knock-free operation in one cylinder can deteriorate combustion processes in other cylinders, which can occur when controlling the boost pressure and air charge temperature in the intake manifold. The research of Servetto et al. [100] showed that some cylinders could be exposed to more favorable boundary conditions for knocking than others due to pressure oscillations in the air manifold. To improve the knocking resistance in each cylinder of a large-bore DF engine, an internal quarter-wave Ushaped bent resonator pipe was introduced in the air manifold cavity. The resonator decreased pressure pulsation by 40%; however, no direct quantification of the resonator effects on knocking phenomenon and engine operational stability was reported.

5.3.9 De-rating

In a situation where knocking cannot be avoided, the engine load is reduced to achieve better operational stability (see Figure 23) [46]. Engine de-rating is considered to be a last resort treatment since marine power supply customers could directly suffer. For example, a reduction in propulsion power can occur when de-rating a ship's main engine.

5.3.10 Skip firing

Without a throttle valve, the operation of a gas engine becomes challenging at low load due to the potential incomplete combustion associated with increased THC, CO and SFC. At low engine load, a lean-burn engine can be operated with relatively high AFER without knocking. Simultaneously, one might face increased NOx emission and misfiring (DF engines). To overcome these issues, lean-burn engines employ a so-called skip firing algorithm[78]. This approach allows deactivation of gas admission to some cylinders while other operating cylinders have sufficiently low AFER to maintain stable combustion. Skip firing provides a significant reduction in THC emission and methane slip [84]. Cylinder deactivation is performed according to a preliminarily developed procedure that ensures the required cylinder balancing [67].

5.4 High-pressure dual-fuel engines

This section is mainly dedicated to the operation of two-stroke engines due to a shortage of information about four-stroke Wärtsilä 32 and 46 engines. According to an unpublished report from Sintef Ocean, the supercharged four-stroke HPDF engine operates in generator mode without the abovementioned mechanism additionally controlling global AFERs.

In general, the introduction of additional high-pressure gas injection valves extends the list of variables to control during HPDF engine operation. Injection timing for both fuels (diesel pilot and gas jet) and the injection pressure are among the important parameters to consider. The main criterion for selecting the correct injection timing of the pilot injection is to guarantee its self-ignition and to further provide fast combustion of the gas jet.

Gas injection control in this context is more sensitive and requires a more comprehensive approach [101]. Due to the characteristics of compressible fluid flow and taking into account the fact that the engine compression pressure and gas injection pressure have the same magnitude, it is important to consider the potential mass flow alteration when changing the gas injection timing. By following the same logic regarding emissions as that in an LPDF, the amount of oil delivered by the micro-pilot injector should be kept as low as possible. MAN Diesel & Turbo SE managed to reduce the quantity for the liquid spark to only 5% of the total fuel index for a 100% engine load[102]. Moreover, the relative energy delivered with oil fuel increases linearly with decreasing engine load.

Injection profiling and other engine settings might be reconsidered to optimize the engine for either fuel economy improvement or NOx emission reduction. A good example of the optimization of an HPDF engine operating in gas mode is shown in Figure 22. Here, a 75% load is used as a reference for fuel economy improvement. This example shows that after engine tuning, the reference load is characterized by a higher, higher in-cylinder pressure at the end of compression (CP) and higher NOx formed compared to the values before tuning. These changes correlate with the list of approaches that Juliussen et al. [60] reported could be applied for engine readjustment:

- Earlier exhaust valve closing. This process helps to increase the effective compression ratio, which can lead to higher temperature and pressure at the end of compression, both of which promote pilot fuel self-ignition and faster and complete combustion. On the other hand, the increased compression temperature results in increased flame temperature and therefore increased NOx emission.
- Turbocharger settings. One of the alternatives of VTG, EWV or ABV was used to maintain constant scavenging pressure while increasing flame temperature (and hence exhaust temperature) in the course of the experiments. Considering the techniques used by this manufacturer to control air charge in two-stroke diesel engines [103], the solution with variable nozzle appears to be the most probable.

Additionally, the insignificant methane slip for all loading ranges confirms the statements about the advantages of this gas engine concept. On other hand, due to the prevailing mixing-controlled high-temperature combustion, the NOx emission naturally exceeds IMO Tier III. Thus, the introduction of additional measures to comply with the regulations is required. The success of the application of external EGR for the same type of non-retrofitted diesel engine to reduce emitted nitric oxides to approximately 4 g/kWh at full engine load illustrates the potential of this approach [103]. At the same time, high dilution of the air charge with exhaust gases could be required under partial load conditions to compensate for the increasing trend of the NOx curve (Figure 22). Despite the fact that MAN Diesel &

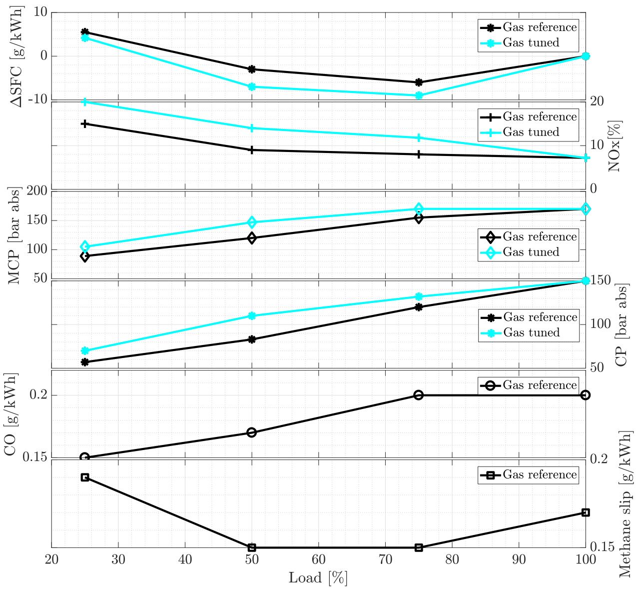

Fig. 22: Engine performance and emission characteristics of a two-stroke engine operated in gas mode. Gas reference represents the initial settings of the engine, while Gas tuned stands for operation with reduced specific fuel consumption. CP is the pressure at the end of compression (Adapted from Juliussen et al. [60])

Turbo SE refers to internal EGR as a solution for NOx emission reduction in two-stroke engines [104], the poor scavenging, for example, by throttle in the exhaust pipe, could substantially complicate the combustion control. Moreover, the remaining UHC, in the form of particles, in the combustion chamber can form deposits. In addition, the deposits are associated with higher risk in DF engines than that in pure diesel engines due to the number of installed injectors.

6 Fuel factor

The chemical composition of a fuel strongly affects its performance and emission from an internal combustion engine. AEsoy et al. [68] discussed the influence of the gas composition and methane number (MN) as a reference characteristic of NG utilized in lean-burn gas engines. The MN is defined as follows: MN is 100 for pure methane gas fuel. The inclusion of hydrocarbons with more than 1 atom of carbon (ethane, propane, etc.) or hydrogen in a methane-dominant gas mixture reduces the MN, while dilution of methane with inert carbon dioxide (as in biogas) or nitrogen is associated with increased MN (could be 100 and over). Based on the work of Portin [105] who overviewed NG compositions in major export terminals and showed that in some locations the MN could be less than 70, it was concluded that the operational window for a lean-burn engine narrowed due to an increasing knocking zone (see Figure 23). This result is due to the lower auto-ignition temperatures of ethane, propane and butane than that of methane [106]. At the same time, the misfire conditions could be not fully defined by the MN, since there is no correlation between the lower flammability limit of a gas mixture and its MN. As an example, the addition of hydrogen would shift the misfire zone to higher AFER, and the presence of paraffins with C⩾2 would lead to an even narrower "normal operation" zone at relatively high BMEP.

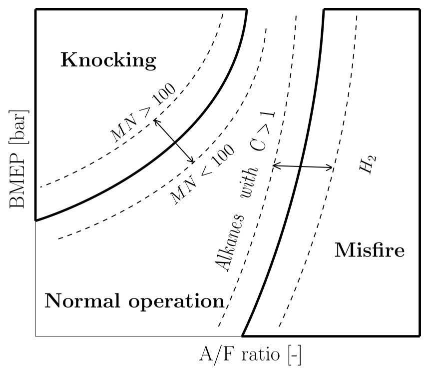

Fig. 23: Effect of the MN and gas mixture composition on the operation window of lean-burn gas engines

Lauer et al. [107] estimated how gas engines could be readjusted and how much thermal efficiency would be lost when an engine is operated with a gas with lower MN but still complying with IMO NOx Tier III regulations. Figures 24 and 25 show that engine readjustment could be executed by decreasing the effective compression ratio, decreasing the air mass fraction in the cylinder charge and retarding the ignition timing on 10^o when the MN is reduced. The relatively high thermal efficiency was maintained regardless of the fuel's MN by maintaining a reduced effective compression ratio (advanced inlet valve closing), that in turn reduces likelihood of knocking.

The effect of the gas composition on knocking was also determined for lean-burn gas engines by Ichikawa et al. [108]. The results of the experiments revealed that the MN is both an excellent indicator of gaseous fuel knocking resistance and a crucial parameter characterizing knocking intensity, as measured by the "cylin-

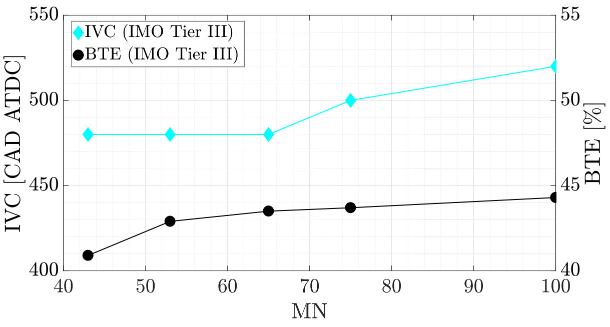

Fig. 24: Effect of MN (adding propane) on the inlet valve closing (IVC) time and the BTE of gas engines to meet IMO NOx Tier III (Adapted from Lauer et al. [107])

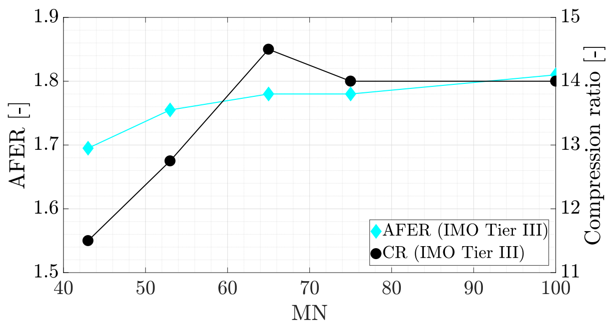

Fig. 25: Effect of MN (adding propane) on the compression ratio and AFER of gas engines to meet IMO NOx Tier III (Adapted from Lauer et al. [107])

der pressure" knock-detection method [99]. Proposed by Karim [109], the relation for estimation knocking conditions shown in Equation 1 takes into account both the operational conditions of the engine, including the air intake temperature (T_o), power output characteristics (A) and operating mode, and the ignition properties of the applied fuel (B).

$$log[power]_{knock} = A + \frac{B}{T_o} \tag{1}$$

The parameters A and B are not related to the octane number for operation on hydrogen [110]. The author presumed that this is due to the difference between the knock nature of hydrocarbon fuels and hydrogen, where combustion is associated with very fast flame propagation.

Previous research shows that one of the most crucial challenges in the operation of lean-burn engines is incomplete combustion, which leads to the emission of methane. In this context, the dilution of pure methane with different alkanes, hydrogen or inerts reduces the quenching distance [111], thereby positively affecting combustion completeness. Another advantage of including higher alkanes is the more favorable ignition properties in terms of the minimum ignition energy (MIE). Blanc et al. [112] experimentally estimated that the MIE of methane is shifted towards higher equivalence ratios, while higher alkanes require lower minimum ignition energy at lower fuel-equivalence ratios. Therefore, the presence of these paraffins will likely improve the ignition properties of lean premixed charge and could be beneficial when, for example, spark plug deterioration occurs.

The addition of hydrogen to the fuel in the context of MIE, flammability limits, quenching distance, burning rate and carbon molecule absence is a well-known "tool" to improve the combustion process and to reduce carbon-based emissions [113–116]. Despite the increased adiabatic flame temperature due to the addition of hydrogen and the associated increased NOx formation, some attention has been given to the lightest molecule even in marine applications [117, 118]. In the study of Portin et al. [117], the conclusion was that an increase in the hydrogen volume fraction requires a reduction in the compression ratio to avoid possible knocking. For example, to increase the volume fraction by 12% (from 28 to 40), the inlet valve closing was readjusted to shift the compression ratio by 1 (from 11 to 10). Moreover, because of different temperatures and pressures of the charge, the engine's air supply system was adequately tuned.

In contrast to hydrogen, biogas has very similar physical properties to NG and can also be considered to be a renewable fuel for marine operations. Bengtsson et al. [119] assessed different fuels for short sea shipping and found that biogas might be a local alternative shipping. Noting the presence of carbon dioxide in the biogas, despite the reduced calorific value, some benefits, including NOx emission formation, could be obtained. Nevertheless, an engine designed to operate on fuel with MN 50-100 must be retuned by AFER and ignition timing since the operation region for a lean-burn gas engine will be shifted towards lower AFER.

MAN Diesel & Turbo SE studied liquefied propane (+ butane) as a fuel for their HPDF engine concept. Being aware of the difference between the lower heating values, densities and stoichiometry of NG and propane (butane), the fuel shift to propane requires additional fuel pressurization (to 550 bar instead of 250-350 bar for NG) and retuning of air admission to ensure complete fuel oxidation under the relatively increased mixture fraction gradients. Experiments conducted with HFO as a pilot fuel revealed, foremost, that liquefied propane could be considered to be an alternative for NG and secondly, that HPDF with minor modifications (e.g., gas injection system) can tolerate a variety of gaseous fuels without significant increases in emission formation

[120]. Moreover, the insensitivity of HPDF engines to different gaseous fuels is highlighted by the fact that some MAN Diesel & Turbo SE products are used to burn volatile organic compounds (boil off gases) withdrawn from LNG tanks [121, 122]. However, some engine operation adjustments are required since different gas compositions have different ignition properties [123], eventually affecting engine performance and efficiency.

7 Comparison of the concepts

The combination of the abovementioned tools helps to finely tune lean-burn gas engines and to improve the combustion process and increase total efficiency. However, optimization is usually applied only for high loads, leaving other operational conditions relatively inefficient. The comparison of SFC for HPDF and lean-burn gas engines shown on Figure 26 reveals that the diesel-like operation of HPDF has significantly lower fuel consumption at low loads, whereas when approaching 100% load, the lean-burn engines demonstrate their superiority. The same picture could be observed by revising maximum indicated thermal efficiencies for different marine gas engine concepts (see Figure 27). The Miller strategy, which includes valve timing and the utilization of a two-stage turbocharger, together with increasing the geometrical compression ratio enabled significant improvement in the thermal efficiencies of prechamber-type lean-burn engines [67]. Against this background, the efficiency of the unoptimized HPDF engine has room for improvement.

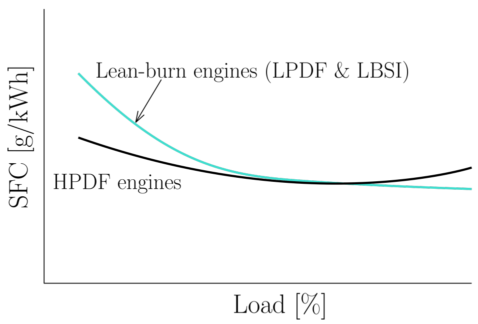

Fig. 26: Specific fuel consumption of gas-diesel and lean-burn engines under different loading conditions[68]

When viewed from the perspective of emissions, although HPDF almost completely avoid methane slip, the diffusion-controlled combustion leads to incomplete oxidation of fuel and hence PM formation. In Table 1, one can see that both the NOx and PM for HPDF are significantly higher than those of lean-burn engines.

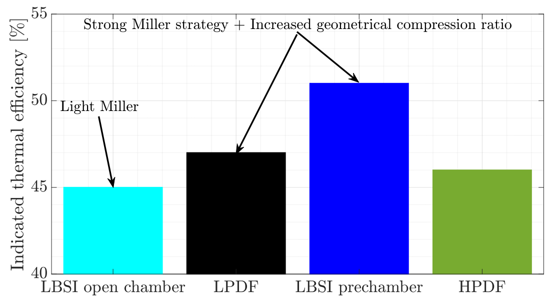

Fig. 27: Comparison of a marine gas engine's indicated thermal efficiencies (Data for lean-burn engines was taken from Schlick [67])

Table 1: Emission reduction from marine gas engines relative to a conventional engine operated with diesel fuel in IMO E2/E3 test modes without exhaust gas aftertreatment. In DF engines, marine diesel oil was used for pilot injections [22].

Emission parameter	LBSI	LPDF	HPDF
CO_2	25-28%	20-26%	20-26%
NOx	85-90%	75-90%	25-30%
PM	>99%	95-98%	30-40%(4- stroke)
SOx	>99%	95-98%	95-97%

8 Conclusion

- Lean-burn combustion is a proven approach for the utilization of NG on vessels and allows significant reduction in CO_2, NOx and PM emissions. To maintain fast combustion and avoid dead volumes, which result in the emission of unburned hydrocarbons, the combustion chamber geometry must be properly designed. It is highly important to thoroughly arrange the gas admission system, especially when it comes to the necessity of a homogeneous air-fuel mixture.
- Due to the narrow region of stable operation at high BMEP in lean-burn engines, an advanced control system that can either readjust the engine operational conditions or lower the load is required. Substantial attention should be given to engine operational scenarios, which are to be considered in control settings of the engine and its outfitting.
- HPDF engines, which have diesel-like operation, are noticeably easier than lean-burn engines in terms of the control system. However, not much work has been made towards increasing the total efficiency by optimizing the combustion chamber design and the combustion event itself, which are considered to be important research areas.
- HPDF engines are more flexible to changes in fuel properties compared with other concepts, bringing additional ease of operation. In contrast to homoge-

neous charge combustion marine gas engines, HPDF engines are characterized by relatively high NOx emissions above the IMO Tier III limits. Therefore, to comply with the regulations, the application of additional countermeasures, for example, EGR, will be required.

Acknowledgements

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Abbreviations

ABV Air-bypass valve
AFER Air-fuel equivalence ratio
BMEP Brake mean effective pressure
BOV Blow off valve
BTE Brake Thermal Efficiency
CAD Crank angle degrees
DF Dual-fuel
EGR Exhaust gas recirculation
EPA Environmental Protection Agency
EWV Exhaust wastegate valve
ICE Internal combustion engine
IMO International Maritime Organization
LBSI Lean-burn spark-ignited
LPDF Low-pressure dual-fuel
MCP Maximum combustion pressure
MN Methane number
NOx Nitric oxides
PM Particulate matters
SCR Selective catalytic reduction
SFC Specific Fuel Consumption
THC Total hydrocarbons
UHC Unburned hydrocarbons
VTG Variable turbine geometry
VVT Variable valve timing

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