

The impact of ammonia-diesel blend combustion on a cleaner environment

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ABSTRACT

Ammonia (NH₃) has emerged as a potential carbon-free additive and alternative fuel component for internal combustion engines. When blended with diesel in compression-ignition (CI) engines, ammonia can enhance the combustion process, increase engine efficiency, and reduce carbon-based emissions. The use of ammonia as a fuel for dual-fuel combustion engines is being extensively researched in the field of internal combustion engines. This study investigated the effects of adding ammonia to diesel fuel in a boosted compression ignition engine, with ammonia proportions ranging from 10% to 90% by mass. A computational fluid dynamics (CFD) model was developed using professional GT-Power software. In this setup, ammonia was injected into the intake manifold, while diesel fuel was injected directly into the cylinder. The main output parameters measured included engine torque, power, brake thermal efficiency, and emissions of NO_x, CO, CO₂, and hydrocarbons (HC). The results of the study demonstrate that using ammonia as a replacement fuel at levels up to 20% can significantly reduce carbon-based engine emissions without compromising engine performance, thereby improving environmental conditions.

Keywords: computational fluid dynamics, zero carbon fuel, combustion, emissions, environment.

INTRODUCTION

The alternative fuels approach for conventional combustion engines has been investigated over several years. Researchers in the automobile field have made remarkable strides in significantly reducing the exhaust emissions produced by diesel engines. This progress has largely been fueled by growing global awareness and concern regarding the detrimental effects of air pollution on public health and the environment. However, the anticipated increase in the number of diesel vehicles poses a potential risk of revisiting air quality issues in the future. In response to this challenge, numerous research studies have been conducted to explore the possibility of replacing traditional diesel fuel with oxygenated alternatives such as methanol, hydrogen, ammonia, ethanol, and butanol. Adopting these alternative fuels in internal combustion engines presents a

promising opportunity to mitigate our reliance on petroleum-based fuels [1].

Ammonia (NH₃) has considered as a potential carbon-free or low-carbon additive and alternative fuel component for internal combustion engines. Ammonia has a reasonable energy density and is relatively easy to store. Additionally, there is a well-established infrastructure for its production, storage, handling, and distribution. However, ammonia is corrosive to certain materials, difficult to ignite due to its high octane number and auto-ignition temperature, and it has a low heating value. When blended with diesel in compression-ignition (CI) engines, ammonia can influence combustion chemistry, flame propagation, and emissions in ways that may help meet stringent regulatory targets. This introduction outlines the motivation, key physical and chemical mechanisms, and current research themes related to ammonia-diesel blends, with emphasis

on emission outcomes, operating strategies, and knowledge gaps. Moreover, ammonia as an additive or fuel component offers several attractive features: it contains no carbon, thus avoiding direct CO₂ from the fuel; it possesses high octane and low cetane, which can influence ignition timing and low-temperature chemistry; and it can participate in radical chemistry that affects oxidation pathways in the in-cylinder mixture [2].

Many researchers studied the gaseous fuels as an alternative fuel for combustion engines. They investigated the methane, methanol, hydrogen, and ethanol to reduce engine emissions and enhance efficiency [3, 4]. Other researchers have investigated different aspects of ammonia as a promising fuel for diesel engines. they considered the safety, materials, and fueling considerations for ammonia-diesel blends in heavy-duty engines [5, 6], a lifecycle assessment of ammonia-diesel pathways: environmental and economic perspectives [7], a combustion diagnostic for ammonia-diesel blends: laser and spectroscopic insights [8], and the pathways, challenges, and opportunities of ammonia in internal combustion engines [9].

Several studies have been conducted on the use of ammonia as a dual fuel or ammonia-containing additive in diesel engines. Smith et al. [10] studied the effect of ammonia–diesel dual-fuel combustion on engine emissions and efficiency using optimization strategies. Wardana et al. [11] investigated the effect of ammonia addition on NO_x and PM emissions for a heavy-duty diesel engine. Chen et al. [12] experimentally studied how to control NO_x emissions for a CI engine using different strategies. Wang et al., [13] considered the techniques to reduce or prevent the ammonia release NH₃ from an exhaust after-treatment system into the atmosphere for a diesel engine. Liu et al. [14] investigated the effect of ammonia-diesel blends for a dual engine under cold start conditions. Wang et al. [15] studied the feasibility and limitations of ammonia blending in smaller engines, with emphasis on transient response. Berwal et al. [16] made a comprehensive review on the practical application of ammonia as a future fuel for combustion.

The combustion of ammonia in dual-fuel engines falls into two categories – low-pressure and high-pressure modes – each defined by a different ammonia injection method [17–19]. In the low-pressure mode, gaseous ammonia is supplied to the cylinder through the intake manifold in a gaseous state to mix with air before being introduced

to the combustion chamber, while in the high-pressure mode, liquid ammonia is directly injected into the cylinder. Research has focused heavily on the low-pressure approach because it requires relatively minor engine redesign and modifications at a lower cost.

Zhou et al. [20] investigated the effect of in-cylinder reforming with gas recirculation on a pilot-diesel-ignition ammonia combustion engine. They introduced this strategy to simultaneously improve thermal efficiency and reduce emissions of unburned NH₃, NO_x, N₂O gases. Their results indicate that with a 3% diesel energy share and low engine speed, the engine's indicated thermal efficiency increased by 15.8%, while unburned NH₃ and N₂O emissions were reduced by 89.3% and 91.2%, respectively. Mi et al. [21] investigated the potential of multiple diesel injection strategies on unburned NH₃ reduction using ammonia-diesel dual fuel engine. They found that the proper use of the pre-main injection strategy can reduce the unburned ammonia from around. Yiqiang et al. [22] studied the effects of the liquid phase of ammonia and ammonia energy share (AES) on combustion, emissions, and engine performance compared to pure biodiesel operation. They concluded that higher AES significantly reduced the local cylinder temperature; therefore, a maximum AES of 50% was achieved. Increasing AES to 50% decreased combustion duration and combustion phasing. However, it deteriorated the indicated thermal efficiency (ITE) and reduced NO_x, CO, and ammonia emissions from 8700 ppm to 4400 ppm under the operating condition of 1500 rpm, 10 bar IMEP, and 70% ammonia energetic ratio. Karl et al. [23] investigated the effect of the injection strategies on engine performance and important emissions like NH₃, NO/NO₂, and N₂O. They showed that the NO_x and N₂O emissions were found to have opposite trends, where the highest NO_x and lowest N₂O concentrations were achieved for the operating points with the highest combustion efficiency. Delaying the combustion phasing improved the combustion efficiency for the 40% and 50% ammonia energy share cases.

Grannell et al. [24] studied ammonia combustion in spark-ignition (SI) engines running on gasoline and found that ammonia can replace a significant portion of gasoline energy, but not entirely due to operational issues like low thermal efficiency. Mørch et al. [25] tested ammonia mixed with hydrogen in an SI engine and found that a 10% hydrogen addition yielded the highest

efficiency and mean effective pressure. Westlye et al. [26] examined NO emissions from a similar engine fueled with 20 vol% hydrogen and 80 vol% ammonia, concluding that NO formation is lower with stoichiometric combustion compared to gasoline. However, leaner mixtures resulted in higher NO emissions. Nitrous oxide (N_2O) emissions, which have a global warming potential 265 times greater than CO_2 over 100 years, also increased with lean mixtures [27]. Zhang et al. [28] conducted an experimental study on ammonia and diesel combustion in a low-speed, two-stroke (CI) engine operating in High-Pressure mode for marine applications. Their findings indicated that as ammonia energy share AES increased, emissions of total hydrocarbons (THC), carbon monoxide (CO), and soot decreased, while nitrogen oxides (NOx) emissions rose. Yousefi et al. [29] investigated the effect of ammonia energy share AES up to 40% and diesel injection timing on the combustion performance of ammonia and diesel in a CI engine in RCCI mode. They found that increasing the AES caused overall poorer combustion characteristics due to the high auto-ignition temperature and low flame speed of ammonia, causing an increase in ammonia and N_2O emissions. However, the combustion efficiency is about 65%. Similar findings were made by Nadimi et al. [30], where for increasing AES, the unburned ammonia concentration in the exhaust increased significantly, and simultaneously decreased the N_2O emissions. Chen et al. [31] explored how the ammonia energy ratio (AER) and the diesel pilot-injection strategy influence the performance of dual-fuel engines. Their findings indicated that as the AER increased, there was a rise in unburned NH_3 and N_2O emissions, which resulted in decreased indicated thermal efficiency. Additionally, greenhouse gas (GHG) emissions were notably higher compared to those in the conventional diesel-only combustion mode. Moreover, they stated that, at an AER of 50%, the ITE reached an optimum value of 47.1% at a DPR of 40%. Compared to the diesel-only combustion mode, the ITE increased by 0.6%, GHG emissions decreased by 12.5%, and NO emissions decreased by 41.6%.

MODEL DESCRIPTION

A six-cylinder boosted compression ignition engine model is generated using GT-power

professional code. Ammonia is injected into the intake port in a gaseous state, and diesel fuel is injected directly into the cylinder. The ammonia percentage was varied from 10–90% by mass. The diesel injection timing is kept constant ($-5^\circ CA$) for all simulation runs. The simulation was conducted with a 3000 rpm engine speed and a stoichiometric Air/Fuel ratio to the total mixture. The total amount of fuel introduced to the cylinder is 150 mg/cycle for both fuels (NH_3 and diesel). In addition, a simulation was performed to neat diesel fuel to compare the results with the ammonia/diesel mixture. The engine specifications are presented in Table 1. The engine's initial conditions are presented in Table 2. The operation conditions are illustrated in Table 3. The specifications for

Table 1. Engine specifications

Parameter	Unit	Value
Bore	mm	119
Stroke	mm	175
Connecting rod length	mm	300
Piston pin offset	mm	0
Displacement/cylinder	L	1.94635
Total displacement	L	11.6781
Number of cylinders		6
Compression ratio		17
Bore/stroke		0.68
IVC	[CA]	-118
EVO	[CA]	100
IVO	[CA]	314
EVC	[CA]	400

Table 2. Initial conditions

Parameter	Unit	Value
Initial pressure	bar	1.9
Initial temperature	K	310
Head temperature	K	570
Piston temperature	K	600
Cylinder temperature	K	480

Table 3. Operation conditions

Parameters	Unit	Value
Engine speed	rpm	300
Start of combustion	$^\circ CA$	-1.6
Start of injection	$^\circ CA$	-5
Vol. eff. ref. pressure	bar	1.78
Vol. eff. ref. temperature	K	318
Mean piston velocity	m/s	17.5

Table 4. Fuel specifications

Chemical formula	Ammonia NH ₃	Diesel C _{13.5} H _{23.6}
Molecular weight (kg/kmol)	15	185.6
Lower heating value (MJ/kg)	18.56	43.25
Critical temperature (K)	405	569.4
Critical pressure (bar)	113.6	24.6
Carbon atoms per molecular	0	13.5
Hydrogen atoms per molecular	3	23.6
Nitrogen atoms per molecular	1	0
Stoichiometric air-fuel ratio	6.04	14.33
Auto ignition temperature (K)	903	529
Flame laminar speed (cm/s)	6-8	~ 100
Ignition limit (vol.)%	15.8-28.0	1.0-6.0
Latent heat of vaporization [kJ/kg]	1370.0	232.4
Fuel liquid density [kg/m ³]	602.8	832.0

methanol and diesel fuel are presented in Table 4. The model was validated using the data available from the literature, and a good agreement was obtained. The sub-models used in the simulation are presented in Table 5.

The program uses the ‘WoschniGT’ model for heat transfer modeling. The wall temperature model is used to impose temperatures for the head, piston, and cylinder walls. ‘EngCylTWall’ is used to calculate the cylinder chamber wall temperatures. To use this model, the user must define additional cylinder geometry and impose boundary conditions such as coolant and oil temperatures. The code then solves for the wall temperatures using a finite element method. This option also allows the user to see the temperature gradients throughout the cylinder, piston, valves, and head walls. If temperature measurements are not available for the cylinder, the following temperatures are typical: head temperature 550–600 K, piston temperature 550–600 K, and cylinder temperature 400 K.

The combustion model used in the simulation is ‘InjMultiProfileconn’ object. This template allows for multiple fuels combustion. The combustion process can be calculated using four different parameters to achieve the best match to experimental data. The direct injection diesel fuel process is simulated using ‘EngCylCombDIPulse’ object. The injector is connected directly to the cylinder. The injected fluid’s heat of vaporization will be taken into account as the combustion energy is released. To model the evaporation of the fuel in the cylinder, an ‘EngCylEvaporation’

Table 5. Sub-models used in the simulation

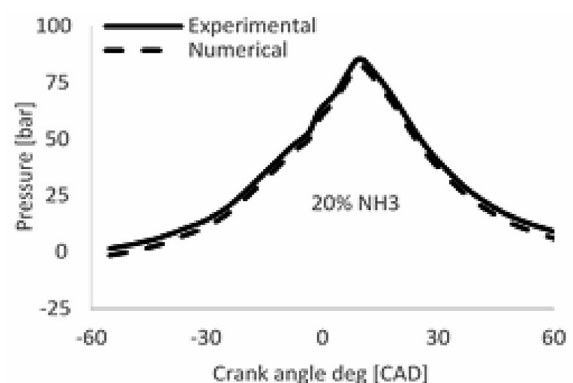
Turbulence model	K-ε turbulence model
Turbulent dispersion model	Turbulent kinetic energy and dissipation
Wall interaction model	Spray wall interaction model
Breakup model	KHRT
Heat transfer model	WoschniGT
Evaporation model	Langmuir-Knudsen
Combustion model	EngCylCombDIPulse
NOx model	Extended Zeldovich
Soot model	Kinetic model

reference object has to be specified in the ‘Evaporation Object’ attribute. The evaporation of the liquid fuel after the initial vaporization will be controlled by the attributes specified in the ‘EngCylEvaporation’ object. Ammonia fuel is injected to the manifold port using ‘InjRateConn’ depends on ammonia mass flow rate, pressure, and temperature.

The simulation model has been calibrated and validated using available data from the literature. In this paper, the cylinder pressure was compared with reference [31]. Following the calibration process, the experimental results showed good agreement with the simulation outcomes.

MODEL VALIDATION

The simulation model was validated using available data from the literature. In this paper, the cylinder pressure versus crank angle and 20% NH₃ percentage was compared and validated using reference [31]. As seen in Figure 1, a good agreement was obtained, indicating that the proposed model can be used for further investigation.


Fig. 1. A comparison of cylinder pressure between experimental data and numerical simulations

RESULTS AND DISCUSSIONS

The effect of various ammonia/diesel ratios has been investigated and discussed. Brake power, brake mean effective pressure, brake thermal efficiency, and combustion efficiency are calculated and discussed. Additionally, cylinder pressure, cylinder temperature, and heat release rate are also presented in the simulation.

In-cylinder pressure

The relationship between cylinder pressure and the ammonia/diesel ratio is illustrated in Figure 2. The data shows that as the percentage of ammonia in the mixture increases, the peak cylinder pressure decreases compared to combustion with neat diesel fuel. Specifically, when the mixture contains 90% ammonia, the cylinder pressure decreases by approximately 30%. This reduction is attributed to the lower combustion efficiency of ammonia, which has a lower laminar flame speed compared to diesel fuel. Moreover, Ammonia has a very low cetane number (difficult to auto-ignite), so it resists ignition compared with diesel.

Heat release rate (HRR)

The effect of ammonia addition on heat release rate is presented in Figure 3. As seen, with the very high amount of ammonia in the mixture the heat release decreased significantly up to 50% compared to neat diesel combustion, resulting in poor combustion and deterioration of engine efficiency.

Moreover, the maximum instantaneous HRR usually decreases as the ammonia fraction increases because the ammonia burns more slowly and contributes less rapid heat release near TDC. Additionally, small NH_3 energy share (less than 20%) results in small reduction in peak HRR and slight retardation; premixed fraction remains largely diesel-dominated. However, moderate NH_3 share between (20–50%) causes noticeable drop in peak HRR, larger retardation of peak phasing (several degrees CA), premixed fraction falls, and diffusion/late burn contribution increases. Conversely, when the NH_3 share is very high (more than 50%), a significant reduction in peak HRR, an increase in late combustion, and misfires or incomplete combustion without retuning are observed.

Cylinder temperature

The effect of ammonia addition on cylinder temperature is illustrated in Figure 4. It can be observed that the cylinder temperature decreases by about 5% when the amount of ammonia NH_3 less than 20%, and up to 35% when the NH_3 exceeds 80%. This indicates that ammonia is less reactive and burns more slowly than diesel, resulting in a reduction or delay in peak heat release. Consequently, this leads to lower peak in-cylinder temperatures. Additionally, because ammonia has a lower energy density on a mass basis compared to diesel, it provides less chemical energy for the same mass of fuel. If the total amount of fuel is not increased (as in this study), this results in a lower total heat release and, therefore, decreases temperatures.

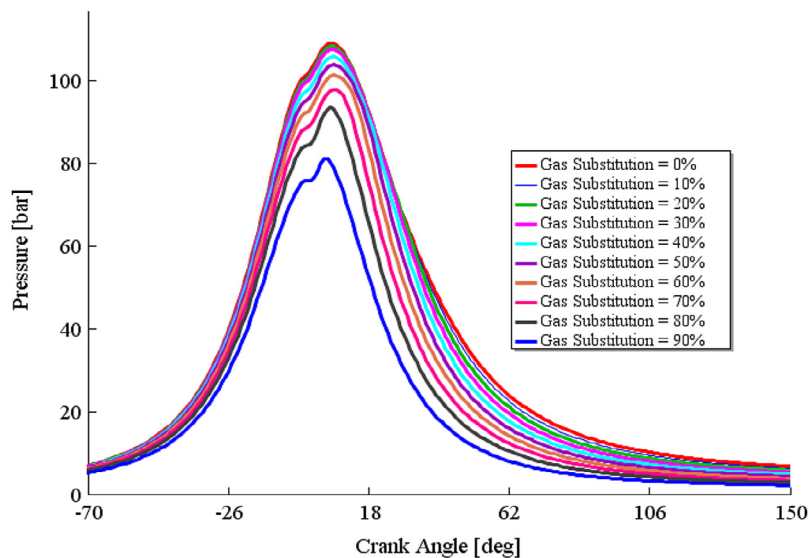


Fig. 2. Variation of cylinder pressure versus crank angle for different NH_3 /diesel ratios

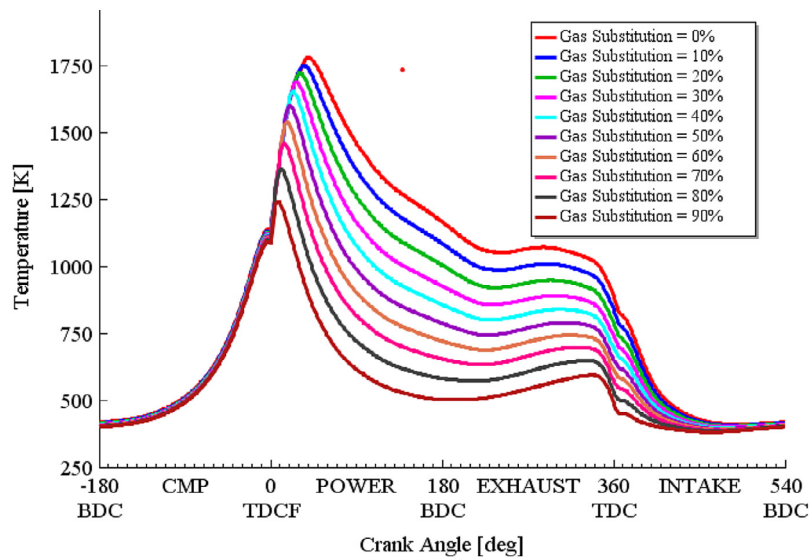


Fig. 3. Variation of HRR versus crank angle for different NH_3 /diesel ratios

Brake thermal efficiency, brake power, brake mean effective pressure, and combustion efficiency

The effect of NH_3 addition on engine key performances is present in Table 6. The brake power, brake thermal efficiency, brake mean effective pressure, and combustion efficiency are presented in the table for different ammonia/diesel ratio. Below these parameters are discussed in detail.

Brake power

The engine brake power (BP) is calculated using the following equation:

$$BP = \frac{bmep \times V_{disp} \times rpm}{n_r} [1/600] \quad (1)$$

where: $bmep$ – brake effective pressure, V_{disp} – engine displacement, n_r – revolutions per cycle ($n_r = 1$ for 2-stroke, $n_r = 2$ for 4-stroke).

As seen from the Table 6, when the concentration of NH_3 in the mixture exceeds 30%, the engine's brake power decreases by approximately 31% compared to neat diesel fuel combustion. Moreover, when the NH_3 content rises above 50%, the BP decreases significantly, and at very high NH_3 ratios, the engine power experiences serious deterioration. This decline in performance is attributed to very low cylinder pressure caused by misfiring and incomplete combustion.

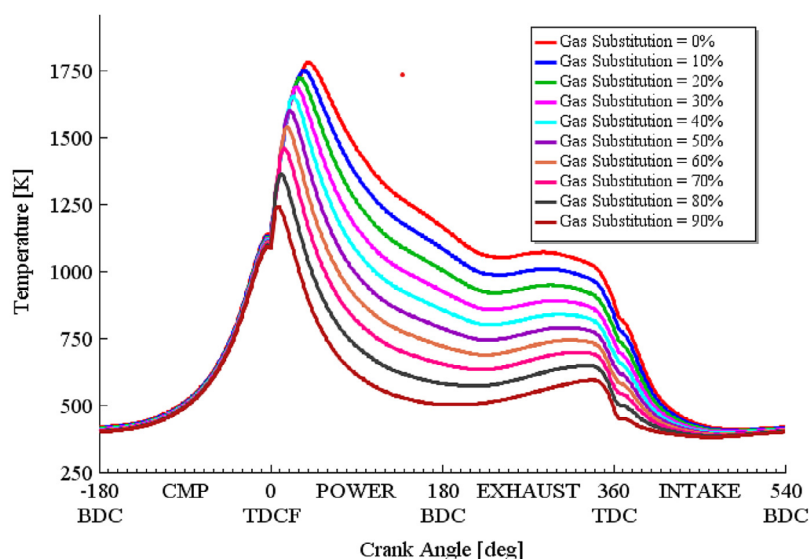


Fig. 4. Variation of cylinder temperature versus crank angle for different NH_3 /diesel ratios

Table 6. Engine key performance versus NH₃ ratios

Parameter	Ammonia substitution [%]									
	0	10	20	30	40	50	60	70	80	90
Break thermal efficiency [%]	26.61	26.83	26.48	25.68	24.06	21.12	16.1	7.38	-8.9	-44.6
Break power [kW]	240.3	224.9	194.7	167.3	136.6	102.2	64.4	23.3	-20	-66.4
BMEP [bar]	8.2	7.4	6.6	5.7	4.6	3.5	2.2	0.79	-0.7	-2.2
Combustion efficiency [%]	99.2	99.6	99.3	99.2	98.7	97.6	95.5	90.9	71.2	58.1

Brake thermal efficiency

The brake thermal efficiency (BTE) is calculated according to the following equation:

$$BTE = \frac{3600}{BSFC \times LHV} \times 100 \quad (2)$$

where: *LHV* – lower heating value of the fuel,
BSFC – brake specific fuel consumption which is calculated according to the following equation:

$$BSFC = \frac{\dot{m}_f \times 10^6}{BP} \quad (3)$$

where: $\dot{m}_f$ – fuel mass flow rate.

The impact of injection timing on brake power, thermal efficiency, brake mean effective pressure, and combustion efficiency is illustrated in Table 6. As seen, when the ammonia ratio is more than 50% per mass in the mixture, the BTE decreased up to 25% compared to neat diesel fuel combustion. Moreover, the ammonia ratio increased over 50%, the BTE has very low or negative value indicating misfiring or incomplete combustion. Similarly, As the ammonia ratio increased more than 50% the BP decreased about 60% compared to pure diesel fuel burning. And for a higher ammonia ratio, the BP decreased drastically, due to engine misfiring or poor combustion.

Table 6 also shows that when the ammonia NH₃ ratio exceeds 50%, the engine experiences significantly reduced or even negative BMEP. This condition indicates misfiring or incomplete combustion, which results in a loss of engine power and thermal efficiency. Negative BMEP in an ammonia/diesel dual-fuel cylinder signifies that the cylinder is consuming more energy than it produces, often due to poor or delayed combustion of ammonia. Ammonia has a low flame speed and requires a high ignition energy, making it more difficult to ignite and causing it to burn more slowly than diesel. If the ammonia/air mixture fails to ignite adequately during the expansion stroke, the effective work done during

this phase may be reduced or delayed beyond the useful portion of the stroke. Furthermore, if combustion occurs too late (after top dead center), the effective expansion pressure contributing to shaft work will be minimal or even negative, particularly in relation to pumping and friction, resulting in negative net engine work. Typically, engines using this dual-fuel system rely on a small quantity of diesel (the pilot fuel) to ignite the ammonia/air mixture. If the amount or timing of the pilot fuel is insufficient, ignition may be weak or inconsistent, resulting in misfires or partial combustion.

Engine emissions

NO_x emissions

In this research, the effect of methanol-diesel mixture, diesel injection timing, and engine speed on NO_x, CO, HC, and CO₂ emissions is investigated. Figure 5 illustrates the impact of the ammonia-to-diesel ratio on NO emissions. It shows that the presence of NH₃ has a significant influence on NO_x emissions. In a dual-fuel engine, diesel is injected to ignite the ammonia-air mixture, and the ratio of NH₃ to diesel is crucial in controlling the combustion process, including the peak temperature and pressure, which directly affect NO_x formation. Additionally, the figure indicates that when the NH₃/diesel ratio is less than 50%, the changes in NO_x emissions are minimal compared to combustion using pure diesel fuel. However, at higher NH₃/diesel ratios, NO_x emissions are significantly reduced about 10–70% due to lower peak temperatures and incomplete combustion. Therefore, the NH₃/diesel ratio should be carefully optimized to balance combustion efficiency and NO_x emissions.

CO emissions

Figure 6 presents the effect of the NH₃/diesel ratio on CO emission. As presented, when the NH₃ amount increased in the fuel mixture, the CO emission decreased. For 50% NH₃ in the

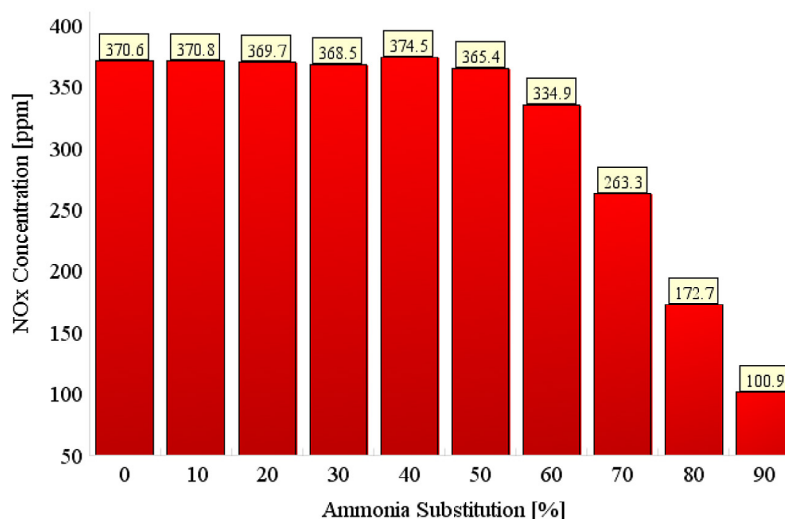


Fig. 5. Variation of NO_x concentration versus NH₃/diesel ratios

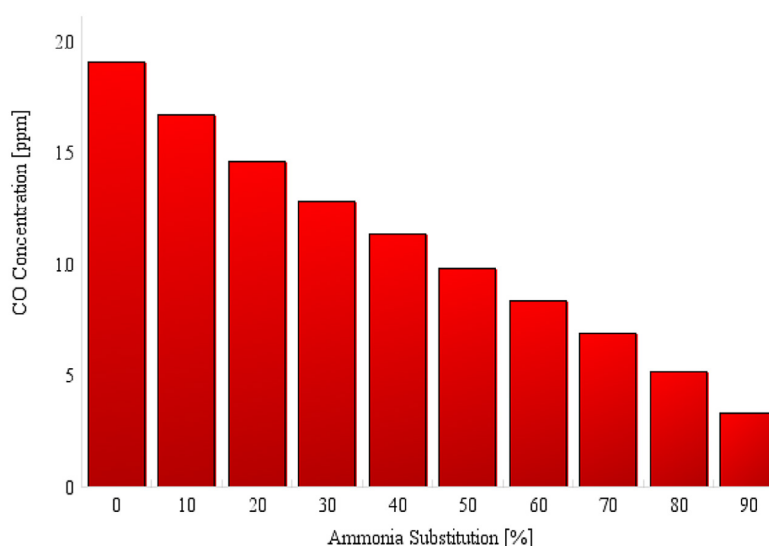


Fig. 6. Variation of CO concentration versus NH₃/diesel ratios

mixture the reduction is about 40% compared to pure diesel combustion. Ammonia is a zero-carbon fuel, so its contribution to combustion inherently doesn't produce CO, unlike diesel. Adding ammonia NH₃ to diesel fuel in dual-fuel engines can significantly reduce CO emissions, due to ammonia's low carbon content and its ability to promote more complete combustion of the hydrocarbon fuel. However, the exact effect on CO depends on factors like the ammonia-to-diesel energy ratio, diesel injection timing, and engine load, which can also influence the formation of other engine emissions.

HC emissions

Figure 7 illustrates the effects of adding ammonia to diesel fuel. Ammonia, being a low-reactivity

fuel, can result in incomplete combustion, leading to higher levels of unburned hydrocarbons and unburned ammonia. However, the impact of ammonia on combustion is influenced by several factors, such as the method of ammonia introduction, its proportion in the mixture, combustion temperature, mixture homogeneity, and injection strategy.

Due to combustion quenching, certain fuel/air mixtures can create pockets that hinder proper oxidation. This, combined with low cylinder temperatures, incomplete combustion, and misfiring, contributes to increased hydrocarbon emissions. As indicated in Figure 7, when the ammonia content exceeds 40% in the mixture, hydrocarbon emissions significantly rise compared to pure diesel fuel. In cases of very high ammonia content, the increase in hydrocarbon emissions becomes

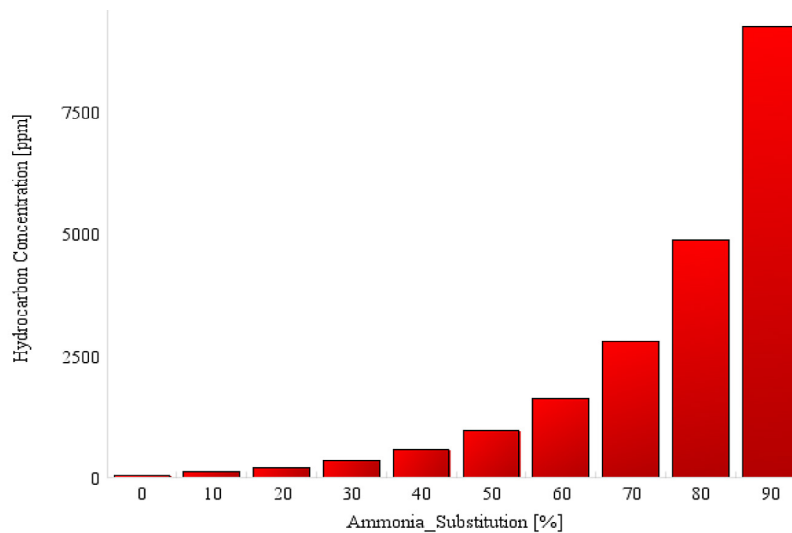


Fig. 7. Variation of HC concentration versus NH_3 /diesel ratio

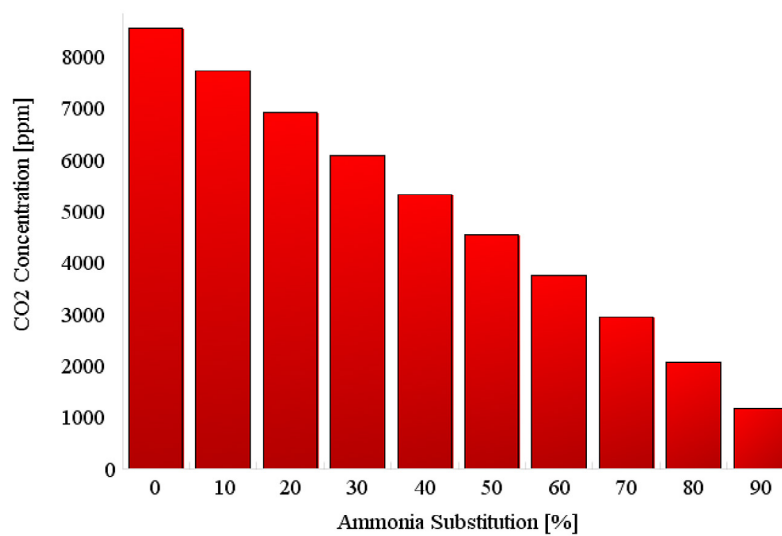


Fig. 8. Variation of CO_2 concentration versus NH_3 /diesel ratio

considerable and poses environmental concerns with an increased factor about 20 times compared to neat diesel combustion. This rise is attributed to the interplay of low cylinder temperatures, misfiring, and incomplete combustion.

CO_2 emissions

Figure 8 illustrates the impact of substituting diesel fuel with ammonia on CO_2 emissions. It clearly shows that using ammonia leads to a reduction in CO_2 emissions. Moreover, with the higher percentage of NH_3 in the mixture the CO_2 decreased significantly with a reduction factor about 7.5 times compared to neat diesel fuel. This is due to low carbon contents in the mixture and poor combustion. Furthermore, the lower emissions of CO_2 contribute to a more environmentally friendly fuel source.

CONCLUSIONS

This paper simulated the effects of ammonia addition to diesel fuel on engine combustion and emissions. The ammonia ratio is varied of 10–90% by mass. The ammonia is injected into the intake manifold, and the diesel fuel is injected into the cylinder. The total amount of fuel introduced to the cylinder is kept constant for all simulation runs. The diesel injection timing is kept constant (-5°CA) for all simulation runs. The research conclusion is as follows:

1. The cylinder pressure, temperature, and heat release rate showed the least reduction when the NH_3 /diesel ratio was below 40%. However, for higher NH_3 /diesel ratios, these parameters decreased significantly, negatively impacting

the combustion process within the cylinder.

2. At an NH_3 ratio of 10% per mass of total injected fuel, the BTE reached an optimum value of 26.73 %. In comparison with the diesel-only combustion mode, the BTE increased by 0.8%, and the combustion efficiency increased by up to 0.4%. For NH_3 ratios higher than 50% the engine performance decreased significantly due to poor combustion and high energy losses.
3. As the amount of NH_3 in the mixture increased, emissions of NO , CO , and CO_2 decreased. However, in cases of very high ammonia content, the increase in hydrocarbon emissions becomes considerable and poses environmental concerns, with an increase factor of about 20 times compared to neat diesel combustion. Nevertheless, at higher NH_3 ratios, the engine's performance deteriorated drastically. It is crucial to consider the trade-off between engine performance and emissions when adjusting the NH_3 /diesel ratio. Moreover, at very high NH_3 ratios, hydrocarbon emissions increased significantly due to incomplete combustion and misfiring.
4. There is no universal "best" ammonia/diesel ratio, as it depends on many factors such as engine type, operating conditions, ammonia supply method, diesel injection timing, and emissions/goals. In the literature, typically ranges from 5–40% ammonia by energy or 5–30% by volume (fuel-equivalent), with the rest provided by diesel pilot injection. The precise ratio should be determined through experimental testing and engine calibration. According to this research, the NH_3 /diesel ratio should not exceed 20% in the mixture to achieve an acceptable reduction in engine emissions while maintaining high engine performance compared to a pure diesel engine.

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