Effects of injection strategies on low-speed marine engines using the dual fuel of high-pressure direct-injection natural gas and diesel

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Abstract
Pilot-ignited high-pressure direct-injection (HPDI) natural gas engines have drawn much attention since being proposed, as these engines can maintain high performance and lead to clean combustion compared with conventional diesel engines. This prospective concept is not only used in vehicle engines but also used in large-scale marine engines. In this study, the effects of injection strategies of dual fuels on combustion characteristics and emissions were investigated using the computational fluid dynamics (CFD) code CONVERGE coupled with a reduced n-heptane/methane mechanism. The results showed that the effects of absolute injection timing (AIT) on the combustion characteristics and emissions are similar to those of traditional diesel engines. As the AIT varies from advanced by 4°CA to retarded by 4°CA, the peak values of the in-cylinder pressure and temperature decrease gradually. However, a conventional trade-off between NOx and soot/CO/HC/ISFC emissions can also be observed as the AIT is retarded. In contrast, relative injection timing (RIT) has more complicated impacts on the combustion and emissions characteristics. The in-cylinder temperature distribution and concentration stratification change as the gas jet initially interacts with the pilot flame. The effects of the gas injection timing on the combustion characteristics and indicated specific fuel consumption are also similar to those of the AIT, but the emissions are quite different. The pilot injection timing shows minimal influence on the main combustion process. Better performance can be obtained with partially premixed combustion with a slightly late injection timing compared with that of the original (ORG) base case.

Keywords
dual fuel, injection strategies, low-speed, marine engine, natural gas
1 INTRODUCTION

Seagoing vessels, which are mainly cargo carriers, can cause substantial emissions. International regulations from the International Maritime Organization (IMO) restricting nitrogen oxide (NOx), sulfur oxide (SOx), particulate matter (PM), and volatile organic compound (VOC) emissions from new and existing ships have been implemented by maritime states as statutory requirements since 1978. In 2016, IMO Tier III has come into force. It affects numerous countries and shipping companies and has been considered by manufacturers including MAN Diesel & Turbo and Wärtsilä. Tier III limits NOx emissions to 3.4 g/kWh in emission control areas, requiring an approximately 75% reduction compared with the 14.4 g/kWh stipulated in Tier II. New regulations also introduced drastic reductions in fuel sulfur content, allowing 0.1% sulfur in marine fuel since 2015. Many methods for the reduction of marine emissions have been investigated globally. Additionally, the transition from using crude oil to other fuel sources due to both political and economic reasons has also been extensively explored.

Natural gas (NG), mainly consisting of methane, has been viewed as a promising clean alternative fuel because of its abundant resources; excellent properties including high thermal efficiency; low pollutant emissions, especially PM; and lower greenhouse gas (GHG) production due to the low carbon/hydrogen ratio of methane.

Different types of gas engines have been summarized in the literature. These engines, categorized mainly by two different ignition principles, operate primarily either by diffusion or by premixed combustion. When burned with diffusion combustion, a pilot of diesel fuel is usually injected into the hot air slightly prior to the injection of natural gas and serves as an ignition source. Natural gas is directly injected into the cylinder near the top dead center (TDC), and this injection requires a high gas injection pressure, that is, 250-300 bar. In such cases, the gaseous fuel is ignited upon its interaction with the pilot fuel flame. As a result, the energy released by the pilot diesel fuel serves only as an ignition source. The in-cylinder combustion is primarily controlled by the mixing process of gaseous fuel with air. Detailed information about this dual-fuel engine ignition and combustion process can be found in the literature.

Pilot-ignited high-pressure direct-injection (HPDI) natural gas engines have been recognized as an attractive concept for their potentially high thermal efficiency compared with diesel engines with lower knock risk and emissions. Many researchers have studied HPDI combustion-type gas engines. Miyake et al studied the gas jet velocity field in the cylinder on a large-bore (420 mm) single-cylinder engine. The gas was injected at approximately 250 bar with various injection configurations and timings. When the pilot diesel fuel was injected through a separate diesel injection system, the thermal efficiency at 85% load with a 5% pilot fuel energy input (diesel) could match that of 100% diesel operation. Additionally, a gaseous fuel injection nozzle with integral injection of the diesel pilot fuel could also provide satisfactory performance. The results showed that almost equal overall performance could be obtained with this injection system compared with that of the 100% diesel operation. Subsequently, Miyake et al and Biwa et al reported the development of an injector for both natural gas and diesel pilot fuel in which the hydraulic actuation for the gas injection was isolated from the gas by supplied sealing oil (300 bar). The experimental results showed that almost the same thermal efficiency, power, and reliability could be obtained compared with those of the conventional diesel engine, even with the reduced methane content (diluted to 80% by nitrogen) in the natural gas. MAN Diesel & Turbo converted a 4T50ME-X test engine to a gas engine. With separate injectors for two kinds of fuel, high-pressure (300 bar in MCR [maximum continuous rating], related to output power) natural gas was injected into the combustion chamber of the four-cylinder, blower-scavenged, two-stroke diesel engine (500 mm bore). Various tests have been conducted for practical applications. Mean effective pressure at a level of 2 MPa has been reached with above 95% heat released by natural gas. Emissions including NOx and CO2 were improved by 23% and 20%, respectively, and the soot emissions remained almost zero. Fundamental studies on a large rapid compression expansion machine (RCEM) have been conducted by Imhof et al and Ishibashi et al to improve the thermal efficiency and reduce the NOx emissions. An observation view of 200 mm in width and 50 mm in height was applied to analyze HPDI combustion. The results showed that the diffusive combustion speed in HPDI combustion was basically dependent on the air entrainment rate into the gas jet, and this effect is similar to that of diesel spray combustion. As a result, increasing the gas injection pressure effectively accelerated diffusion combustion. In addition, NOx emissions could be drastically reduced as exhaust gas recirculation (EGR) was applied by reducing the oxygen content of the charge air to 17.5%, indicating that EGR could be an effective measure to meet the Tier III emission regulation.

Computational fluid dynamics (CFD) modeling has been recognized as a reliable method for the investigation of combustion processes and emission formation, especially in compression ignition engines involving diffusion combustion. Many studies have been conducted to investigate the diesel/gas injection process, the oxidation of diesel and methane, and the formation of pollutants in HPDI combustion through CFD modeling studies coupled with chemical kinetic mechanisms. Agarwal and Assanis carried out detailed multidimensional modeling.
TABLE 1  Main specifications of the 4T50ME-GI research engine

| Parameters          | Value                |
|---------------------|----------------------|
| Type                | 4T50ME-GI            |
| Number of cylinders | 4                    |
| Bore/stroke         | 500 mm/2200 mm       |
| Connecting rod      | 2885 mm              |
| Geometric compression ratio | 18.14  |
| Engine speed @ MCR  | 123 rpm              |
| Power @ MCR         | 7050 kW              |
| IMEP @ MCR          | 20 bar               |
| Turbocharger        | MAN TCA55-VTA        |

of a direct-injection natural gas engine with self-ignition conditions for the purpose of examining ignition, combustion, and formation processes of NO by using KIVA-3 code with the eddy breakup model coupled with a chemical kinetic mechanism for natural gas simulation. Brett et al.28 developed an improved chemical kinetic model to simulate natural gas ignition under a wide range of experimental conditions fueled with various compositions of methane, oxygen, and argon. Li et al.29 presented a CFD combustion model together with an injector model to simulate in-cylinder mixing and combustion processes. They calibrated the model against a set of experimental results. It was revealed that the model could perform accurately under no-EGR conditions; however, under high EGR conditions, the model exhibited larger prediction errors. Lee et al.30 modeled the combustion of two fuels based on different detailed chemical mechanisms to find a mechanism that could adequately predict the combustion process with reasonable computational efficiency. Based on this modeling work, they concluded that the trends in the simulation results were not sensitive to the mechanism applied. Kheirkhah31 studied the effects of different injection strategies on the performance of HPDI combustion by using the GOLD code (a CFD code based on OpenFOAM). A large eddy simulation (LES) turbulence model was employed to resolve turbulent charge and injection motions. The results showed that both late postinjection (LPI) and slightly premixed combustion (SPC) were beneficial for PM reduction. However, the diesel-gas kinetic interactions were not well resolved, which resulted in discrepancies between the experimental and numerical combustion phases.

From the above literature, it is evident that previous numerical studies on HPDI combustion have been mainly conducted on small-scale engines, while combustion modeling investigations on large-dimension engines, such as low-speed marine engines, have seldom been reported from the authors’ review. This paper computationally investigated the effects of different injection strategies of diesel and natural gas on the combustion and emission characteristics of a large-scale marine HPDI gas engine by using the 3D CFD code CONVERGE coupled with a reduced n-heptane mechanism, including the combustion submechanism of methane. First, the combustion model was validated against experimental results from the literature. Then, extensive modeling investigations on the different injection strategies were explored. Three kinds of injection strategies, including absolute injection timing (AIT), gas injection timing, and pilot diesel injection timing, were considered to explore their effects on the mixing, combustion, and emission characteristics. The modeling results were collected and analyzed in detail and could provide guidance for dual-fuel engine design in the future.

2  | NUMERICAL SIMULATION MODEL

2.1  | Test engine

Computational investigations were conducted on a MAN (located in Germany) Diesel & Turbo 4T50ME-GI research engine, which is a four-cylinder, low-speed, two-stroke, uniflow-scavenged marine engine, with an electronically controlled exhaust valve and fuel injection system modified from those of the 4T50ME-X engine to introduce natural gas combustion.23 Table 1 presents the main specifications of the research engine with a large cylinder bore of 500 mm. In addition, the engine has a geometric compression ratio of 18.14 to improve the fuel economy. The engine speed at full load conditions (also called MCR) is 123 rpm, which decreases as a function of the load according to a simulated propeller curve.32 Two groups of injectors are symmetrically equipped on the cylinder head to accomplish the flexible injection strategies fueled with gas/diesel. Figure 1 shows an illustration of such an arrangement as published in Reference.33 When switching to the diesel mode, a heavy fuel oil (HFO) amount that covers the full load is injected from the pilot injectors. In the dual-fuel mode, marine diesel oil (MDO) or HFO is first injected as a pilot fuel, and when the pilot flame reaches the front face of the gas injector nozzle, natural gas is injected. The natural gas is ignited immediately by the pilot flame.12 More detailed specifications of the engine can be found in the literature.5,23,34,35

2.2  | Numerical simulation model

The simulation code used is the 3D CFD software CONVERGE,36 a commercially available CFD code. CONVERGE uses an innovative modified cut-cell Cartesian method for grid generation with the possibility of adding local refinement and adaptive mesh resolution (AMR).36,37 Table 2 shows the list of submodels applied for simulation. The n-heptane is used to simulate the oxidation behavior of diesel,
whereas natural gas is represented by methane. A reduced n-heptane mechanism consisting of 109 species and 334 reactions was used to represent the combustion chemistries of n-heptane and natural gas. The reduced n-heptane and natural gas (methane) submechanisms were adopted from the detailed LLNL PRF mechanism \cite{38} and the Aramco 1.3 mechanism, \cite{39} respectively, which have been extensively validated against various experimental data. The DRGEP and sensitivity analyze approach are used to remove unimportant species and reaction pathways. Ignition delay predictions using the present reduced mechanism were compared with shock tube experimental data.\cite{40,41} As shown in Figure 2, the ignition delay predictions of n-heptane are in good agreement against shock tube experimental data at medium and high temperature. At low temperature, the calculated ignition delay times are slightly higher than experimental data. As shown in Figure 3, the calculated ignition delay times of methane agree well with experimental data at low and high equivalence ratios. At stoichiometric condition, the calculated ignition delay times of methane are slightly higher than experimental data. Since in HPDI natural gas engines, the in-cylinder temperature near TDC is higher than 900 K, and the reduced mechanism is suitable to simulate the combustion of n-heptane and methane. The physical properties of liquid diesel were represented by those of tetradecane (C\textsubscript{14}H\textsubscript{30}). Additionally, the no-time-counter (NTC) collision method by Schmidt and Rutland was used to describe the droplet collision effects.\cite{42} The spray-wall interaction was simulated using the O’Rourke model,\cite{43} and the Kelvin-Helmholtz/Rayleigh-Taylor (KH-RT) model was applied for the spray atomization and breakup simulation.\cite{44} In addition, the mass flow rate was considered to be the boundary

\begin{table}[h]
\centering
\caption{List of submodels used in the simulation}
\begin{tabular}{|l|l|l|}
\hline
Phenomenon & Model & Reference \\
\hline
Combustion & SAGE & \cite{38} \\
Droplet collision & NTC collision method & \cite{39} \\
Spray-wall interaction & O’Rourke model & \cite{40} \\
Spray atomization and breakup & Hybrid KH-RT & \cite{41} \\
Gas injection & Inflow boundary & \cite{42} \\
Turbulence & RNG \kappa-\epsilon model & \cite{46} \\
NOx & Extended Zeldovich model & \cite{47} \\
Soot & Hiroyasu soot model & \cite{48} \\
\hline
\end{tabular}
\end{table}
condition for the gas fuel injection. Given that the instantaneous mass flow rate was not experimentally measured, an assumed injection profile was adopted. As referenced from the work of Scarcelli et al., the injection profile was trapezoidal, and the mass flow rate was assumed to be constant with a 1°CA ramp-up and 1°CA ramp-down. The Reynolds-averaged Navier-Stokes (RANS)-based renormalization Group (RNG) κ-ε submodel was used to simulate the in-cylinder turbulence. The SAGE solver was adopted to simulate the combustion process. The extended Zeldovich model and Hiroyasu soot model were used to simulate the NOx and soot emissions, respectively.

To validate whether the inflow boundary can be used for high-pressure gas injection, the evolution of the methane concentration in terms of mass fraction from CFD is compared with the experimental results in Figure 4. The red contours indicate a mass fraction of 1.0 (100%), while blue represents a concentration of zero. The comparison shows reasonable agreement in terms of capturing the jet penetration.

For the engine simulation, the mesh was generated automatically by using an adaptive generation method with up to 1,500,000 grids. Figure 5 shows the three-dimensional geometric structure of the model and the computational mesh at the bottom dead center (BDC), and this structure includes a scavenge box, cylinder, and exhaust port.

Bench experimental conditions and the corresponding results, including the combustion characteristics and emission data, were obtained from the literature as modeling inputs and validations. In this work, the 75% load has been selected for further study mainly due to its highest weighting factor (0.5) in the ISO 8178-E3 four-mode test cycle, which defines the maximum NOx emission limits for marine diesel engines according to the IMO regulations. The opening and closing timings of the scavenge ports and exhaust valve were −217/−143°CA and −277/−73.5°CA, respectively. The cyclic injection quantities of the pilot diesel fuel and natural gas were 0.13 and 26 g, respectively, and the start-of-injection timings of diesel and gas fuel were −2°CA and −1°CA, respectively. The simulation started from the exhaust valve opening (EVO) and ran a whole cycle.

3 | MODELING VALIDATION

The mesh grid size can significantly affect the computational time and accuracy. The CONVERGE code adopts a flexibly adaptive mesh that can simultaneously save

FIGURE 4 Methane mass fraction—comparison between experiment and simulation
calculation time while ensuring accuracy. The grid can be refined by adjusting the base grid, fixed embedding and AMR. In this work, a mesh analysis method similar to that of Sun et al., who analyzed the effect of mesh grid size on in-cylinder pressure in a 340 mm marine engine model, was introduced. The cylinder, scavenge port, and injection were embedded. The effects of the base grid and the AMR are shown in Figure 6. Figure 6A shows that the compression pressure profile of the 0.01 m base grid case is in reasonable agreement with that of the 0.005 m base grid case. The computational accuracy increased gradually with increasing grid size. Additionally, the calculated pressure profile of the 0.0025 m AMR case is also in good agreement with that of the 0.00125 m AMR case, as shown in Figure 6B. Given that the calculation time increases significantly when more refined mesh is used, a base grid of 0.01 m and an AMR of 0.0025 m were adopted to maintain the computational accuracy and efficiency simultaneously.

Figure 7 shows a comparison between the calculated and measured in-cylinder pressure and heat release rate (HRR) profiles. The trend of the calculated results is in reasonable agreement with experimental data, but the calculated HRR is slightly higher than that of experiment. On one hand, the simulation result was obtained by calculating the heat that chemical reaction released heat minus the in-cylinder wall transfer heat, while the experimental HRR was derived by the measured averaged in-cylinder pressure and temperature. On the other hand, the interaction between turbulence and chemical in current model needs to be further improved. Nevertheless, the current difference between experiment and simulation will not have a significant impact on subsequent qualitative analysis. Detailed comparisons of the main combustion characteristics are presented in Table 3. The maximum error for the maximum in-cylinder pressure ($P_{\text{max}}$) phasing is 5.6%, while the simulated combustion phasing (CA 50), indicated specific fuel consumption (ISFC), and NOx and CO emissions are all very close to those of the experimental data, indicating that the model setup should be reasonable and reliable.
A parametric study was conducted to study the effects of injection strategies on the combustion characteristics and emissions of high-pressure direct-injection low-speed natural gas engines operated at 75% load conditions (n = 112 rpm). Both the AIT (the injection interval keeping constant, simultaneously advances or retards the pilot fuel injection timing, and gas injection timing) and RIT (the pilot fuel injection timing was fixed at −2°CA ATDC, and the gas injection timing was advanced or retarded from −1°CA ATDC. Alternatively, the gas injection timing was fixed at −1°CA ATDC, and the pilot fuel injection timing was advanced or retarded from −2°CA ATDC) injection sweeps were tested. Other parameters (ie amount of injected fuel, injection profile, and scavenging condition) were kept constant for all cases to ensure that the combustion characteristics, and emissions could only be affected by the injection strategies. A schematic of the injection sequence is shown in Figure 8, and the detailed injection strategies are listed in Tables 4-6.

4 | RESULTS AND DISCUSSION

4.1 | The effect of absolute injection timing (AIT)

4.1.1 | Combustion characteristics

The in-cylinder pressure and HRR profiles at different AITs are shown in Figure 9. A normal combustion process in an HPDI gas engine is identified to take place over five stages (Figure 10), which mainly include the pilot fuel ignition delay, pilot fuel premixed combustion, rapid combustion...
of premixed gaseous fuel, mixing-controlled combustion stage, and late combustion. The ignition delay of diesel is defined as the duration from the pilot start of injection to the start of pilot combustion, and the ignition delay of natural gas is defined as the duration from the gas start of injection to the start of rapid combustion of natural gas. As shown in Figure 10, the ignition delay of natural gas is much shorter than that of diesel and thus can be neglected in HPDI gas engines. Figure 9 shows that the combustion phase is delayed significantly as the AIT is retarded. Since the pilot injection timing and gas injection timing retarded, the ignition delay times are changed slightly, and the start of combustion is delayed. Naturally, the combustion phase is delayed. The in-cylinder pressure profile exhibits a two-peak characteristic with a much retarded AIT timing due to the late main heat release, where the first peak takes place at the top dead center as a consequence of the retarded initiation of combustion events, while the second peak induced by the heat release of main fuel natural gas occurs after a drop caused by the downward movement of piston in the expansion stroke. Additionally, it can also be observed that the shapes of the different HRR profiles are similar despite the different combustion phasing. The combustion duration is defined as the duration from the start of combustion to when 99.9% of heat has been released. For all cases, the ignition is controlled by the pilot fuel injection timing, and the combustion duration is related to ignition delay of pilot fuel. The ignition delays of pilot fuel of ORG and A_2 cases are the shortest, providing less time for mixing, reducing the fuel utilization efficiency. Accordingly, the combustion gets worse, and the peak mean in-cylinder temperature decreases. The late combustion

| Table 4 | Setup for the absolute injection timing |
|---------|----------------------------------------|
| Case    | Pilot injection timing (°CA ATDC) | Gas injection timing (°CA ATDC) |
| A_4 (Advanced 4°CA) | −6 | −5 |
| A_2     | −4 | −3 |
| ORG (Original case) | −2 | −1 |
| D_2 (Delayed 2°CA) | 0  | 1  |
| D_4     | 2  | 3  |

| Table 5 | Setup for gas injection timing of relative injection timing |
|---------|--------------------------------------------------------|
| Case    | Pilot injection timing (°CA ATDC) | Gas injection timing (°CA ATDC) |
| A_4 (Advanced 4°CA) | −2 | −5 |
| A_2     | −2 | −3 |
| ORG (Original case) | −2 | −1 |
| D_2 (Delayed 2°CA) | −2 | 1  |
| D_4     | −2 | 3  |

| Table 6 | Setup for pilot injection timing of relative injection timing |
|---------|--------------------------------------------------------|
| Case    | Pilot injection timing (°CA ATDC) | Gas injection timing (°CA ATDC) |
| A_4 (Advanced 4°CA) | −6 | −1 |
| A_2     | −4 | −1 |
| ORG (Original case) | −2 | −1 |
| D_2 (Delayed 2°CA) | 0  | −1 |
| D_4     | 2  | −1 |

FIGURE 9 The in-cylinder pressure and heat release rate profiles at different absolute injection timings
reduces the time between combustion and exhaust valve opening, decreasing the time available for the kinetics of oxidation to occur. The retarded combustion phase in diesel engines generally decreases NOx but increases unburned fuel, intermediate species such as CO and soot, and exhaust temperatures. The in-cylinder temperatures and equivalent ratio distributions at 0°CA, 5°CA, 10°CA, 15°CA, and 20°CA ATDC for different AIT cases are shown in Figure 12. For each slice, the temperature is present on the left, while the equivalent ratio is on the right. The black line with the arrow represents the air flow motion, from which the contribution of the entrainment of the high-speed gas jet on the in-cylinder mixing and combustion can be seen. Additionally, it is evident that the combustion phase and the entrainment effect are delayed as the AIT is retarded. Compared with the original (ORG) case, a large high-temperature region exists at 20°CA ATDC for the D_4 case. Combustion does not start in the inner region with a high concentration of gas fuel (larger than 2.0) but in areas with an appropriate equivalent ratio (1.0~2.0). A low equivalent ratio region (<0.4), which cannot be ignited, initially exists between the cylinder wall and the gas jet direction and also exists in the intermediate region of the cylinder in the late combustion stage. Additionally, a large combustion region with high temperature can also be seen near the cylinder wall, and this region may result in a high thermal load for the engine cylinder and adverse effects on the emissions. To solve these problems, a better distribution of fuel and air may be achieved by optimizing the gas injector arrangement and in-cylinder airflow.

4.1.2 Emissions

The ISFC and emissions of soot, NOx, and CO+HC different AITs are presented in Figure 13. This figure shows that NOx emissions are reduced with retarded injection timing, and this effect is similar to that of the traditional diesel engine due to the lower mean in-cylinder temperature. However, it seems that there is a limit to the extent to which NOx emissions can be mitigated by retarding gas injection. Additionally, there is a similar increasing trend for the soot and HC+CO emissions with the retarded injection timing. While the trade-off between NOx and ISFC has not changed, the ISFC increases with the retarded injection timing. The emissions of these species significantly depend on the mixture distribution. With retarded injection timings, less time is available for the oxidation process. On the other hand, although the postcombustion temperature is higher, as shown in Figure 11, which is beneficial for the oxidation process, the decreases in power output may offset this effect, thus resulting in higher indicated specific emissions.

4.2 The effect of relative injection timing (RIT)

4.2.1 Combustion characteristics

The in-cylinder pressure and HRR profiles for different RITs are depicted in Figure 14, in which the cases of gas injection timing are presented in the upper side and those of pilot
FIGURE 12  The in-cylinder temperature and equivalent ratio distributions at 0°CA, 5°CA, 10°CA, 15°CA, and 20°CA for different absolute injection timing cases. (The left side of a slice is the in-cylinder temperature distribution, and the right side of a slice is the equivalence ratio distribution.)
injection timing in the lower side. As shown in Figure 14, by varying the pilot and gas fuel injection timings, the behavior of the combustion process can be changed significantly. The effect of gas injection timing on the combustion process is similar but not exactly the same as that of the AIT. When the gas injection timing varied from A_4 to D_4 with the pilot fuel injection timing fixed at −2°CA ATDC, the peak value of the in-cylinder pressure decreased from 19 to 14.4 MPa, accompanied by a retarded combustion phase. Although the pilot injection timing is unchanged, the gas injection timing is retarded, and the main combustion process is delayed due to most of the heat release from the oxidation of natural gas. Thus, the peak value of in-cylinder pressure decreases. Compared with the ORG case, an earlier gas injection timing means that partially gaseous fuel was mixed with air before pilot fuel injection. The atomization, evaporation, and combustion process of liquid fuel will proceed in atmospheres of natural gas-air mixtures, which can elongate the ignition delay of pilot fuel. The cooling effect and diluting effect of gaseous fuel on the in-cylinder charge also contribute greatly to this situation. As more pilot fuel is injected before ignition, the amount of liquid fuel-air mixture increases, and then, more heat is released simultaneously when autoignition occurs. This effect should be the major reason for the significant pressure increase in the initial combustion stage, along with the substantial HRR before the main combustion stage. Additionally, the in-cylinder pressure depicts two peaks if a later gas injection timing is adopted. When the natural gas is injected into cylinder, the pilot fuel is almost burnt out. Accordingly, the first peak takes place after TDC as a consequence of the combustion of a small amount of pilot diesel while the second peak induced by the heat release of main fuel natural gas occurs after a drop caused by the downward movement of piston. As the gas injection timing is retarded, there is more time available for combustion of the pilot fuel before the

**FIGURE 13** Predicted indicated specific emissions of Soot, NOx, and HC + CO at different absolute injection timings

| Injection Timing | Soot (g/kW·h) | NOx | CO + HC (g/kW·h) | ISFC (g/kW·h) |
|------------------|---------------|-----|-----------------|---------------|
| A_4              | 0.035         | 0.20| 0.45            | 0.16          |
| A_2              | 0.030         | 0.25| 0.40            | 0.15          |
| ORG              | 0.025         | 0.30| 0.35            | 0.14          |
| D_2              | 0.020         | 0.35| 0.30            | 0.13          |
| D_4              | 0.025         | 0.40| 0.45            | 0.16          |

**FIGURE 14** The in-cylinder pressure and heat release rate profiles for different relative injection timings. A, varied gas fuel injection timing and B, varied pilot fuel injection timing
gas jet mixes with air and interacts with the diesel spray flame. In addition, a separation between the combustion processes of the pilot fuel and gaseous fuel evidently occurs in the HRR profiles when the gas injection timing is retarded, and the heat release process of the pilot fuel almost finishes when gas is injected into the cylinder in the D₄ case; thus, misfire may occur if the gas injection timing is further retarded.

When the gaseous fuel injection timing is fixed at −1° and the variation in the pilot fuel injection timing is taken into account, the main combustion process remains almost unchanged. The peak value of the in-cylinder pressure decreases from 16.8 to 16.3 MPa with a negligible difference in the peak phase as the pilot injection timing is varied from the A₄ case to the D₄ case. The same is true for the HRR profiles. It seems that the influence of the pilot injection timing is minor on the main combustion process but remarkable on ignition and the initial combustion stage. This finding can be explained by the fact that the heat released by the pilot fuel is quite small, that is, below 5%. Since the gas timing is unchanged, the variation in the main combustion process is ignorable. When the pilot injection timing is retarded compared with that of the ORG case, steep profiles of the in-cylinder pressure occur in the initial combustion stage, accompanied by narrow and towering peaks of the HRR at the same crank angle. The reason is similar to that of the gas injection timing cases. Retarded timings result in interactions between the pilot spray and partially premixed gas fuel. The cooling effect and dilution effect of the gas fuel make it more difficult for the pilot fuel to form a mixture appropriate for ignition. Then, more fuel-air mixtures can be formed, and consequently, more heat is initially released. Additionally, the heat released in the late combustion stage is higher, and this heat may come from the oxidation of incomplete combustion production.

Figure 15 shows the calculated mean in-cylinder charge temperature vs crank angle for different RIT cases. As shown in these plots, when the pilot fuel injection timing is fixed at −2° and the gas injection timing varies from A₄ to D₄, the peak value of the mean in-cylinder temperature decreases from 1649 to 1605 K, accompanied by a retarded peak phase. It is evident that the effect of the gaseous fuel injection timing on the mean in-cylinder temperature is similar to that of the AIT, although the variation ranges decrease slightly. In contrast, the pilot fuel injection timing has a quite different influence on the mean in-cylinder temperature. As the pilot injection timing is retarded with a constant gaseous fuel injection timing, the ignition process is delayed, whereas the peak phase remains almost unchanged. Since the gas injection timing keeps constant, the variation of main combustion process is ignored, and thus, the peak phase is almost unchanged. Compared with the ORG case, advanced pilot injection timings have a negligible influence on the temperature trace, but retarded timings result in relatively low in-cylinder temperatures during the main combustion process and slightly high temperatures in the late combustion stage. Advanced pilot injection timing has great influence on the ignition of pilot fuel, but the ignition timing of natural gas is almost unchanged. Therefore, the variation in temperature trace is negligible. Retarded pilot injection timing results in a later ignition of pilot fuel, and the sequent ignition of natural gas is delayed. Although the main combustion phase is almost unchanged, but there is less time for fuel-air mixing, and thus, the combustion efficiency is reduced and the temperature is lower. Accordingly, the combustion temperature of late combustion stage is higher due to more fuel consumed in this stage. The peak value of the mean in-cylinder temperature decreases by 41 K, and the temperature in the late combustion stage increases by 8 K, and these results are consistent with the HRR profiles shown in Figure 14. Additionally, the gap in the peak values between the gas injection timing cases
and the pilot cases is nearly equal. Figure 16 depicts the temperature distribution and the contour lines with equivalent ratio $= 1$ for different RIT cases. As the gas injection timing is changed, there are differences when the gaseous fuel interacts with the pilot fuel flame. Thus, the pilot injection timing definitely affects the following combustion process. Figure 16A shows that retarded gas injection timing delays the main combustion process. Compared with the ORG case, in which the interaction between the gas jet and pilot fuel flame is obvious at 0°, the partially premixed gas burns (Figure 14A shows a very high HRR in the initial heat release phase) rapidly in the advanced gas injection timing cases. The earlier the gas injection timing is, the more partially premixed gas that exists in front of the diesel spray, and consequently, the more heat that is released after the mixture is ignited by the pilot fuel flame. This effect is the reason why the HRR increases sharply in the initial stage of the A_4 case. If the gas injection timing is gradually retarded, the combustion of the pilot fuel almost finishes before it ignites the gas jet. The initial interaction region is away from the high-temperature region of the pilot flame, as shown in the 5° plot in Figure 16A. Therefore, the gaseous fuel ignition region may be smaller, which ultimately results in a relatively larger high-temperature combustion zone in the 20° case. For the pilot timing cases, Figure 16B shows that the main combustion process remains almost unchanged as the pilot injection timing is varied. In contrast to the varied gas fuel injection timing, the retarded pilot fuel injection timing exhibits a partially premixed gas burn (very high HRR in the initial heat release phase, as shown in Figure 14B). The later the pilot injection timing is, the more partially premixed gas burn that exists. For a retarded pilot injection timing, more gas fuel is injected into the cylinder before spontaneous ignition of the pilot fuel; therefore, more time is available for gas fuel-air mixing. The distribution of high-temperature regions is similar among these cases, but differences in the equivalent ratio distributions still exist.

FIGURE 16  The temperature distribution and the contour lines for equivalent ratio $= 1$ in the cylinder for different relative injection timing cases
4.2.2 Emissions

Figure 17 depicts the effect of different RITs on the soot, NOx, and HC+CO emissions. The figure shows that the RIT has dramatic impacts on the pollutant formation and emissions. Compared with the AIT cases, the influence of the RITs is more complicated. Figure 17A shows that the NOx and HC+CO emissions are not changed much when the gas injection timing is shifted from A_2 to D_2, and the highest and lowest soot and HC + CO emissions can be obtained in the A_2 case and A_4 case, respectively. For NOx emissions, the peak value is 12.8 g/kW·h in A_4, which is lower than that of the AIT’s 14.8 g/kW·h. It is observed that partially premixed gas fuel is beneficial to reduce both the NOx and soot emissions; however, the ratio of gas fuel is the key parameter that needs further investigation and optimization. The ISFC is increased with retarded gas injection timing. In addition, considering both the soot and HC + CO emissions, slightly retarding the gas injection timing is also feasible.

Figure 17B shows that the pilot injection timing also has significant impacts on the pollutant formation and emissions. When the pilot timing is retarded by 2°CA ATDC compared with that of the ORG case, the soot and HC+CO emissions are the highest among all these cases. However, a drastic reduction in these two emissions is observed when the pilot timing is further retarded. Additionally, an advanced pilot timing is also beneficial for decreasing soot and HC + CO emissions. For NOx emissions, the highest and lowest values appear in the A_2 case and D_2 case, respectively. However, a further advanced or retarded pilot timing results in a reverse trend. Additionally, the ISFC shows a reverse trend with NOx emissions. Taking all these emissions into account, the A_4 and D_4 cases have some advantages compared with the ORG cases.

5 CONCLUSION

The injection strategies (pilot diesel injection timing and gas injection timing) have a great influence on the combustion and emissions of dual-fuel marine engines. In the current study, numerical investigations have been conducted to study the
influences of different pilot injection timings and gas injection timings on the combustion and emission characteristics of a large-bore two-stroke natural gas/diesel dual-fuel marine engine. The major results can be summarized as follows:

1. The effects of AIT on combustion and emission characteristics are similar to those of traditional diesel engines. As the AIT varies from advanced by 4°CA to retarded by 4°CA, the peak values of the in-cylinder pressure and temperature decrease gradually, accompanied by a retardation of the combustion phase. Additionally, a conventional trade-off between NOx and soot/CO/HC/ISFC emissions can also be observed as the AIT is retarded.

2. RIT has more complicated impacts on the combustion characteristics and emissions. Advanced gas fuel injection and retarded pilot fuel injection timings lead to partially premixed gas burn, and further advanced or retarded timings result in more partially premixed gas burn. As the gas injection timing varies from advanced by 4°CA to retarded by 4°CA, the peak value of the in-cylinder pressure decreases from 19 MPa to 14.4 MPa, accompanied by a retarded combustion phase.

3. An advanced pilot fuel injection timing has little effect on the in-cylinder pressure and combustion phase. A retarded pilot fuel injection timing results in a slightly decreased in-cylinder pressure and a slightly delayed combustion phase. A gas injection timing advanced by 4°CA leads to the lowest soot and HC+CO emissions, but the NOx emissions are the highest. However, the lowest ISFC occurs at a pilot fuel injection timing advanced by 2°CA. A slightly retarded gas fuel injection timing has a negligible effect on NOx emissions. A further retarded gas fuel injection timing reduces NOx emissions, but soot and HC+CO emissions increase. A pilot fuel injection timing retarded by 2°CA results in the lowest NOx emissions, but the soot and CO+HC emissions are the highest.

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NOMENCLATURE

HPDI High-pressure direct-injection
ORG ORiGinal
IMO International Maritime Organization
NOx Nitrogen oxide
SOx Sulfur oxide
PM Particulate matter
VOC Volatile organic compound
NG Natural gas
GHG Greenhouse gas

TDC Top dead center
MCR Maximum continuous rating
RCEM Rapid compression expansion machine
EGR Exhaust gas recirculation
CFD Computational fluid dynamics
LES Large eddy simulation
LPI Late postinjection
PSOI Pilot fuel start of injection
PEOI Pilot fuel end of injection
AIT Absolute injection timing
RIT Relative injection timing
ISFC Indicated specific fuel consumption
MDO Marine diesel oil
AMR Adaptive mesh resolution
NTC No-time-counter
RANS Reynolds-averaged Navier-Stokes
KH-RT Kelvin-Helmholtz/Rayleigh-Taylor
BDC Bottom dead center
EVO Exhaust valve opening
HRR Heat release rate
RNG Reynolds-averaged Navier-Stokes-based renormalization group
Pmax Maximum in-cylinder pressure
IMEP Indicated mean effective pressure
HFO Heavy fuel oil
SPC Slightly premixed combustion
GSOI Natural gas fuel start of injection
Geoi Natural gas fuel end of injection

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REFERENCES

1. Ackermann G. Compendium Marine Engineering: Operation, Monitoring, Maintenance. Hamburg, Germany: Seehafen Verlag; 2009.
2. Becker R. MARPOL 73/78: an overview in international environmental enforcement. Geo Int’l Envtl L Rev. 1997;10:625.
3. Emission Standards. IMO Marine Engine. 2008. http://www.DieselNet.com/standards. Accessed November 13, 2008.
4. Yang ZL, Zhang D, Caglayan O, et al. Selection of techniques for reducing shipping NOx and SOx emissions. Transp Res D Transp Environ. 2012;17(6):478-486.
5. Raptotasios SI, Sakellaridis NF, Papagiannakis RG, Hountalas DT. Application of a multi-zone combustion model to investigate the NOx reduction potential of two-stroke marine diesel engines using EGR. Appl Energy. 2015;157:814-823.
6. Liu H, Zhang H, Wang H, Zou X, Yao M. A numerical study on combustion and emission characteristics of marine engine through miller cycle coupled with EGR and water emulsified fuel (No. 2016-01-2187). SAE Technical Paper. 2016.
7. Yamanishi D, Maeda K, Dan T. Application of a newly developed water mixture fuel generator to IMO regulations. Mar Eng. 2018;53(3):380-385.
8. Kazuyuki M, Junichi H, Atsushi S, et al. Development of NOx reduction system that combines an oxygen reduction membrane with water mixed fuel. Proceedings of the 28th CIMAC Congress. 2016.
9. Kyrtatos P, Herrmann K, Hoyer K, Boulouchos K. Combination of EGR and fuel-water emulsions for simultaneous NOx and soot reduction in a medium speed diesel engine. In: 28th CIMAC World Congress. 2016.

10. Christen C, Brand D. IMO tier 3: gas and dual fuel engines as a clean and efficient solution. In: Proceedings: Conseil International Des Machines A Combustion (CIMAC) Congress. 2013.

11. Nerheim L. The position of the gas engine in competition with diesel engine and gas turbine. In: 6th Dessau gas Engine Conference, Dessau-Roßlau. 2009:26-27.

12. Takasaki K. Observations on the development and practical application of marine gas engines. Class NK Tech Bull. 2012;30:1-9.

13. Liu J, Wang J, Zhao H. Optimization of the injection parameters and combustion chamber geometries of a diesel/natural gas RCCI engine. Energy. 2018;164:837-852.

14. Zheng QP, Zhang HM. A computational study of combustion ignition natural gas engine with separated chamber. Fuel. 2005;84(12-13):1515-1523.

15. Chen Z, Zhang F, Xu B, Zhang Q, Liu J. Influence of methane content on a LNG heavy-duty engine with high compression ratio. Energy. 2017;128:329-336.

16. Liu J, Zhang X, Wang T, Zhang J, Wang H. Experimental and numerical study of the pollution formation in a diesel/CNG dual fuel engine. Fuel. 2015;159:418-429.

17. Liu J, Zhao H, Wang J, Zhang N. Optimization of the injection parameters of a diesel/natural gas dual fuel engine with multi-objective evolutionary algorithms. Appl Therm Eng. 2019;150:70-79.

18. Papagiannakis RG, Kotsiopoulos PN, Zamnis TC, Yfantis EA, Hountalas DT, Rakopoulos CD. Theoretical study of the effects of engine parameters on performance and emissions of a pilot ignited natural gas diesel engine. Energy. 2010;35(2):1129-1138.

19. Demosthenous E, Mastorakos E, Stewart Cant R. Direct numerical simulations of dual-fuel non-premixed autoignition. Combust Sci Technol. 2016;188(4-5):542-555.

20. Miyake M. The development of high output, highly efficient gas burning diesel engines. 15th CIMAC. 1983.

21. Miyake M. Gas Injection Diesel Engine and its Application to LNG Vessel. Tokyo: ISME; 1983.

22. Biwa T, Beppu O, Pedersen PS, Grone O, Schnorr O, Fogh M. Development of the 28/32 Gas Injection Engine. Kobe: ISME Kobe; 1990.

23. Juliussen L. MAN B&W ME-GI Engines Recent Research & Results. Kobe: ISME; 2011.

24. Imhof D, Tsuru D, Tajima H, Takasaki K. High-pressure natural gas injection (GI) marine engine research with a rapid compression expansion machine. 27th CIMAC, Shanghai, Paper 12. 2013.

25. Ishibashi R, Tsuru D. An optical investigation of combustion process of a direct high-pressure injection of natural gas. J Mar Sci Technol. 2017;22(3):447-458.

26. Li Y, Li H, Guo H, Li Y, Yao M. A numerical investigation on methane combustion and emissions from a natural gas-diesel dual fuel engine using CFD model. Appl Energy. 2017;205:153-162.

27. Agarwal A, Assanis D. Multi-dimensional modeling of ignition, combustion and nitric oxide formation in direct injection natural gas engines. SAE Trans. 2000;1088-1103.

28. Brett L, MacNamara J, Musch P, Simmie JM. Simulation of methane autoignition in a rapid compression machine with creviced pistons. Combust Flame. 2001;124(1-2):326-329.

29. Li G, Lennox T, Goudie D, Dunn M. Modeling HPDI natural gas heavy duty engine combustion. In: ASME 2005 Internal Combustion Engine Division Fall Technical Conference. New York, NY: American Society of Mechanical Engineers; 2005:405-413.

30. Lee W, Montgomery D. Numerical investigation of the performance of a high pressure direct injection (HPDI) natural gas engine. In: ASME Internal Combustion Engine Division Fall Technical Conference. New York, NY: American Society of Mechanical Engineers; 2014.

31. Kheirkhhah P. CFD Modeling of Injection Strategies in a High-Pressure Direct-Injection (HPDI) Natural Gas Engine [Doctoral dissertation]. University of British Columbia; 2015.

32. Diesel MAN. Turbo. Basic Principles of Ship Propulsion. Copenhagen: MAN Diesel & Turbo; 2011.

33. Mitsubishi Heavy Industries, LTD. Mitsubishi Sulzer DFD Engine. Japan: Mitsubishi Heavy Industries, LTD; 1986.

34. Andreason A. Modelling of the Oxidation of Fuel Sulfur in Low Speed Two-Stroke Diesel Engines. Bergen: CIMAC; 2010.

35. Ryu Y, Lee Y, Nam J. Performance and emission characteristics of additives-enhanced heavy fuel oil in large two-stroke marine diesel engine. Fuel. 2016;182:850-856.

36. Richards KJ, Senecal PK, Pomraning E. CONVERSE Manual (Version 2.3). Madison, WI: Convergent Science Inc.; 2016.

37. Senecal PK, Richards KJ, Pomraning E, et al. A new parallel cut-cell Cartesian CFD code for rapid grid generation applied to in-cylinder diesel engine simulations (No. 2007-01-0159). SAE Technical Paper. 2007.

38. Curran HJ, Gaffuri P, Pitz WJ, Westbrook CK. A comprehensive modeling study of iso-octane oxidation. Combust Flame. 2002;129(3):253-280.

39. Metcalfe WK, Burke SM, Ahmed SS, Curran HJ. A hierarchical and comparative kinetic modeling study of C1–C2 hydrocarbon and oxygenated fuels. Int J Chem Kinet. 2013;45(10):638-675.

40. Ciezki HK, Adomeit G. Shock-tube investigation of self-ignition of n-heptane-air mixtures under engine relevant conditions. Combust Flame. 1993;93(4):421-433.

41. Hu E, Li X, Meng X, et al. Laminar flame speeds and ignition delay times of methane–air mixtures at elevated temperatures and pressures. Fuel. 2015;158:1-10.

42. Taskiran OO, Ergeneman M. Trajectory based droplet collision model for spray modeling. Fuel. 2014;115:896-900.

43. O’Rourke PJ, Amsden AA. A spray/wall interaction submodel for the KIVA-3 wall film model. SAE Trans. 2000;281-298.

44. Beale JC, Reitz RD. Modeling spray atomization with the Kelvin-Helmholtz-Rayleigh-Taylor hybrid model. Atom Sprays. 1999;9(6):623-650.

45. Le Moine J, Senecal PK, Kaiser SA, et al. A computational study of the mixture preparation in a direct–injection hydrogen engine. J Eng Gas Turbines Power. 2015;137(11):111508.

46. Scarcelli R, Wallner T, Matthews N, Salazar V, Kaiser S. Mixture formation in direct injection hydrogen engines: CFD and optical analysis of single- and multi-hole nozzles. SAE Int J Engines. 2011;4(2):2361-2375.

47. Scarcelli R, Wallner T, Matthews N, et al. Numerical and optical evolution of gaseous jets in direct injection hydrogen engines. SAE Technical Paper, 2011.

48. Scarcelli R, Wallner T, Obermair H, Salazar VM, Kaiser SA. CFD and optical investigations of fluid dynamics and mixture formation in a DI-H2ICE. In ASME 2010 Internal Combustion Engine.
49. Yakhot V, Orszag SA. Renormalization group analysis of turbulence. I. Basic theory. *J Sci Comput*. 1986;1(1):3-51.
50. Dukowicz JK. A particle-fluid numerical model for liquid sprays. *J Comput Phys*. 1980;35(2):229-253.
51. Saario A, Rebola A, Coelho PJ, Costa M, Oksanen A. Heavy fuel oil combustion in a cylindrical laboratory furnace: measurements and modeling. *Fuel*. 2005;84(4):359-369.
52. Rao V, Honnery D. Application of a multi-step soot model in a thermodynamic diesel engine model. *Fuel*. 2014;135:269-278.
53. White TR. Simultaneous Diesel and Natural Gas Injection for Dual-Fuelling Compression-Ignition Engines. Sydney, NSW: University of New South Wales; 2006.
54. ISO BS. 8178-1: 2006. Reciprocating internal combustion engines. Exhaust emission measurement. Test-bed measurement of gaseous and particulate exhaust emissions, 2009.
55. Sun X, Liang X, Shu G, Wang Y, Wang Y, Yu H. Effect of different combustion models and alternative fuels on two-stroke marine diesel engine performance. *Appl Therm Eng*. 2017;115:597-606.
56. Larson CR. Injection Study of a Diesel Engine Fueled With Pilot-Ignited, Directly-Injected Natural gas [Doctoral dissertation]. University of British Columbia; 2003.
57. Heywood JB. *Internal Combustion Engine Fundamental*. New York: McGraw-Hill; 1988.

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