

Article

Effect of Injection Timing on Ammonia–Natural Gas Co-Combustion Performance of Marine Low-Speed Two-Stroke High-Pressure Direct Injection Engines

Shiyu Wang ^{1,†}, Rongsheng Lin ^{2,†}, Fubo Wang ³, Peng Zhang ⁴ , Xinyue Liu ¹, Namin Zhang ¹, Yanjie Ma ¹, Wenfeng Wu ⁵ and Hongliang Yu ^{1,*} 

¹ School of Ocean, Yantai University, Yantai 264005, China; 18052591779@163.com (S.W.); 17860393997@163.com (X.L.); 13373908732@163.com (N.Z.); 15010909273@163.com (Y.M.)

² Jurong Energy (Xinjiang) Co., Ltd., Urumqi 841603, China

³ Yantai CIMC Raffles Offshore Limited, Yantai 264000, China

⁴ Marine Engineering College, Dalian Maritime University, Dalian 116026, China

⁵ School of Naval Architecture and Maritime, Zhejiang Ocean University, Zhoushan 316022, China; wuwenfeng0611@126.com

* Correspondence: yuhongliang19852@163.com

† These authors contributed equally to this work.

Abstract

The development of a marine ammonia–natural gas co-combustion engine aims to achieve high power with low carbon and low nitrogen oxide (NO_x) emissions. Using AVL Fire software, a numerical model of a marine dual-fuel engine with a large cylinder diameter and diesel ignition for ammonia and natural gas co-combustion was constructed. The effects of ammonia injection timing (AI) and natural gas injection timing (NGI) on the combustion process, as well as on NO_x and carbon dioxide (CO₂) emissions, were investigated. The results indicate that, under full-load, low-speed engine conditions, when the energy fractions of three fuels are constant, the heat release rate and pressure are more sensitive to AI. Advancing the injection timing of NH₃ and CH₄ can achieve higher indicated mean effective pressure and indicated thermal efficiency. Under various ammonia and natural gas injection strategies, the marine engine meets the NO_x emission requirements of International Maritime Organization's Tier III. Furthermore, advancing AI or NGI reduces greenhouse gas emissions. Specifically, when ammonia is injected at a high pressure before the top dead center (TDC), advancing injection by every 2 degrees of the crank angle (°CA) reduces equivalent CO₂ emissions by 1.3%. Similarly, when natural gas is injected at a high pressure before the TDC, advancing injection by every 2 °CA reduces equivalent CO₂ emissions by 1.7%.

Keywords: natural gas; ammonia; co-combustion; marine low-speed two-stroke engine; low carbon emissions; injection timing



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1. Introduction

In 2018, the shipping industry accounted for 2.89% of global anthropogenic carbon dioxide (CO₂) emissions [1]. With stricter emission regulations issued by the International Maritime Organization (IMO) and national governments implementing stricter emission standards, the adoption of low/zero-carbon alternative fuels in marine main engines represents one of the most effective pathways to decarbonize shipping. Marine main engines, particularly diesel–natural gas dual-fuel engines, primarily use natural gas as fuel.

This clean, low-carbon energy source helps reduce greenhouse gas emissions and complies with nitrogen oxide (NO_x) emission regulations for marine engines. Natural gas mainly consists of methane (CH_4) and heavier alkanes, with methane burning cleanly due to its low carbon-to-hydrogen ratio [2,3]. However, methane's high C-H bond energy results in slow combustion, low flame propagation rates, and incomplete combustion. These factors lead to limited CO_2 reduction, low thermal efficiency in practical applications, and significant fluctuations in the combustion cycle. Additionally, the high energy required for pure methane autoignition presents a challenge for achieving efficient combustion in diesel engines [4].

Ocean-going vessels that use ammonia as fuel can significantly reduce emissions of CO_2 , SO_x , NO_x , and PM. However, compared to other carbon-based fuels, ammonia presents challenges, such as ignition difficulty and poor combustion stability. Pure ammonia requires a compression ratio between 35 and 100 for use in compression ignition engines, making it unsuitable for such applications [5]. Ragucci et al. [6] conducted mild hydrogen-blended ammonia (MILD) combustion tests and observed that the flame temperature followed a curve that first increased and then decreased with the equivalence ratio, peaking at $\phi = 1$; deviations from an equivalence ratio of 1 resulted in a global decrease in combustion temperature while simultaneously blocking the $\text{NH} \rightarrow \text{NO}$ conversion pathway. They explicitly proposed low- NO_x design criteria: actively matching the equivalence ratio range, limiting the peak flame temperature, and synergistically suppressing NO_x from the ammonia fuel reaction. Ghufran et al. [7] conducted kinetic simulations of a ternary ammonia/methane/hydrogen fuel mixture and found that both fuel-types NO_x and thermodynamic NO_x are highly dependent on the coupling relationship between temperature and the equivalence ratio; at a given temperature, the NO_x production rate increases exponentially as the equivalence ratio approaches 1, and reducing the peak combustion temperature by adjusting the global and local equivalence ratios is a low-cost, low- NO_x strategy for ammonia-based fuels. Peng et al. [8] conducted multi-parameter coupled experiments on ammonia–diesel dual-fuel systems using the response surface method. The study found that the air–fuel ratio and peak cylinder temperature were the two primary variables affecting NO_x . When the air–fuel ratio increases to near 1, the combustion temperature rises simultaneously, and NO_x increases significantly; by lowering the combustion temperature through lean-burn air–fuel ratio matching, NO_x can be suppressed without a significant loss of thermal efficiency, verifying that the coordinated control of temperature and air–fuel ratio is the core solution for source-level NO_x reduction in ammonia internal combustion engines. Morel et al. [9] conducted temperature measurement experiments on an ammonia/methane swirl burner across a wide range of equivalence ratios. The study found that NO_x in a pure ammonia flame peaked in the high-temperature lean-burn range at $\phi = 0.7\text{--}0.8$, while fuel enrichment ($\phi > 1.2$) simultaneously lowered the flame temperature, reduced O radicals, and blocked the pathway for NH oxidation to NO; altering the equivalence ratio essentially changes the peak combustion temperature, and the coupling of these two factors jointly determines the total NO_x production from the ammonia fuel. Based on the above research, it can be concluded that careful control of combustion temperature and the equivalence ratio is necessary to suppress NO_x formation. The global warming potential of N_2O is 300 times that of CO_2 [10].

To address these issues, the current approach involves adding highly reactive fuels and modifying the ignition method. Since natural gas shares similar properties with ammonia fuel—such as density, viscosity, and heat capacity—the ammonia and natural gas through direct injection into the cylinder simultaneously reduce the high CO_2 emissions associated with natural gas combustion and eliminates the need for highly reactive fuels to ignite pure ammonia.

Extensive research on ammonia-fueled engines is being conducted on a large scale [11–20]. Findings indicate that blending ammonia with other highly reactive fuels significantly improves combustion performance, demonstrating the feasible application of such blends in internal combustion engines. To enhance ammonia fuel utilization in these engines, fundamental combustion studies on ammonia-blended fuels have increased, providing numerous key combustion parameters. Shu et al. [21] characterized the flame velocity of ammonia–methanol and compared the curvature and tensile Mach lengths. Experimental results indicated a linear correlation between laminar flame velocity and methanol volume fraction. Dai et al. [22] analyzed the variation in the ignition delay of CH_4 and NH_3 under high pressure, demonstrating that methane effectively reduces ammonia's ignition delay time. Zhang et al. [23] investigated the influence of the methanol mixing ratio, ambient pressure, and ambient temperature on the flame propagation velocity in an ammonia premixed gas. Experimental results showed that reducing the methanol blending ratio decreased the flame propagation velocity. Barbas et al. [24] measured carbon monoxide (CO) and nitric oxide (NO) emissions from ammonia–methanol mixtures in atmospheric laminar burners. Results indicated that increasing the oxygen excess factor reduced NO and CO emissions. Ariemma et al. [25] conducted an experimental study on the mixed combustion of NH_3 and CH_4 . Results indicate that the blend produces higher NO_x emissions compared to pure ammonia and pure methane scenarios. Kurata et al. [26] compared the combustion characteristics of pure NH_3 and dual fuel of NH_3 – CH_4 . Results indicated that ammonia and NO emissions primarily occur in the rich combustion zone, where unburned ammonia reacted with NO via selective non-catalytic reduction. Increasing the ammonia proportion reduced NO conversion efficiency and led to increased NO emissions. Okafor et al. [27] used imaging techniques to study the flow patterns, flame structure, and emission characteristics of an ammonia–methanol mixed fuel. Results indicated that NO_x emission trends correlated with OH concentration variations; the rich–lean combustion technique effectively reduced NO_x emissions. Li et al. [28] analyzed combustion process of NH_3 – CH_4 mixtures for low NO_x emissions under gas turbine operating conditions, providing evidence that a two-stage combustion process can reduce NO_x emissions.

Research on ammonia-fueled marine engines currently focuses primarily on two modes: igniting low-pressure premixed ammonia gas with high-activity fuel, and igniting high-pressure direct-injected (HPDI) liquid ammonia with high-activity fuel. Since liquid ammonia possesses approximately 44% of the energy per unit mass compared to conventional diesel fuel, maintaining equivalent power output in dual-fuel mode versus pure diesel mode necessitates either increasing the amount of liquid ammonia injected into the cylinders. However, studies [29,30] indicate that raising the liquid ammonia injection pressure beyond 100 MPa remains technically challenging. Therefore, under conditions of prolonged liquid ammonia injection duration, reducing ignition fuel consumption while maintaining ignition effectiveness has become a key objective for lowering carbon emissions. Zhu et al. [31] investigated the effect of a prechamber in a low-speed engine on the combustion of premixed ammonia fuel. Their results indicated that the power output under dual-fuel operation was comparable to that of a diesel engine, but the emissions of unburned ammonia and N_2O were high. Zhou et al. [32] conducted a study on the performance of an ammonia–diesel blend in a low-speed engine in high-pressure dual-direct injection versus low-pressure premixed micro-ignition modes. The results showed that, at a maximum ammonia substitution rate of 97%, the ITE of HPDI can reach 51.6%, and the NO_x and unburned NH_3 emissions were lower than those of LPDI. Liu et al. [33,34] conducted a numerical study of a low-speed engine with a 520 mm bore. Studies have shown that, when the ammonia substitution rate exceeds 95%, delayed injection ignition of diesel reduces the engine's ITE to below 49.0%, while NO_x emissions meet Tier III standards.

Because ammonia has an extremely high autoignition temperature and its flame propagates extremely slowly, in conventional compression ignition engines, a mixture of natural gas and ammonia cannot ignite the ammonia using compression heat alone, and natural gas also exhibits high knock resistance. When mixed, the two have absolutely no ability to ignite spontaneously. When diesel is used to ignite ammonia, even though ammonia is carbon-free, there are two major drawbacks: high NO_x and N_2O emissions caused by the nitrogen in the fuel, and unstable combustion at low loads. In our research, we designed a “three-layer injection” system—an inner layer of diesel (high reactivity) + a middle layer of natural gas (medium reactivity) + an outer layer of ammonia (low reactivity)—which creates a “reactivity gradient” within the combustion chamber. This ensures a smooth heat release rate (avoiding the violent combustion caused by the spontaneous ignition of a homogeneous ammonia mixture) while confining the natural gas outside the wall-adjacent region, thereby reducing the escape of unburned natural gas. In preliminary studies, while maintaining a 3% energy contribution from diesel for ignition, we screened synergistic combustion scenarios with varying energy contributions from ammonia and natural gas. The results indicated that the most stable combustion and the lowest overall equivalent emissions were achieved with a 3% diesel, 48.5% ammonia, and 48.5% natural gas mixture. Therefore, this study investigated the ammonia and natural gas injection timing on the co-combustion process in such engines under diesel ignition mode.

2. Research Methods

2.1. Three-Dimensional Simulation Model

The study focuses on a HPDI marine dual-fuel engine (Yantai, China). Table 1 presents the engine’s fundamental parameters, and the single-cylinder geometric model is illustrated in Figure 1. The ignition diesel is injected into the cylinder by diesel injectors before the TDC. Ammonia and natural gas are injected directly into the cylinder by gas injectors. Numerical simulations [35–37] were performed using AVL Fire software (Version 8.0) [38]. The present study utilized the RNG $k-\varepsilon$ two-equation turbulence model [39] for the analysis. To replicate the diesel spray ignition process, the KH-RT model [40] was implemented. The simulation incorporated the SAGE chemical kinetics solver to examine the combustion phenomena, employing a fully mixed combustion model [41–43]. Given the analogous combustion characteristics of diesel and n-heptane, n-heptane was selected as a surrogate fuel for diesel in the simulation. The $\text{NH}_3\text{-CH}_4$ co-combustion chemical kinetic mechanism developed by Ma et al. [44], comprising 76 species and 464 reactions, was adopted. Furthermore, the extended Zeldovich mechanism, which is extensively utilized for predicting thermally driven NO_x formation, was applied due to its demonstrated high accuracy in forecasting NO production [45].

Table 1. Specifications of the engine.

Parameter	Value	Parameter	Value
Power/kW	13,300	Cylinder temperature/K	485
Bore/mm	600	Cylinder cover temperature/K	578
Stroke/mm	2400	Piston temperature/K	661
Speed/ $\text{r}\cdot\text{min}^{-1}$	99	Initial pressure/bar	3.0
Compression ratio	18.8	Fuel mass/(g/kW·h)	141.2 (CH_4) 2.9 (Diesel)
Initial temperature/K	307.2	Scavenging port closing & opening times/ $^\circ\text{CA}$	−138/138
Exhaust valve temperature/K	871	Exhaust valve opening & closing times/ $^\circ\text{CA}$	119/−88

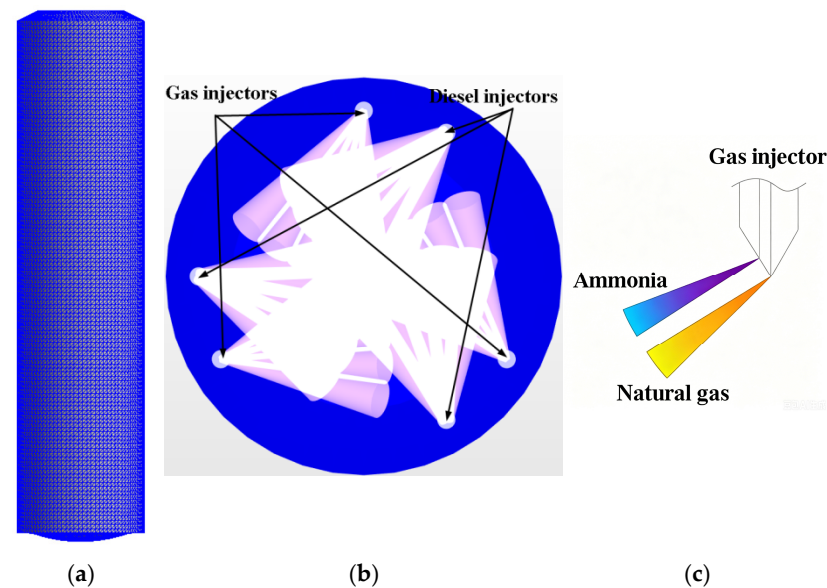


Figure 1. Schematic three-dimensional models. (a) Mesh model. (b) Injector locations. (c) Gas injector.

We assume that the total fuel energy for the three fuel types (diesel, ammonia, and natural gas) is equal to the total fuel energy of the dual-fuel type (natural gas and diesel) at 100% load. The excess air ratio remains constant at 1.85, and the ignition timing for diesel injection is fixed at -4 degrees of the crank angle ($^{\circ}\text{CA}$). The simulation conditions are detailed in Table 2. Cases 1~5 simulate ammonia injection timing delayed from -4 $^{\circ}\text{CA}$ to 4 $^{\circ}\text{CA}$ under the ammonia injection strategy, while cases 6~10 simulate methane injection timing delayed from -4 $^{\circ}\text{CA}$ to 4 $^{\circ}\text{CA}$ under the natural gas injection strategy. DI denotes diesel injection timing for ignition, AI denotes ammonia injection timing, and NGI denotes natural gas injection timing. The energy fraction of each individual fuel is defined as the proportion of its injected energy relative to the total energy supplied by all fuels, expressed as a percentage.

$$E_d = (m_d \times q_d) / (m_d \times q_d + m_{\text{NH}_3} \times q_{\text{NH}_3} + m_{\text{CH}_4} \times q_{\text{CH}_4}) \quad (1)$$

$$E_{\text{NH}_3} = (m_{\text{NH}_3} \times q_{\text{NH}_3}) / (m_d \times q_d + m_{\text{NH}_3} \times q_{\text{NH}_3} + m_{\text{CH}_4} \times q_{\text{CH}_4}) \quad (2)$$

$$E_{\text{CH}_4} = (m_{\text{CH}_4} \times q_{\text{CH}_4}) / (m_d \times q_d + m_{\text{NH}_3} \times q_{\text{NH}_3} + m_{\text{CH}_4} \times q_{\text{CH}_4}) \quad (3)$$

Table 2. Simulation conditions for co-combustion performance of marine low-speed engines.

Case	Fuel Energy Fraction	DI/ $^{\circ}\text{CA}$	AI/ $^{\circ}\text{CA}$	NGI/ $^{\circ}\text{CA}$
1	D3-A48.5-M48.5	-4	-4	1
2	D3-A48.5-M48.5	-4	-2	1
3	D3-A48.5-M48.5	-4	0	1
4	D3-A48.5-M48.5	-4	2	1
5	D3-A48.5-M48.5	-4	4	1
6	D3-A48.5-M48.5	-4	1	-4
7	D3-A48.5-M48.5	-4	1	-2
8	D3-A48.5-M48.5	-4	1	0
9	D3-A48.5-M48.5	-4	1	2
10	D3-A48.5-M48.5	-4	1	4

In the formulas, m_d denotes the mass of diesel and q_d denotes the LHV of diesel. Similarly, m_{NH_3} denotes the mass of ammonia and q_{NH_3} corresponds to the LHV of ammonia. Additionally, m_{CH_4} indicates the mass of natural gas and q_{CH_4} denotes the LHV of natural gas.

As η represents ITE, η can be expressed as follows:

$$\eta = W_i / (m_d \times q_d + m_{\text{NH}_3} \times q_{\text{NH}_3} + m_{\text{CH}_4} \times q_{\text{CH}_4}) \quad (4)$$

In the formula, W_i represents the indicated power.

2.2. Modeling and Validation

Since marine engine cylinder bores are relatively large, mesh adaptivity was evaluated using reference mesh sizes of 10 mm, 15 mm, 20 mm, 30 mm, and 50 mm. Figure 2 depicts the influence of mesh size on the heat release rate (HRR) and cylinder pressure. The calculated pressure curves were compared with experimental values (EVs): the peak pressures were 17.92 MPa for the 10 mm mesh, 18.57 MPa for the 15 mm mesh, 18.94 MPa for the 20 mm mesh, 19.32 MPa for the 30 mm mesh, and 19.39 MPa for the 50 mm mesh. The deviations from the experimental value of 18.9 MPa were -5.19% , -1.75% , 0.21% , 2.22% , and 2.59% . Comparing the pressure and HRR curves, the calculation results most closely match the experimental data when using a 20 mm reference grid.

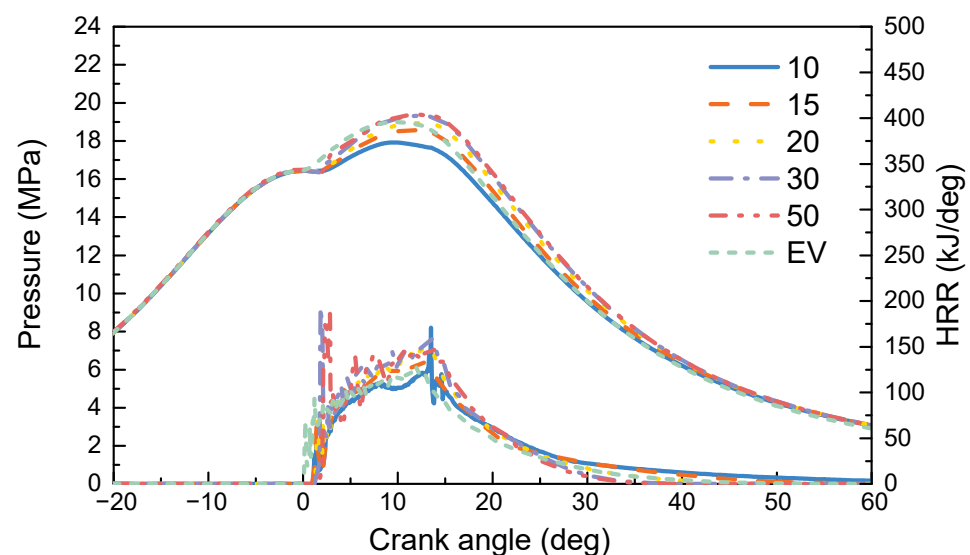


Figure 2. Influence of mesh size on HRR and cylinder pressure.

Experimental data were obtained through engine testing conducted at CIMC Raffles Shipyard in Yantai. Load testing adhered to specifications mandated by classification societies and engine manufacturers. The engine was initiated in pure diesel mode and operated at 10% of its Maximum Continuous Rating (MCR) for a duration of 30 min to facilitate warm-up. After stabilization, the engine was switched to dual-fuel mode. It then operated at 25% load for 45 min, 50% load for 30 min, followed by 30 min at 75% load, 30 min at 85% load, and concluded with 30 min at 100% load. At each load level, the recorded data encompassed compression pressure, engine speed, maximum combustion pressure (P_{max}), boost air pressure and temperature, exhaust gas temperature for each cylinder, and output power, as well as fuel and natural gas consumption. After completing all tests, the load was gradually reduced to allow the engine to cool before normal shutdown. Figure 3 presents a comparative analysis between the experimental data and the simulated outcomes concerning cylinder pressure and HRR. The linear correlation between the simulated pressure and HRR demonstrates a strong agreement with the experimental observations. Specifically, the experimentally measured peak pressure was 18.9 MPa, whereas the simulated peak pressure was 18.94 MPa, indicating a minimal deviation of merely 0.2%.

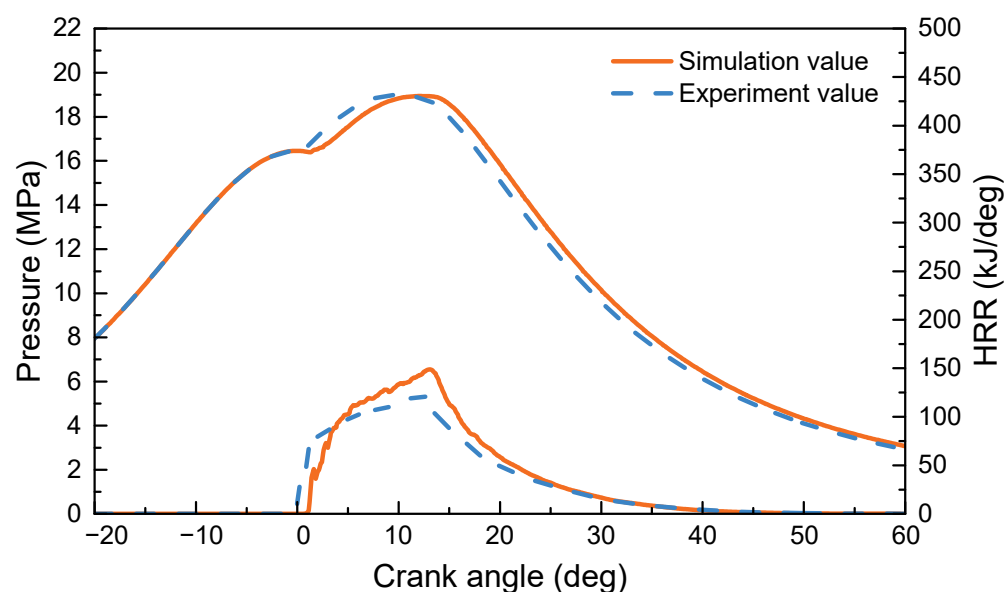


Figure 3. Comparison of experimental and simulation results for dual-fuel mode at 100% load.

The experimental data were sourced from the official marine engine test report issued by the shipyard. Table 3 presents a comparison between the simulated results and the experimental measurements. As illustrated in Table 3, the discrepancies between the simulated outcomes and the experimental values were minimal, indicating that the model and initial conditions employed in this study effectively captured the core performance of the engine and the characteristics of combustion emissions.

Table 3. Comparison of experimental and simulation data.

Parameters	Simulation	Experiment	Deviation
NO _x /ppm	129.4	126.5	2.29%
CO ₂ /%	4.38	4.23	3.55%
HC/ppm	346.1	363.4	4.76%
CO/ppm	167.9	177.6	5.46%
P _{max} /MPa	18.94	18.9	0.21%

3. Results and Discussion

Table 4 illustrates that the three fuels can achieve comparable ITE under more constrained geometric and operational conditions, while also providing greater controllability.

Table 4. Comparative table.

Parameter	Zhou et al. [32]	This Work	Implication of Difference
Bore size	340 mm	600 mm	Big bore limits the NH ₃ combustion rate, requiring reactivity assistance
Energy fraction	Diesel 3%, ammonia 97%	Diesel 3%, ammonia 48.5%, and natural gas 48.5%	Our condition represents a more natural gas–diesel-like retrofit scenario
Load range	75%	100%	Our strategy prioritizes load and emission adaptability over peak efficiency

Table 4. Cont.

Parameter	Zhou et al. [32]	This Work	Implication of Difference
Fuel combination	NH ₃ +diesel	NH ₃ + CH ₄ + diesel	The third fuel provides an extra tuning option for combustion phasing control
Primary objective	Maximizing ITE at extreme NH ₃ ratio	Balancing efficiency, emissions, and operability under geometric constraints	Complementary research directions

To analyze the mechanism of NO_x formation, changes in the in-cylinder NH₃, NO_x, temperature, and O₂ under case 4 were examined, as shown in Figure 4. As illustrated in Figure 4, during the co-combustion of ammonia and natural gas, numerous localized ammonia-enriched zones form within the engine cylinder. The regions of NO_x emissions closely correspond to these ammonia-enriched zones, indicating that fuel-induced nitrogen oxidation occurs continuously across a wide area of the ammonia spray zone, resulting in fuel-type NO_x. Additionally, some thermally generated NO_x is produced at the high-temperature interface between oxygen and the ammonia-enriched zones; these areas act as ignition zones where NH₃ begins to combust. The NO_x distribution demonstrates that, under co-combustion conditions of ammonia and natural gas, fuel-type NO_x constitutes the majority of NO_x emissions.

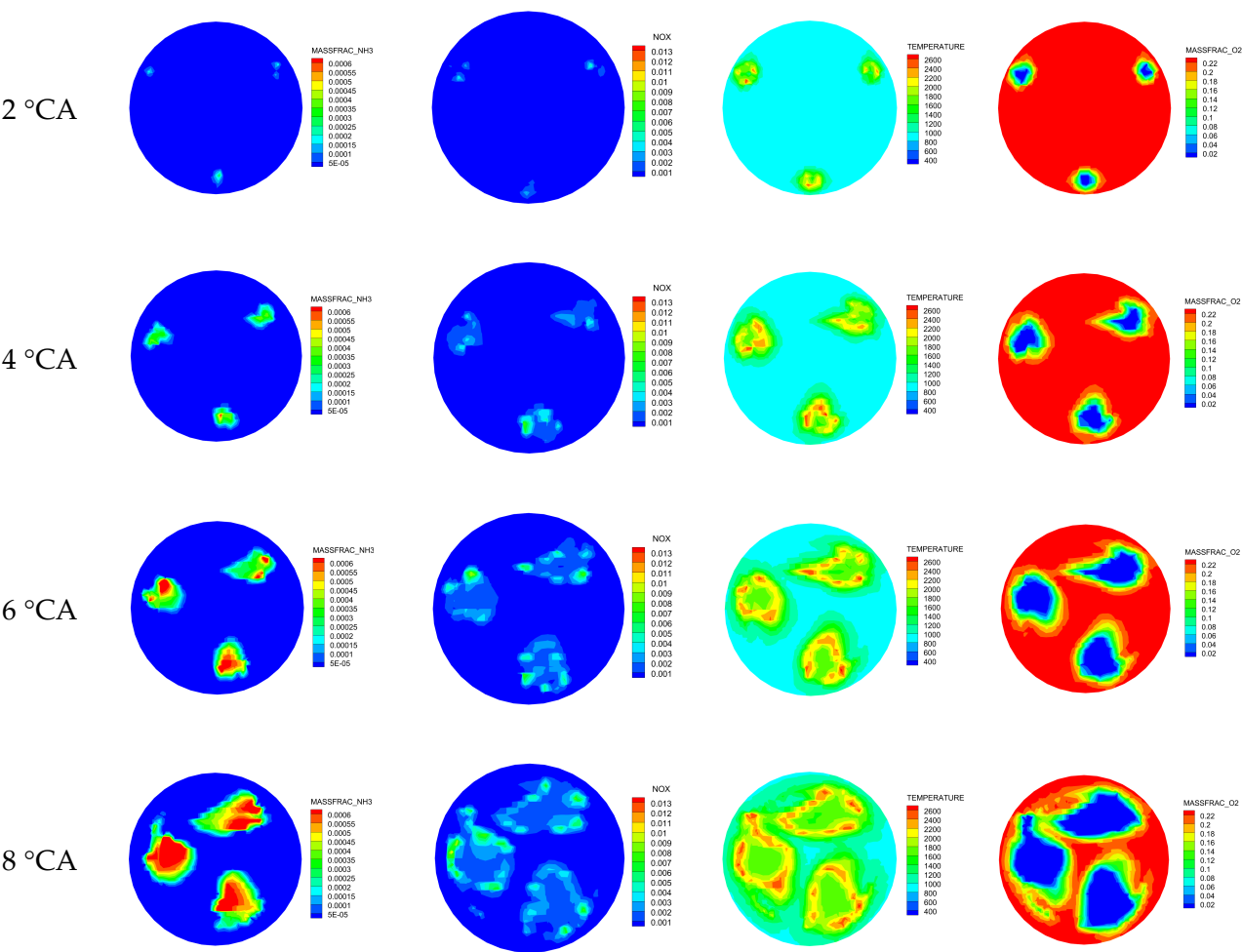


Figure 4. Cont.

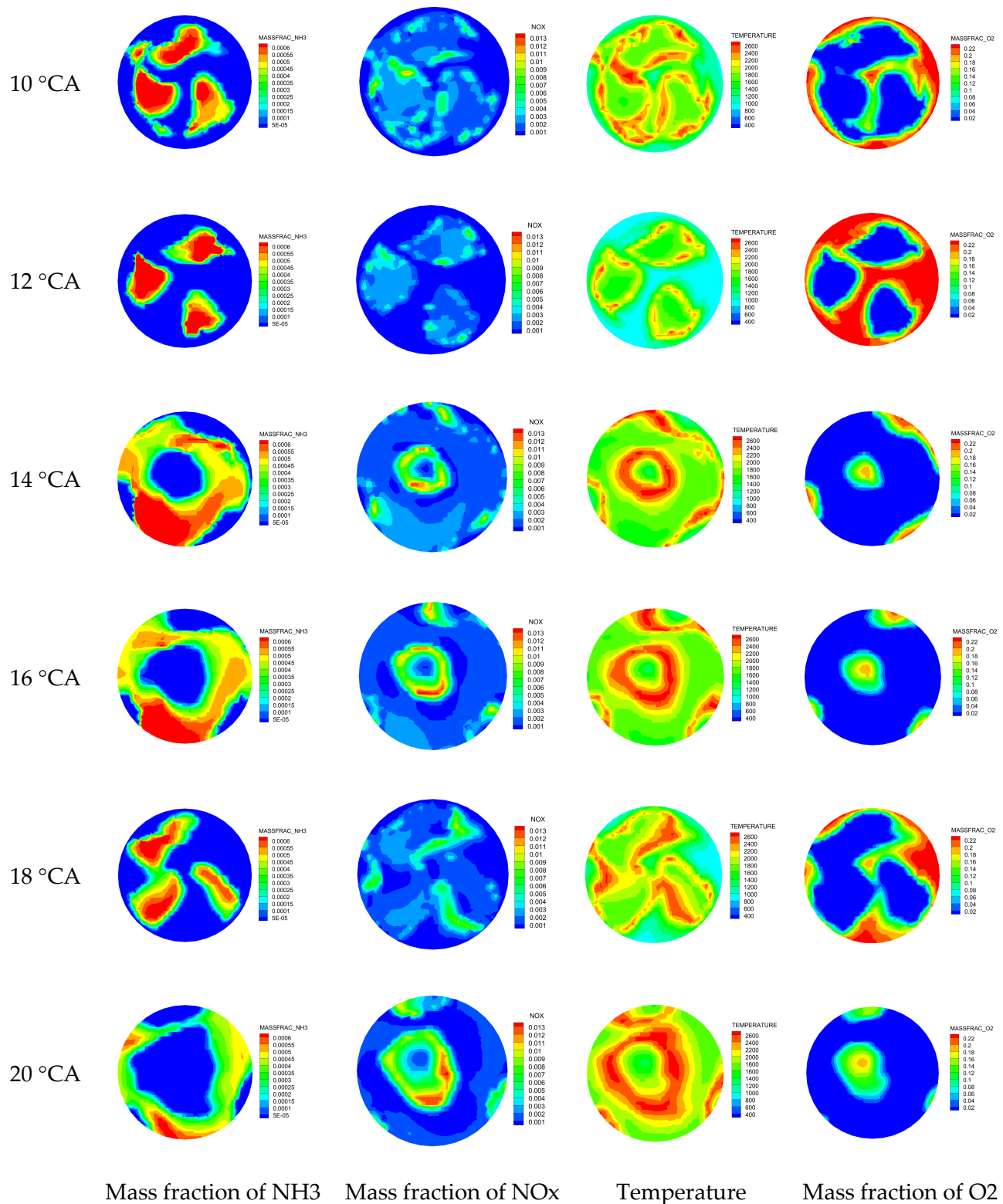


Figure 4. Changes in NH₃, NO_x, temperature, and O₂ in cylinder in case 4.

3.1. Impacts of AI on Combustion Processes and Emission Characteristics

Figure 5 depicts the impact of AI on pressure and HRR. As shown in Figure 5a, shifting the AI from 4 °CA to −4 °CA results in a substantial increase in P_{max} , which is mainly attributed to an elevated contribution of premixed ammonia combustion. Early ammonia injection into the cylinder causes premature ignition under the influence of diesel fuel, providing substantial ignition energy for the subsequent ammonia and natural gas

injections. Partial pre-ignition of ammonia reduces the peak of HRR, as shown in Figure 5b. For each 2 °CA increment in AI advance, the $P_{\max}$ increases by 0.78 MPa, and the phase of $P_{\max}$ shifts 0.71 °CA toward the top dead center (TDC). Due to the extended interval between ammonia and natural gas injection timings in case 1 and case 2, distinct secondary ignition phenomena appear in their heat release curves: the first ignition occurs with ammonia, followed by natural gas. In case 3, case 4, and case 5, where AI is close to or lags behind NGI, the ignition events of ammonia and natural gas occur nearly simultaneously. Since the natural gas flame propagation velocity exceeds that of ammonia, no distinct secondary ignition phenomenon appears in the heat release curves.

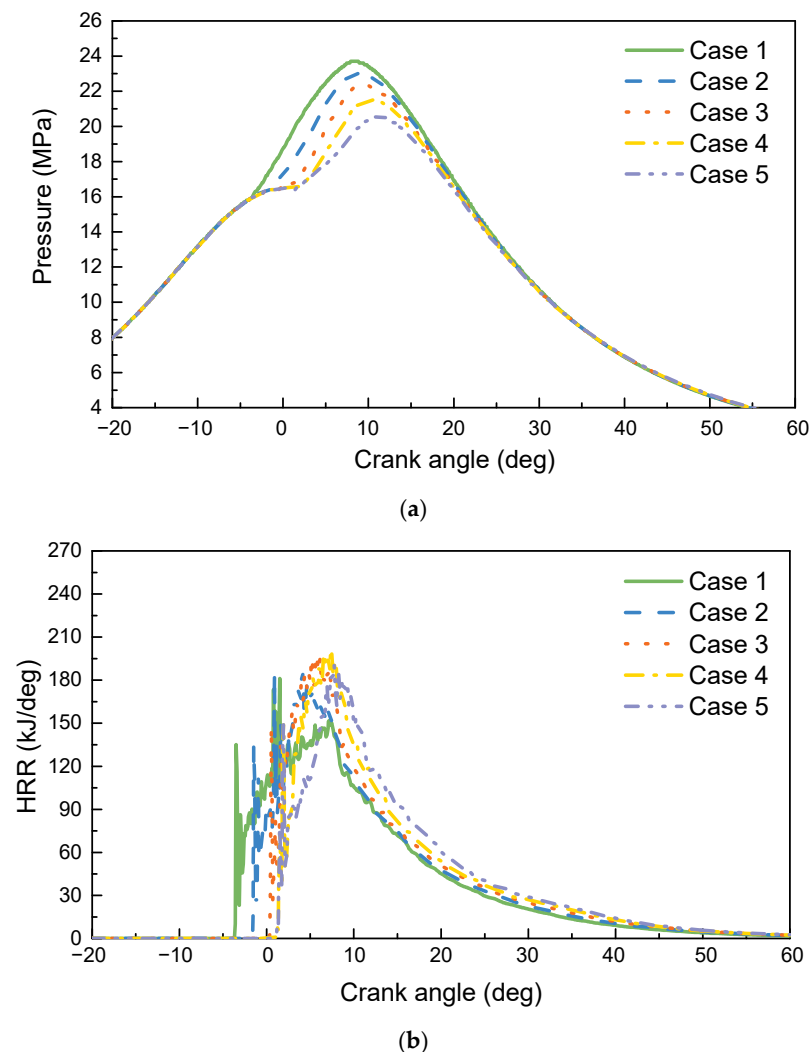


Figure 5. Impacts of AI on pressure and HRR. (a) Pressure within the cylinder. (b) HRR.

CA refers to the corresponding crankshaft angle, and 10% refers to the percentage of the cumulative heat release. CA50 defines the combustion phase. The interval from CA0 to CA10 represents the ignition delay, while the period spanning from CA10 to CA90 corresponds to the combustion duration [46,47]. Figure 6 depicts the combustion progress corresponding to various AI. Increasing the advancement of the AI results in a longer ignition delay, shifting both the ignition timing (CA10) and combustion phase toward pre-TDC. When ammonia is injected before or after the TDC, the combustion duration tends to increase with delayed injection. Notably, the increase in combustion duration is more pronounced when ammonia injection is delayed until after the TDC.

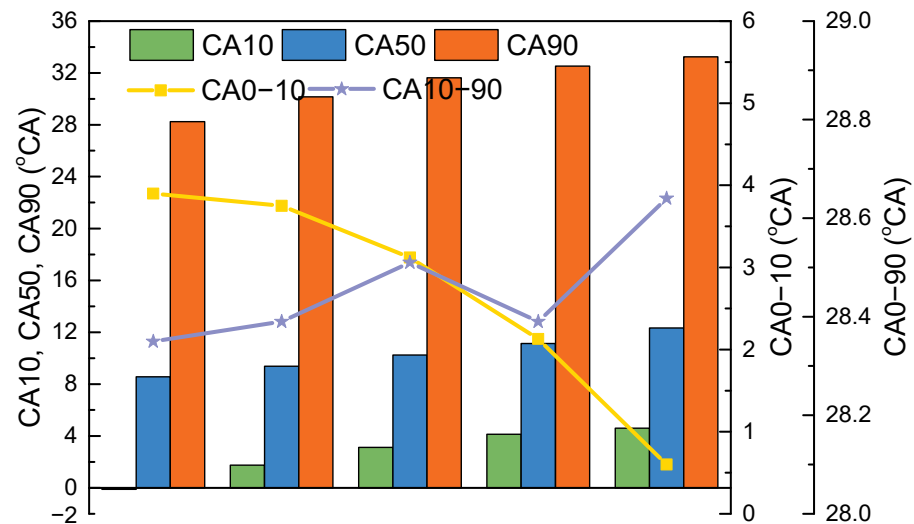


Figure 6. Impacts of AI on duration of combustion.

Figure 7 illustrates the effects of AI on ITE and indicated mean effective pressure (IMEP). It is important to note that IMEP refers to the total indicated mean effective pressure, excluding pumping losses, and measures only the work produced during the compression and expansion strokes. Both ITE and IMEP show an increasing trend as the AI is advanced. Specifically, ITE gradually increases from 47.73% to 50.62%, while IMEP rises from 2.04 MPa under case 5 to 2.15 MPa under case 1. This analysis indicates that, during the co-combustion of NH_3 and CH_4 in marine engines, early HPDI of ammonia into the cylinder enhances both thermal efficiency and IMEP.

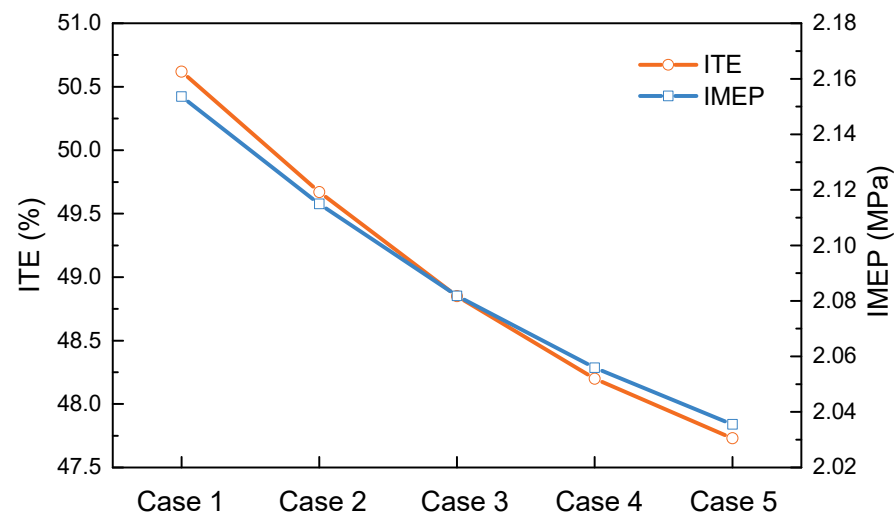


Figure 7. IMEP and ITE under different AIs.

When marine engines operate using co-combustion of NH_3 and CH_4 , the greenhouse effect primarily results from emissions of CO_2 , CH_4 , and N_2O . The global warming potential of CH_4 is 23 times that of CO_2 [48], while that of N_2O is 300 times greater [49]. Therefore, the greenhouse effect of unburned CH_4 is defined as the equivalent CO_2 emissions (unburned CH_4 multiplied by 23), and the greenhouse effect of N_2O is defined as the equivalent CO_2 emissions (the N_2O quantity multiplied by 300). The total equivalent CO_2 emissions of the engine are the sum of the equivalents CO_2 from methane, the equivalents CO_2 from N_2O , and the direct CO_2 emissions. Figure 8 illustrates the effects of AI on greenhouse gas and NO_x emissions from marine engines. The figure shows that early ammonia injection increases NO_x emissions from 2.54 g/(kW·h) under case 5 to 3.05 g/(kW·h) under

case 1, while remaining in compliance with the IMO Tier III emission standard limit of 3.4 g/(kW·h) [50]. Early ammonia injection has a negligible effect on CO₂ emissions, as CO₂ primarily originates from CH₄ combustion. Postponing the injection of ammonia results in a higher concentration of ammonia fuel within the cylinder during the latter stages of combustion, thereby reducing the combustion rate [51]. This reduces the co-combustion velocity between natural gas and ammonia, resulting in a slight increase in the equivalent CO₂ emissions from CH₄ combustion. In general, AI exerts a minimal impact on the overall equivalent CO₂ emissions of marine engines, which consistently approximate 250 g/(kW·h). When ammonia was injected before TDC, the total engine equivalent CO₂ emissions declined from 256.52 g/(kW·h) under case 3 to 249.94 g/(kW·h) under case 1. When ammonia was injected 2 °CA earlier before TDC, the total equivalent CO₂ emissions (from case 3 to case 1) decreased by 1.3%.

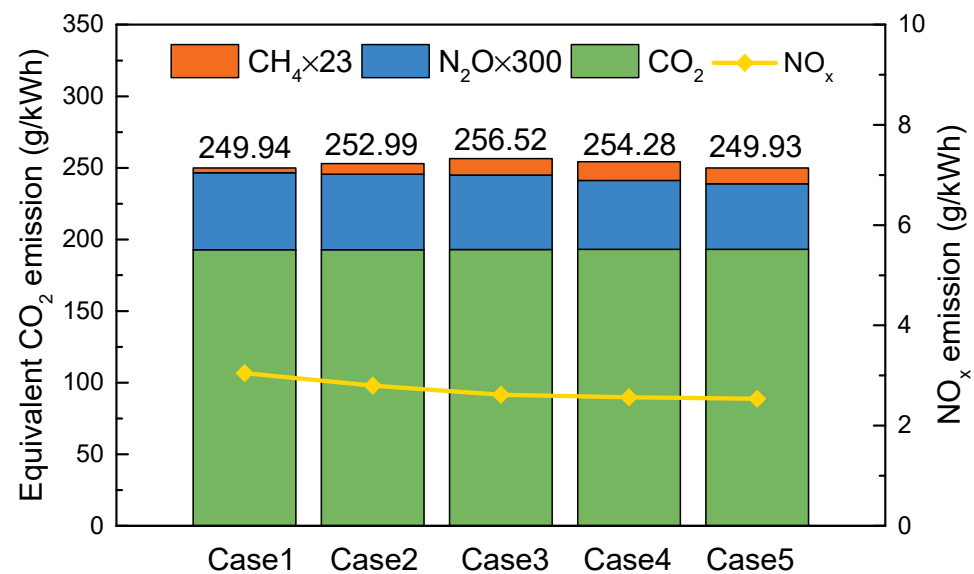


Figure 8. Emissions under different AIs.

3.2. Effects of NGI on Combustion and Emissions

In Figure 9a, advancing the NGI from 4 °CA to −4 °CA results in a significant increase in the P_{max} , primarily due to the combustion of a large proportion of premixed methane. Early injection of natural gas into the cylinder causes premature ignition under the influence of diesel, providing substantial ignition energy for the subsequent injection of ammonia and natural gas. The partial combustion of the early-injected natural gas reduces the peak of HRR, as shown in Figure 9b. For every 2 °CA advance of NGI, the P_{max} increases by 0.74 MPa, and the phase of P_{max} shifts by 0.69 °CA toward TDC. Dual ignition phenomena also occur under case 6 and case 7. Due to the significant time gap between NGI and AI in these cases, their heat release curves distinctly show two ignition events: the first from natural gas ignition and the second from ammonia ignition. In case 7, case 8, and case 9, since the NGI is close to or lags behind AI, the ignition times of ammonia and natural gas are similar. Consequently, no distinct secondary ignition phenomena are observed in the curves of HRR.

Figure 10 depicts the combustion progress corresponding to various NGI. As shown, advancing the NGI increases the ignition delay, causing both the ignition time and combustion phase to shift toward the pre-TDC. When the NGI is delayed, the combustion duration initially exhibits a decrease, followed by a subsequent increase.

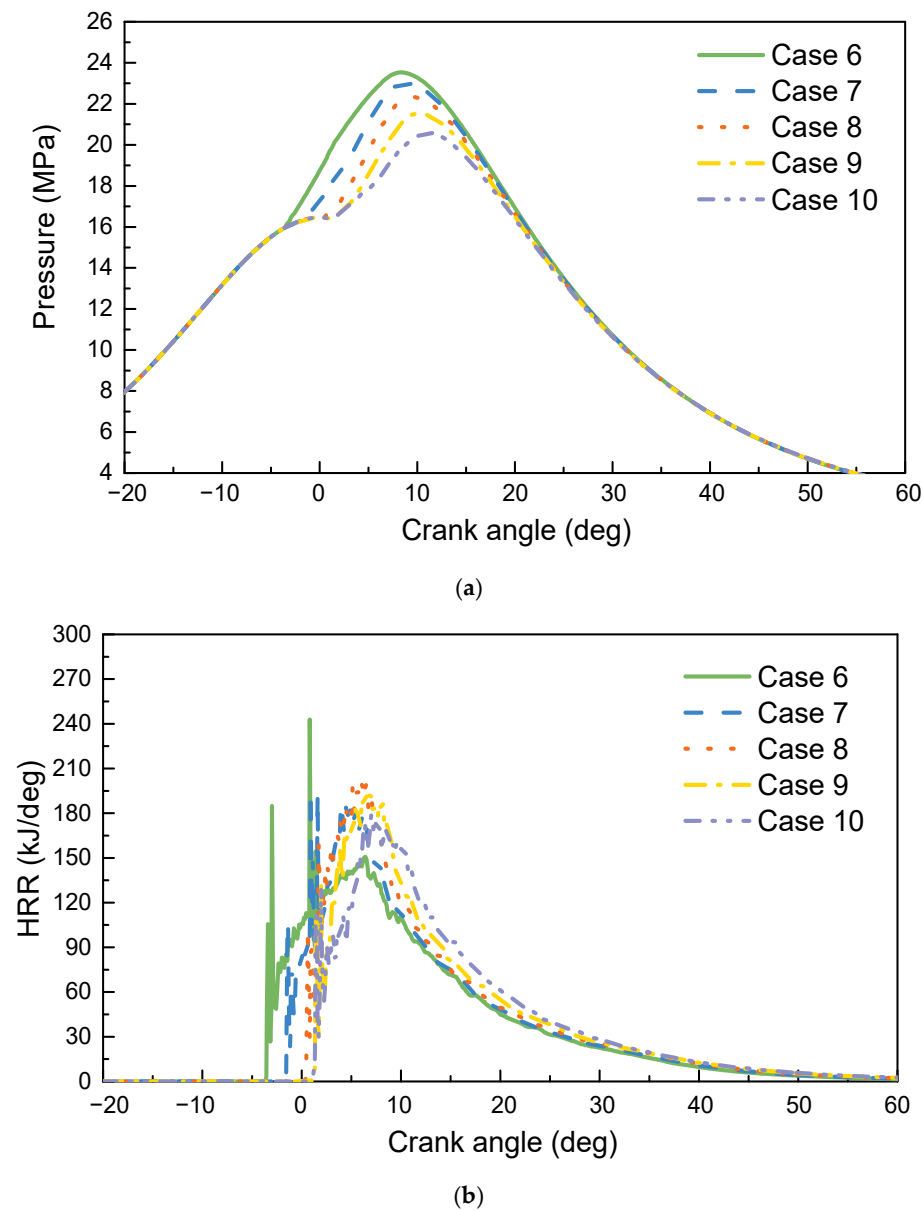


Figure 9. Impacts of NGI on pressure and HRR. (a) Pressure within the cylinder. (b) HRR.

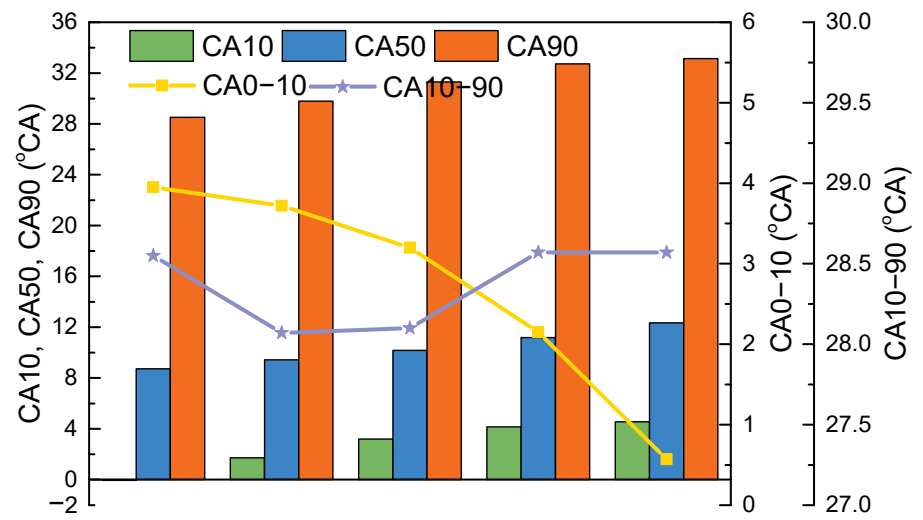


Figure 10. Impacts of NGI on duration of combustion.

Figure 11 illustrates the effects of NGI on ITE and IMEP. Similar to the trend observed with AI, both ITE and IMEP increase as the NGI is advanced. ITE gradually rises from 47.81% under case 10 to 50.54% under case 6, while IMEP increases from 2.03 MPa in case 10 to 2.15 MPa in case 6. During the co-combustion in marine engines, advancing the HPDI of natural gas into the cylinder enhances both thermal efficiency and IMEP.

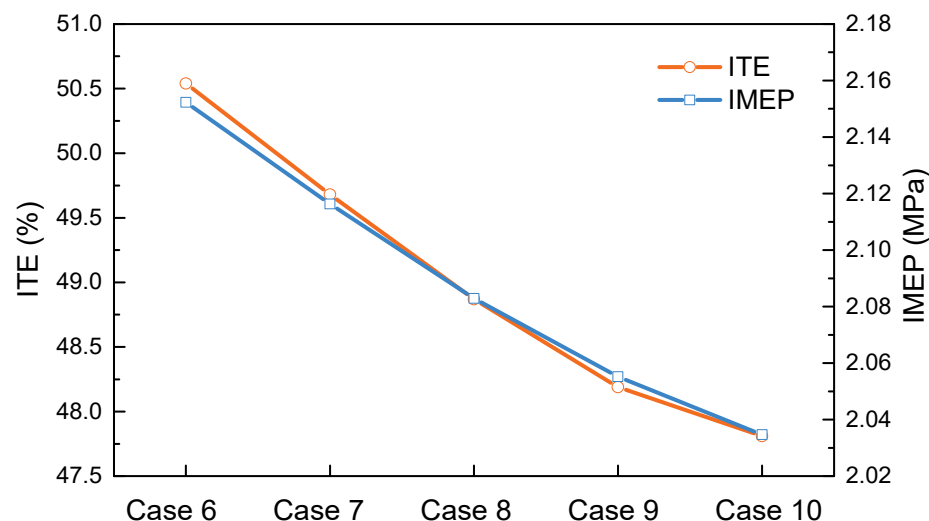


Figure 11. IMEP and ITE under different NGIs.

Figure 12 illustrates the effect of NGI on greenhouse gas and NO_x emissions from marine engines. As shown, advancing the NGI increases NO_x emissions from 2.43 g/(kW·h) under case 10 to 3.17 g/(kW·h) under case 6, with both values remaining below the Tier III emission standard imposes a maximum limit of 3.4 g/(kW·h). The effect of advancing NGI on N_2O emissions is negligible, while equivalent CO_2 emissions from CH_4 gradually decrease. Overall, total engine equivalent CO_2 emissions decline from 256.52 g/(kW·h) under case 10 to 244.19 g/(kW·h) under case 6.

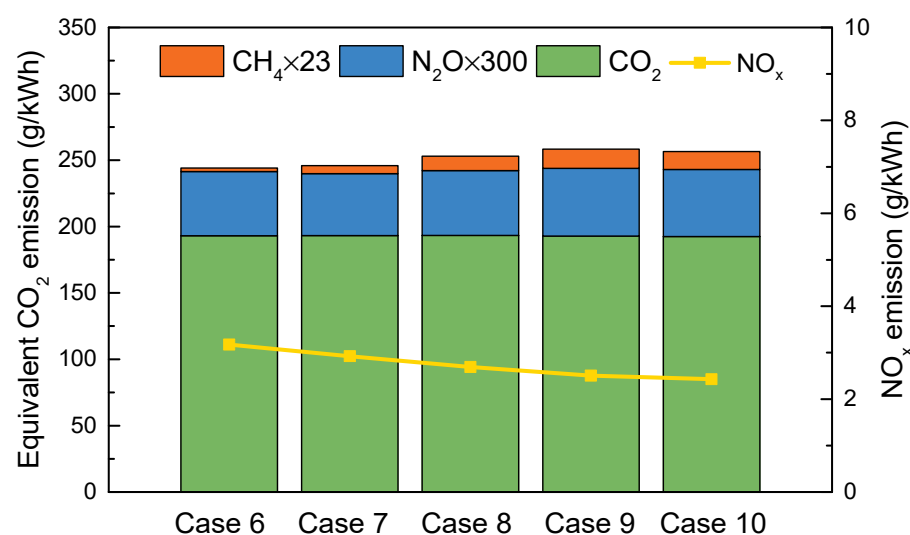


Figure 12. Emissions under different NGIs.

4. Conclusions

When ammonia and natural gas were injected before the TDC, both fuels exhibited pronounced secondary ignition phenomena within the cylinder. Under both ammonia and natural gas injection strategies, cylinder pressure and HRR were more sensitive to the AI.

For each 2 °CA increment in AI's advance, the $P_{\max}$ increased by 0.78 MPa, and phase of $P_{\max}$ shifted 0.71 °CA closer to TDC. For natural gas, each 2 °CA advance in injection timing increased the $P_{\max}$ by 0.74 MPa and shifted the phase of $P_{\max}$ toward the TDC by 0.69 °CA. Advancing HPDI of both NH_3 and CH_4 shortened the combustion duration. Their effects on ITE and IMEP of marine engine followed similar patterns, resulting in higher ITE and IMEP. The maximum ITE reached 50.62%, and the maximum IMEP was 2.15 MPa. Under all ammonia and natural gas injection strategies, NO_x emissions complied with IMO Tier III limits. When ammonia was injected 2 °CA earlier during high-pressure in-cylinder injection before TDC, equivalent CO_2 emissions decreased by 1.3%. Similarly, when natural gas was injected 2 °CA earlier under the same conditions, equivalent CO_2 emissions decreased by 1.7%.

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