

NUMERICAL INVESTIGATION OF THE EFFECTS OF AMMONIA ENERGY SHARE ON POWER, TORQUE, AND EMISSION CHARACTERISTICS OF AN AMMONIA–DIESEL DUAL-FUEL ENGINE

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Abstract - This study investigates the effects of ammonia energy share on power, torque, and emissions of an ammonia–diesel dual-fuel engine using an AVL BOOST simulation model. The DESSUN 4DSP-75 diesel engine model was developed and validated under baseline diesel operation. The validated model was then converted to dual-fuel operation by supplying ammonia through the intake pipe, while diesel was directly injected into the cylinder as the pilot fuel. Ammonia energy shares of 5%, 10%, 15%, and 20% were investigated over the engine speed range of 1400-3000 rpm. The results show that increasing ammonia energy share decreases engine power and torque, while brake specific fuel consumption increases. This is mainly attributed to ammonia's lower heating value, higher auto-ignition temperature, and lower burning velocity compared with diesel. Soot emissions decrease significantly because ammonia contains no carbon, whereas CO and NO_x emissions increase due to slower combustion, incomplete oxidation, and possible fuel-NO_x formation.

Key words - Ammonia; combustion; dual-fuel engines; emissions; AVL BOOST

1. Introduction

In the context of the energy transition and increasingly stringent requirements for greenhouse gas emission reduction, internal combustion engines continue to play an important role in transportation, industry, marine applications, agriculture, and power generation owing to their high-power density, reliability, long-duration operation capability, and well-developed fuel infrastructure. Although vehicle electrification is being strongly promoted, the complete replacement of internal combustion engines in the short term still faces significant challenges, particularly in sectors requiring high power output, heavy-duty operation, and long operating ranges. Therefore, rather than completely eliminating internal combustion engines, research on improving efficiency, reducing fuel consumption, and lowering emissions remains a practically meaningful approach during the transition toward more sustainable energy systems [1].

In this direction, blended-fuel strategies have been widely investigated as a practical approach to improving engine performance and reducing emissions. Van-Trang Nguyen et al. [2] investigated a stationary diesel engine fueled with diesel-LPG blends using AVL BOOST. Their study showed that LPG addition improved engine power and reduced brake specific fuel consumption, while also decreasing CO and soot emissions. However, higher LPG ratios increased NO_x emissions, indicating that blended-

fuel strategies require a careful balance between performance improvement and emission trade-offs. Similarly, Dung and Dong [3] numerically investigated a reactivity-controlled compression ignition (RCCI) engine based on a direct-injection diesel engine using AVL BOOST, demonstrating the potential of dual-fuel combustion to improve combustion characteristics and engine performance while reducing emissions.

Among emission reduction strategies, the use of carbon-free alternative fuels has attracted increasing attention. Ammonia (NH₃) is a promising candidate because its molecules contain no carbon atoms. Therefore, ammonia utilization can inherently reduce direct CO₂ emissions compared with conventional hydrocarbon fuels. In addition to its role as a fuel, ammonia is also considered a promising hydrogen carrier due to its more favorable storage and transportation characteristics compared with hydrogen. Studies on future energy systems have also indicated that ammonia can be produced from renewable energy sources and used as a medium- to long-term energy storage medium, contributing to decarbonization in the power generation and transportation sectors [4]. In addition, compared with hydrogen, ammonia offers advantages in terms of its relatively well-developed production, storage, and distribution infrastructure. It can also be stored under lower-pressure conditions, which facilitates its application in power systems [5].

However, the direct use of ammonia in diesel engines still faces several challenges due to its poor combustion characteristics. Ammonia has a high auto-ignition temperature, narrow flammability limits, and low burning velocity, making combustion difficult to initiate and sustain under compression-ignition engine conditions. These characteristics may lead to prolonged ignition delay, a lower heat release rate, reduced peak firing pressure, and delayed combustion during the expansion stroke. Consequently, engine thermal efficiency and power output may deteriorate when the ammonia fraction becomes excessively high. In addition, because ammonia contains nitrogen, its combustion may produce specific emissions such as unburned ammonia, NO_x and N₂O, among which N₂O is a potent greenhouse gas that can offset part of the CO₂ reduction benefit [5, 6].

To overcome these limitations, ammonia-diesel dual-fuel operation is considered a feasible solution for

compression-ignition engines. In this mode, ammonia is typically introduced into the intake port to mix with air, while diesel is directly injected into the cylinder as a pilot fuel to initiate the combustion of the ammonia-air mixture. This approach takes advantage of the good auto-ignition capability of diesel and the carbon-free nature of ammonia, thereby maintaining reliable ignition in diesel engines while reducing hydrocarbon fuel consumption [7]. Previous fundamental studies have shown that ammonia is difficult to auto-ignite in compression-ignition engines without assistance, but it can burn more effectively when a pilot fuel with an appropriate cetane number is used [8]. Many experimental studies have demonstrated that diesel engines can operate with ammonia-diesel mixtures at different substitution ratios. In addition, Hung et al. [9] investigated the effects of laminar burning velocity and ignition timing on the performance and NO_x emissions of a pure-ammonia-fueled motorcycle engine, providing insights into combustion optimization and emission control.

Reiter and Kong showed that, under constant-power conditions, an energy split of approximately 40-60% diesel combined with 60-40% ammonia could provide favorable fuel efficiency. Their results also indicated that partially replacing diesel with NH₃ reduces CO₂ and soot emissions because NH₃ contains no carbon and decreases the amount of diesel involved in soot formation. However, due to lower combustion temperatures and less complete fuel oxidation, CO, HC, and unburned NH₃ emissions tend to increase. In addition, as the NH₃ fraction increases, ignition delay becomes longer and the peak in-cylinder pressure decreases, indicating that the ammonia energy share directly affects the combustion characteristics and work output of the engine [7].

Recent studies have further confirmed that the NH₃ energy share is a key parameter that simultaneously governs the efficiency, combustion process, and emissions of dual-fuel engines. Yousefi et al. reported that increasing the NH₃ energy share tends to reduce thermal efficiency due to the low burning velocity of ammonia. At the same diesel injection timing, increasing the NH₃ energy share from 0% to 40% can reduce NO_x emissions, but it increases N₂O and unburned NH₃ emissions, making the greenhouse gas reduction benefit non-linear with diesel substitution. Their study also showed that advancing the diesel injection timing can help reduce N₂O and overall greenhouse gas emissions while maintaining efficiency close to conventional diesel operation [10]. This indicates that the effective application of NH₃ in diesel engines requires simultaneous consideration of the NH₃ energy share and the combustion control parameters.

In addition to performance and emission indicators, optical studies have also provided direct evidence of the influence of NH₃ energy share on the combustion process inside the combustion chamber. Mao et al. used a full optical engine combined with a high-speed camera to investigate NH₃-diesel and NH₃-PODE₃ operation modes. The results showed that, as the NH₃ energy share increased, the start of combustion tended to be delayed and the initial

flame propagation rate decreased. This indicates that the ammonia fraction not only affects emissions, but also directly influences flame formation, flame development, and the energy release rate in the cylinder [11]. Therefore, the analysis of combustion characteristics such as in-cylinder pressure, heat release rate, combustion timing, and combustion temperature is essential when evaluating ammonia-diesel dual-fuel engines.

More recently, Nguyen et al. [12] investigated the combustion, performance, and emission characteristics of a stationary single-cylinder diesel engine fueled with ammonia-diesel blends at ammonia energy ratios ranging from 0% to 60% using AVL BOOST. The results showed that low ammonia fractions slightly improved engine power and reduced NO_x and CO emissions, but increased brake specific fuel consumption due to the lower heating value of ammonia.

Based on the above studies, ammonia shows strong potential as a partial substitute for diesel because of its carbon-free nature, which can reduce the carbon source involved in CO₂ and soot formation. However, ammonia has poor auto-ignition capability, low burning velocity, and requires more favorable combustion conditions than diesel. Therefore, increasing the ammonia fraction may reduce power, torque, and fuel conversion efficiency. In addition, since ammonia contains nitrogen in its molecular structure, emissions such as NO_x and unburned ammonia should also be carefully considered. For this reason, ammonia energy sharing is a key parameter governing both performance and emissions in ammonia-diesel dual-fuel engines.

Although previous experimental and numerical studies have established the general influence of ammonia substitution on combustion, engine performance, and emissions, the combined effects of ammonia energy share and engine operating speed have not been systematically evaluated, particularly for turbocharged multi-cylinder diesel engines under a consistent engine calibration. The previous AVL BOOST study cited above considered a stationary single-cylinder diesel engine with ammonia energy ratios up to 10% [12]. In contrast, the present study employs a validated turbocharged four-cylinder direct-injection engine model to investigate ammonia energy shares from 0% to 20% over an engine-speed range of 1400–3000 rpm while maintaining identical combustion-model parameters, pilot-injection calibration, and intake-supply conditions. This approach enables the influence of ammonia energy share to be isolated and clarifies how its effects on combustion, engine performance, and emissions evolve with engine speed. The results provide a systematic assessment of the speed-dependent performance–emission trade-off associated with ammonia substitution and offer guidance for selecting appropriate ammonia energy shares under different operating conditions.

2. Materials and Methods

2.1. Research object and fuels

The research object is the DESSUN 4DSP-75 diesel engine, a four-cylinder, four-stroke, turbocharged, direct-

injection engine. This engine was selected because its technical specifications are available for model development and it is suitable for investigating ammonia-diesel dual-fuel operation. The main specifications of the engine are presented in Table 1.

Table 1. Engine model specification [13]

No.	Specification	Value
1	Engine type	In-line, 4-stroke
2	Injection type	Direct injection
3	Firing order	1-3-4-2
4	Injection timing (°)	$15 \pm 1^\circ$ CA BTDC
5	Cylinder type	Dry liner
6	Air intake type	Turbocharger
7	Bore x stroke (mm)	108×118
8	Total displacement	4.325 L
9	Compression ratio	17.5:1
10	Power output	75 kW at 3000 rpm
11	Torque output	296 N.m at 2000 rpm
12	Intake valve opening	24.5° BTDC
13	Intake valve closing	55.5° ABDC
14	Exhaust valve opening	54° BBDC
15	Exhaust valve closing	26° ATDC
16	Brake specific fuel consumption (BSFC)	225 g/kWh at 2000 rpm

In contrast, ammonia offers an important environmental advantage because it contains no carbon atoms. Therefore, replacing part of diesel with ammonia can reduce the carbon source available for soot and carbon-based emissions. However, ammonia also has several unfavorable combustion properties. Its lower heating value (LHV) 18.61 MJ/kg is much smaller than that of diesel fuel (42.38 MJ/kg), meaning that a larger fuel mass is required to provide the same energy input Table 2.

Table 2. Properties of ammonia [16a] and diesel fuels [14]

No.	Parameter	NH ₃	Diesel	Unit
1	Octane number	120	-	-
2	Cetane number	-	40-55	-
3	Liquid density	602.8	832	kg/m ³
4	Stoichiometric air-fuel ratio	6.05	14.32	kg/kg
5	Lower heating value (LHV)	18.61	42.38	MJ/kg
6	Latent heat of vaporization	1370	232.4	kJ/kg
7	Auto-ignition temperature	924	527-558	K

In addition, ammonia has a very high auto-ignition temperature (924 K) and low combustion reactivity, making it difficult to ignite directly in compression-ignition engines. Its high latent heat of vaporization can also reduce the local mixture temperature, which may further delay ignition and slow flame development. For this reason, diesel was used as the pilot fuel in this study, while ammonia was supplied through the intake pipe to form the dual-fuel mixture.

2.2. Theoretical background and calculation methods

To validate the one-dimensional simulation model, the main brake performance parameters of the engine, including brake power, brake mean effective pressure (BMEP) and brake specific fuel consumption (BSFC) were

calculated using fundamental engine relations. These parameters were then compared with the reference data of the DESSUN 4DSP-75 diesel engine.

The brake power was calculated from the engine speed and brake torque as follows:

$$N_e = \frac{2\pi n M_e}{60 \times 1000} \quad (1)$$

Where N_e is the brake power (kW), n is the engine speed (rpm), and M_e is the brake torque (N.m).

The BMEP was determined as:

$$BMEP = \frac{4\pi M_e}{V_d} \quad (2)$$

Where V_d is the total displacement volume of the engine. BMEP was used as an indicator of the engine load and work-producing capability.

The BSFC was calculated by:

$$BSFC = \frac{\dot{m}_f}{N_e} \quad (3)$$

Where $\dot{m}_f$ is the fuel mass flow rate.

In the ammonia-diesel dual-fuel mode, ammonia was used as a partial energy substitute for diesel fuel. Since ammonia and diesel have different LHV, the substitution level should be evaluated on an energy basis rather than only on a mass basis. Therefore, the ratio of ammonia energy (RAE) was adopted to quantify the contribution of ammonia to the total fuel energy supplied to the engine, following previous studies [15, 16]

$$RAE = \frac{\dot{m}_{NH_3} \times LHV_{NH_3}}{\dot{m}_{diesel} \times LHV_{diesel} + \dot{m}_{NH_3} \times LHV_{NH_3}} \quad (4)$$

Where, $\dot{m}_{NH_3}$ denotes the ammonia mass introduced into the intake pipe and $\dot{m}_{diesel}$ denotes the diesel mass injected into the cylinder for each engine cycle. LHV_{NH_3} and LHV_{diesel} are the lower heating values of ammonia and diesel, respectively. The corresponding fuel mass and energy inputs for each simulation case are presented in Table 2.

Because ammonia and diesel have substantially different lower heating values, brake thermal efficiency (BTE) was additionally evaluated to provide an energy-based measure of the dual-fuel engine performance:

$$\eta_b = \frac{N_e}{\dot{m}_{diesel} \times LHV_{diesel} + \dot{m}_{NH_3} \times LHV_{NH_3}} \times 100 \quad (5)$$

Where, η_b is the brake thermal efficiency (%), N_e is the brake power, and $\dot{m}_{diesel}$ and $\dot{m}_{NH_3}$ are the diesel and ammonia mass flow rates, respectively. All quantities in Eq. (5) were expressed in consistent energy-rate units.

2.3. AVL BOOST simulation model setup

The DESSUN 4DSP-75 diesel engine model was developed in AVL BOOST based on the engine specifications listed in Table 1. The model was established to simulate the main operating processes of the engine, including intake, turbocharging, fuel injection, combustion, heat transfer, and exhaust flow.

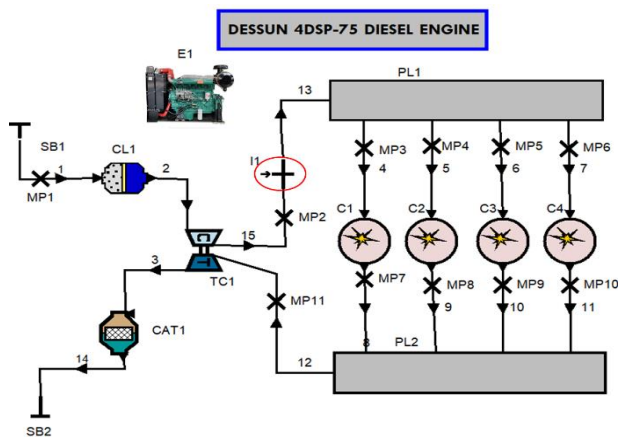


Figure 1. AVL BOOST model of the DESSUN 4DSP-75 engine

As shown in Figure 1, the model consists of the inlet and outlet boundary conditions, air cleaner, turbocharger, intake and exhaust pipes, intake plenum, exhaust plenum, four cylinders, diesel injection elements, catalytic converter, ammonia injector, and measuring points.

In the intake system, fresh air enters from the inlet boundary condition **SB1** and then passes through the air cleaner **CL1**, where the intake pressure loss is represented. The airflow is subsequently compressed by the compressor side of the turbocharger **TC1** before entering the intake plenum **PL1**. From the intake plenum, the compressed air is distributed to the four cylinders **C1**, **C2**, **C3** and **C4**. After combustion occurs in the cylinders, the exhaust gases are collected in the exhaust plenum **PL2** and then pass through the turbine side of the turbocharger **TC1**, where part of the exhaust gas energy is recovered to drive the compressor. The exhaust gases subsequently pass through the catalytic converter **CAT1** and leave the system through the outlet boundary condition **SB2**. This arrangement allows the model to represent the main gas-exchange process of the turbocharged diesel engine.

Diesel fuel was directly injected into the combustion chamber. The combustion process was simulated using the AVL MCC model. This model is suitable for direct-injection diesel engines because it considers both premixed combustion and mixing-controlled combustion. Therefore, it can describe the relationship between fuel injection, fuel-air mixing, heat release, and combustion development inside the cylinder, while also supporting the prediction of engine performance and emission trends [17, 18]. The in-cylinder heat transfer was calculated using the Woschni 1990 model. The piston, cylinder-head, and liner heat-transfer calibration factors were each set to 1.0 and retained unchanged for all operating conditions to provide a consistent basis for comparison. Mechanical losses were represented using the FMEP characteristic as a function of engine speed.

After the baseline diesel model was validated, it was converted to ammonia–diesel dual-fuel operation by adding a continuous NH_3 injector **I1** between Pipes 13 and 15, downstream of the compressor and upstream of the intake plenum **PL1**. NH_3 was introduced as a local species at a supply temperature of 25°C. Because the injector operated in continuous mode, no crank-angle-based NH_3

injection timing or injection duration was specified. The intake pressure at the injector location was determined by the one-dimensional gas-exchange calculation, and the heat-of-evaporation option was disabled; consequently, latent-heat effects associated with NH_3 phase change were not included in the model. The injector location upstream of **PL1** was selected to provide a sufficient mixing length before the intake charge was distributed to cylinders **C1**–**C4**. Within the one-dimensional formulation of AVL BOOST framework, the intake mixture was represented by cross-sectionally averaged flow properties prior to cylinder distribution. Diesel was retained as the directly injected pilot fuel to initiate combustion, while NH_3 was supplied through the intake system as the secondary fuel.

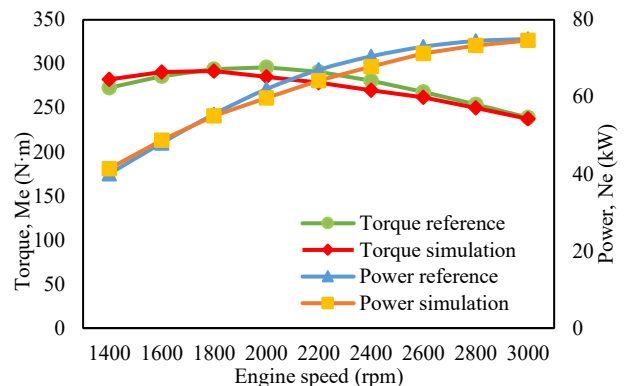


Figure 2. Comparison between simulated results and manufacturer reference data for torque and power characteristics of the engine

After developing the DESSUN 4DSP-75 engine model in AVL BOOST, the baseline diesel model was validated by comparing the simulated torque, power, and BSFC with the manufacturer reference data over the engine speed range of 1400–3000 rpm. The comparison of torque, power, and the relative errors are summarized in Figure 2. The results show that the simulation model reasonably reproduces the full-load performance trends of the baseline diesel engine. The torque increases at low engine speeds, reaches its maximum value in the medium-speed range, and then decreases as the engine speed increases. Meanwhile, the brake power increases with engine speed. The maximum errors of torque, power, and BSFC are 4.24%, 4.18%, and 4.41%, respectively. These deviations are within an acceptable range for a one-dimensional engine simulation model under baseline diesel operation.

For the validated baseline diesel model, the MCC combustion-model parameters were established during the baseline model development and retained for all subsequent simulations. The adopted MCC settings included an ignition-delay calibration factor of 1.7, combustion, turbulence, and dissipation parameters of 1.0, a premixed combustion parameter of 0.7, a premixed combustion duration factor of 2.0, the revised TKE calculation option, a combustion excess-air-ratio development parameter of 0, and an evaporation velocity parameter of 0.70353. Since the AVL MCC combustion model was employed, Wiebe-function parameters were not applicable. Likewise, no separate user-defined ignition-delay correlation or prescribed combustion-duration function was employed.

The above validation was performed for the baseline diesel engine by comparing the simulated brake performance with the manufacturer reference data. Experimental data for the same engine operating in ammonia–diesel dual-fuel mode, such as in-cylinder pressure, heat-release rate, ignition delay, and engine-out emissions, were not available for additional validation. Consequently, the validated baseline model was extended to investigate the influence of ammonia energy share while maintaining a common engine configuration, combustion-model parameters, and simulation assumptions for all investigated cases. Under these controlled conditions, the simulation results provide a consistent comparative assessment of the effects of ammonia energy share on engine performance, combustion characteristics, and emissions.

Within the adopted AVL BOOST MCC-based one-dimensional engine model, the emission analysis was conducted for NO_x, CO, and soot, which represent the pollutant species supported by the implemented combustion and emission sub-models. The simulation framework was therefore employed to provide a consistent comparative evaluation of the effects of ammonia energy share on engine performance and the predicted emissions under identical engine operating conditions.

2.4. Simulation cases for ammonia–diesel dual-fuel operation

After the ammonia–diesel dual-fuel model was established, simulation cases were defined based on the ratio of ammonia energy (RAE). In this study, five RAE values of diesel baseline, 5%, 10%, 15%, and 20% were selected to evaluate the influence of ammonia energy contribution on engine performance, combustion behavior, and emissions. The 0% RAE case represents the baseline diesel operation, while the remaining cases represent increasing ammonia energy substitution levels.

Simulations were performed over an engine-speed range of 1400–3000 rpm. At each engine speed, the prescribed RAE was achieved by adjusting the diesel pilot-fuel mass flow rate, $\dot{m}_{\text{diesel}}$, and the ammonia mass flow rate supplied through the intake pipe, $\dot{m}_{\text{NH}_3}$, according to Eq. (4). To ensure a controlled comparison among the investigated RAE cases, the diesel pilot-injection timing was fixed at the baseline diesel setting of approximately 15°CA BTDC, and the normalized rate-of-injection profile was retained unchanged throughout all simulations. The diesel pilot-fuel mass was adjusted together with the NH₃ mass according to Eq. (4) solely to achieve the prescribed RAE and was not treated as an independent optimization variable. The NH₃ supply configuration, including the injector location, supply temperature, continuous-supply mode, and intake-system configuration, was retained unchanged for all simulation cases to ensure a consistent basis for comparison. The validated MCC combustion-model parameters and Woschni 1990 heat-transfer settings were also retained unchanged for all simulation cases. Consequently, the observed differences among the investigated cases primarily reflect the influence of ammonia energy share under a common pilot-injection calibration, unchanged supply configuration, and consistent simulation conditions. The corresponding NH₃ quantities for the investigated RAE cases are presented in Table 3.

Table 3. Ammonia mass supplied through the intake pipe for ammonia–diesel dual-fuel simulation cases

Engine speed (rpm)	Diesel baseline mg	RAE (mg/cycle/cylinder)			
		5%	10%	15%	20%
1400	56.43	25.79	51.57	77.36	103.15
1600	58.00	26.51	53.01	79.52	106.02
1800	58.58	26.77	53.54	80.32	107.09
2000	58.13	26.56	53.13	79.69	106.25
2200	57.10	26.10	52.19	78.29	104.38
2400	55.57	25.39	50.79	76.18	101.58
2600	54.05	24.70	49.40	74.10	98.80
2800	52.11	23.81	47.62	71.44	95.25
3000	50.00	22.85	45.70	68.55	91.40

3. Results and Discussion

3.1. Effect of ammonia energy share on engine performance

Figure 3 and Figure 4 show that both engine power and torque decrease as the ammonia energy share increases from 0% to 20%. The baseline diesel case consistently gives the highest values, while the ammonia 20% case shows the lowest values over the entire investigated speed range. This indicates that replacing part of diesel with ammonia reduces the work-producing capability of the engine.

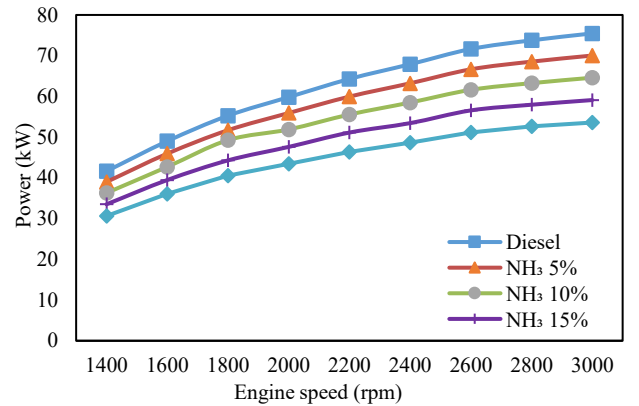


Figure 3. Effect of ammonia energy share on engine power

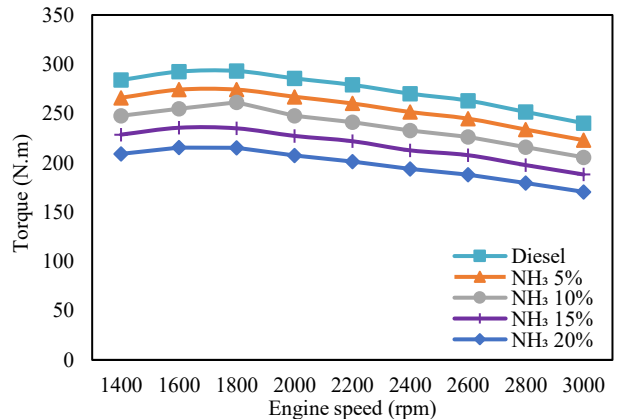


Figure 4. Effect of ammonia energy share on engine torque

For brake power, all cases show an increasing trend as engine speed rises from 1400 to 3000 rpm. This behavior is attributed to a higher engine speed increase in the number of working cycles per unit time, leading to higher

power output. However, the power reduction becomes more noticeable at higher ammonia shares, especially in the high-speed range from about 2400 to 3000 rpm. In this region, the time available for combustion becomes shorter, while ammonia has a low burning velocity and poor auto-ignition capability. As a result, combustion may be prolonged into the expansion stroke, reducing the fraction of fuel energy converted into useful work.

For torque, the value increases slightly at low engine speeds, reaches its maximum at around 1600-1800 rpm, and then gradually decreases as the engine speed continues to increase. This is a typical characteristic of diesel engines. In the medium speed range, the intake, compression, and combustion processes occur more favorably, resulting in higher torque. At higher speeds, increased mechanical losses, pumping losses, and shorter combustion time lead to lower torque. The torque reduction becomes more evident as the ammonia energy share increases because the diesel pilot quantity decreases, weakening the ignition source and reducing the combustion intensity.

The main reason for the reduction in power and torque is related to the fuel properties of ammonia. As shown in Table 2, the LHV of ammonia is only 18.61 MJ/kg, much lower than the LHV of diesel, 42.38 MJ/kg. In addition, ammonia has a very high auto-ignition temperature of 924 K, compared with only 527-558 K for diesel. These properties indicate that ammonia is harder to ignite, burns more slowly, and releases less energy per unit mass than diesel. When the ammonia share increases, the diesel pilot quantity decreases, making ignition more difficult, slowing down heat release and reducing the pressure acting on the piston. Consequently, torque decreases since brake power depends directly on torque and engine speed, brake power also decreases.

Overall, the differences among ammonia energy shares are more pronounced in the medium to high-speed range, particularly from about 2000 to 3000 rpm. These results suggest that ammonia can be used to partially replace diesel, but its energy share should be limited and combined with optimization of diesel pilot quantity, injection timing, and combustion strategy to avoid significant deterioration in engine dynamic performance.

Figure 5(a) and 5(b) present the variations in BSFC and BTE, respectively. At all investigated engine speeds, the baseline diesel case exhibits the lowest BSFC and the highest BTE, whereas the NH_3 20% case exhibits the highest BSFC and the lowest BTE. As the ammonia energy share increases, a greater total fuel mass is required to produce the same brake power because NH_3 has a lower LHV than diesel. Consequently, part of the increase in mass-based BSFC is associated with the different energy contents of the two fuels.

The BTE results further demonstrate that the observed performance deterioration cannot be attributed solely to the lower LHV of NH_3 . BTE also decreases progressively with increasing ammonia energy share, indicating that the engine converts a smaller fraction of the supplied chemical energy into useful brake work. This reduction is primarily

associated with the lower reactivity and burning velocity of NH_3 , together with the reduced diesel pilot-fuel quantity, which delay ignition and slow the heat-release process under the adopted common pilot-injection calibration. The inclusion of BTE confirms that the conclusions drawn from the BSFC analysis remain valid when engine performance is evaluated on an energy basis rather than solely on a mass-fuel basis.

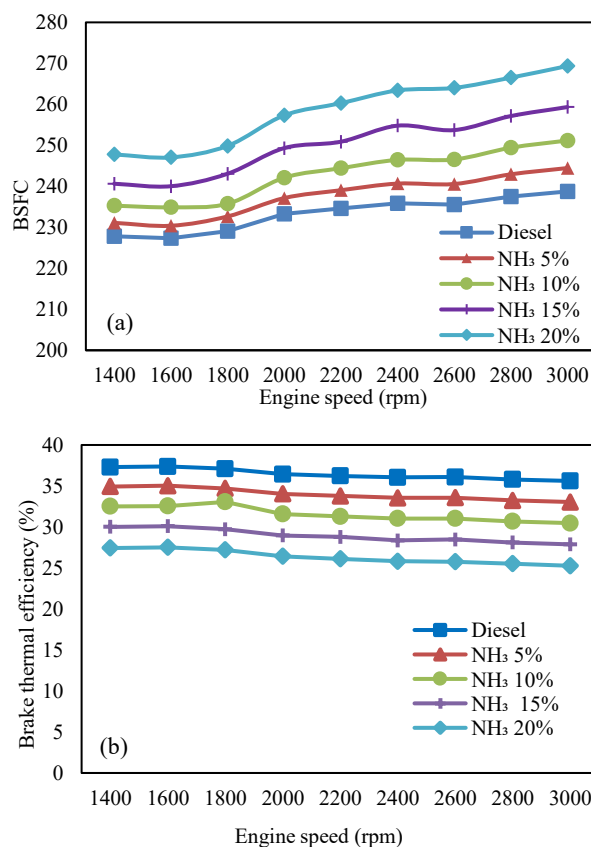


Figure 5. Effect of ammonia energy share on (a) brake specific fuel consumption and (b) brake thermal efficiency

The differences among the investigated cases become more pronounced in the medium-to-high-speed range (approximately 2000 to 3000 rpm), where the shorter available combustion duration further limits the effective utilization of NH_3 . Consequently, the increase in BSFC and the decrease in BTE become progressively larger as the ammonia energy share increases.

The more pronounced reduction in engine power at higher engine speeds is associated with the common pilot-injection calibration adopted throughout this study. As engine speed increases, the shorter available combustion duration amplifies the influence of the lower reactivity and burning velocity of NH_3 , resulting in delayed combustion phasing and less effective energy conversion. Although optimization of the diesel pilot-injection timing or pilot-fuel quantity could improve ignition initiation and combustion phasing, such adjustments would also alter the combustion process and associated emission characteristics. Therefore, a common pilot-injection calibration was maintained for all simulation cases to provide a consistent basis for evaluating the influence of ammonia energy share. Future studies should investigate

the combined optimization of ammonia energy share and diesel pilot-injection strategy to further improve the performance–emission trade-off.

3.2. Effect of ammonia energy share on combustion characteristics

Figures 6 and 7 illustrate the variations in peak cylinder pressure and peak firing temperature with engine speed under different ammonia energy shares. Overall, both peak pressure and peak temperature decrease as the ammonia energy share increases from 0% to 20%. The diesel case always gives the highest values, while the ammonia 20% case gives the lowest values over the entire investigated speed range. This indicates that increasing ammonia weakens the combustion intensity inside the cylinder.

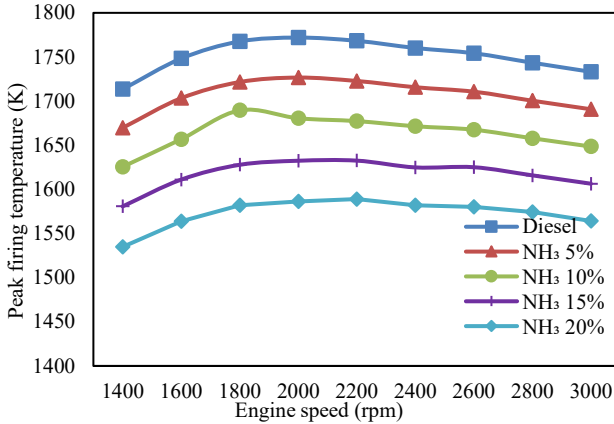


Figure 6. Effect of ammonia energy share on peak firing temperature

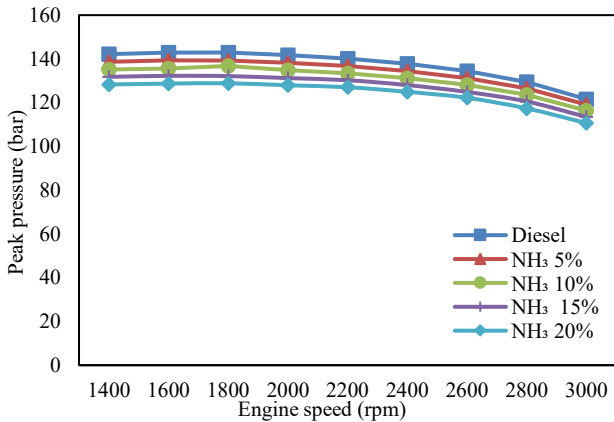


Figure 7. Effect of ammonia energy share on peak cylinder pressure

With respect to engine speed, peak cylinder pressure remains relatively high in the low and medium-speed ranges, then decreases more clearly at higher speeds, especially from about 2600 to 3000 rpm. Peak firing temperature increases up to around 1800–2200 rpm and then slightly decreases at higher speeds. This trend is reasonable because the intake, compression, mixing and combustion processes are generally more favorable in the medium-speed range. At higher speeds, the time available for combustion becomes shorter, making heat release less complete and more likely to shift into the expansion stroke.

The main reason is related to the fuel properties of ammonia. Compared with diesel, ammonia has a lower

LHV, higher auto-ignition temperature, and lower burning velocity. As the ammonia share increases, the diesel pilot quantity decreases, weakening the ignition source and slowing the flame development. As a result, the heat release rate decreases, leading to lower peak cylinder pressure and peak firing temperature.

These results explain the reduction in power and torque discussed earlier. Lower peak pressure reduces the force acting on the piston, while lower peak temperature indicates weaker heat release intensity. Therefore, less useful work is produced during the expansion stroke as the ammonia energy share increases.

3.3. Effect of ammonia energy share on emissions

Figure 8–10 show the effects of ammonia energy share on NO_x, CO, and soot emissions. Overall, NO_x and CO emissions increase when the ammonia energy share increases, while soot emissions decrease significantly. This indicates a clear trade-off in ammonia–diesel dual-fuel operation: ammonia helps reduce carbon-based particulate emissions, but it may increase gaseous emissions such as NO_x and CO.

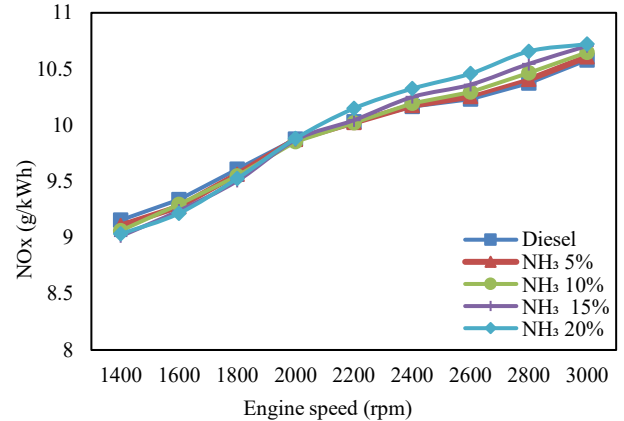


Figure 8. Effect of ammonia energy share on engine NO_x emissions

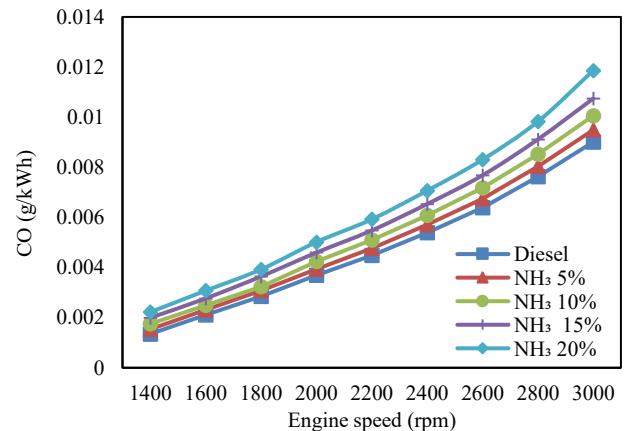


Figure 9. Effect of ammonia energy share on CO emissions

For NO_x emissions, the values increase with engine speed for all cases. The difference between diesel and ammonia cases is relatively small at low speeds, but it becomes more noticeable at medium and high speeds. The ammonia 20% case gives the highest NO_x emissions, especially from about 2200 to 3000 rpm. Although

ammonia addition reduces peak combustion temperature, ammonia contains nitrogen in its molecular structure. Therefore, fuel-bound nitrogen may contribute to NO_x formation during combustion.

For CO emissions, the increasing trend with ammonia is more obvious. CO rises continuously as engine speed increases and the ammonia 20% case gives the highest CO emissions over the whole speed range. This is mainly due to the poor auto-ignition capability and low burning velocity of ammonia. As the ammonia share increases, combustion becomes slower and less complete. At higher speeds, the available combustion time is shorter, so the oxidation of CO to CO_2 becomes less effective.

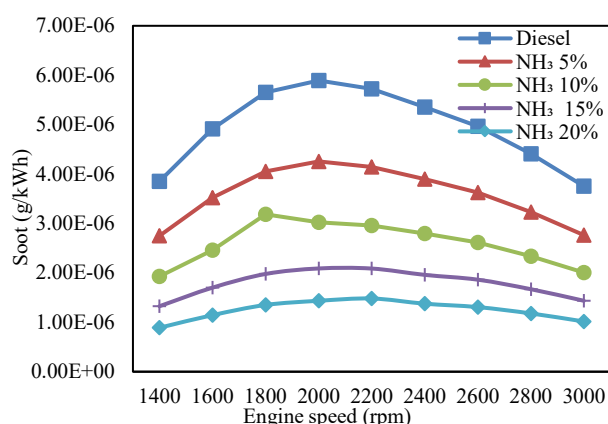


Figure 10. Effect of ammonia energy share on soot emissions

In contrast, soot emissions decrease clearly as the ammonia energy share increases. The diesel case produces the highest soot, while the ammonia 20% case gives the lowest values. This is because ammonia contains no carbon atoms; therefore, replacing part of diesel with ammonia reduces the carbon source available for soot formation. The soot curve also increases from low speed to around 2000–2200 rpm and then decreases at higher speeds, which may be related to changes in combustion temperature, fuel-air mixing, and soot oxidation.

The emission results show that ammonia has a strong advantage in reducing soot emissions, but high ammonia shares can increase NO_x and CO. Therefore, the ammonia energy share should be optimized together with diesel pilot quantity, injection timing, air-fuel mixing and possibly EGR or exhaust after-treatment systems to achieve a better balance between soot reduction and gaseous emission control.

The simulation results show that ammonia energy share is a key parameter affecting the performance, combustion process and emissions of the ammonia-diesel dual-fuel engine. As the ammonia energy share increases from 0% to 20%, brake power and torque decrease over the entire investigated speed range while BSFC increases. This trend is mainly caused by the unfavorable combustion properties of ammonia, including its LHV, higher auto-ignition temperature, and lower burning velocity compared with diesel. As the diesel pilot quantity decreases, the ignition source becomes weaker, heat release slows down, peak cylinder pressure decreases and the force acting on the

piston is reduced. Therefore, the conversion of fuel energy into useful work becomes less effective.

The combustion results further support this explanation. Both peak firing temperature and peak cylinder pressure decrease with increasing ammonia energy share, indicating a lower combustion intensity inside the cylinder. This directly explains the reductions in brake power and torque. The difference among ammonia cases becomes more evident in the medium and high-speed ranges, where the available combustion time is shorter while ammonia requires more favorable conditions to burn completely. Regarding emissions, increasing ammonia energy share creates a clear trade-off. Soot emissions decrease significantly because ammonia contains no carbon atoms, reducing the carbon source available for soot formation. This is the main advantage of ammonia in reducing particulate-related emissions and diesel consumption. However, CO and NO_x emissions tend to increase. CO increases due to slower combustion and incomplete oxidation, while NO_x may increase because the nitrogen contained in ammonia can contribute to fuel- NO_x formation.

Overall, the present study demonstrates that the influence of ammonia energy share is not uniform across the investigated operating range. As engine speed increases, the shorter available combustion time amplifies the adverse effects of ammonia's lower reactivity, leading to greater reductions in power, torque, peak cylinder pressure, and peak firing temperature, together with larger increases in BSFC, NO_x , and CO emissions, whereas the soot-reduction benefit is maintained because ammonia contains no carbon. These findings indicate that RAE should be selected in conjunction with engine operating speed to achieve an appropriate balance between engine performance and emissions. Future work will focus on optimizing ammonia energy share together with diesel pilot-injection strategy and combustion-control parameters to further improve dual-fuel engine performance. In addition, intake system modification or stronger in-cylinder swirl could improve ammonia-air mixing, while EGR or exhaust after-treatment systems may be considered for more effective NO_x control.

4. Conclusions

This study evaluates the effect of ammonia energy share on the performance and emissions of an ammonia-diesel dual-fuel engine using an AVL BOOST simulation model. When the ammonia energy share increases from 0% to 20%, engine power and torque decrease, while brake specific fuel consumption increases. This is mainly because ammonia has a lower LHV, a higher auto-ignition temperature and a lower burning velocity than diesel. As the diesel pilot quantity decreases, ignition and heat release become weaker, reducing the useful work produced inside the cylinder. Regarding emissions, soot decreases clearly because ammonia contains no carbon, thereby reducing the carbon source involved in soot formation. However, CO and NO_x tend to increase due to slower combustion, incomplete oxidation, and the possible

formation of NO_x from nitrogen contained in ammonia. The results indicate that ammonia has the potential to reduce soot emissions and diesel consumption in dual-fuel engines. However, a high ammonia energy share may reduce engine performance and increase CO and NO_x emissions. The present study provides useful insight into the feasibility of ammonia utilization in compression-ignition engines and offers a basis for future optimization of ammonia–diesel dual-fuel combustion strategies. The present findings provide a comparative assessment of the influence of ammonia energy share on engine performance and emissions under the investigated engine configuration and operating conditions. While the baseline diesel model was validated against manufacturer reference data, future experimental investigations under ammonia–diesel dual-fuel operation will provide additional validation of the predicted combustion and emission characteristics. Building upon the validated simulation framework developed in this study, future work will incorporate detailed ammonia oxidation chemistry to enable the prediction of NH₃ slip and N₂O emissions, thereby providing a more comprehensive assessment of ammonia-fueled engine emissions.

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