AMMONIA AS A ZERO-CARBON ALTERNATIVE FUEL FOR HEAVY-DUTY COMPRESSION-IGNITION (CI) ENGINES

For heavy-duty compression-ignition (diesel) engines, ammonia is becoming an attractive carbon-free fuel option. However, the direct use of ammonia is limited by its high ignition energy and low chemical reactivity, requiring a dual-fuel operation with diesel acting as the ignition source. The effects of ammonia energy percentage, fuel injection techniques, and operating circumstances on engines performance and emissions are the main topics of this article's analysis of current developments in ammonia-diesel combustion. The findings show that, within a certain stable operating range, raising the ammonia substitution ratio can enhance indicated thermal efficiency and lower CO_2 and particulate matter emissions by encouraging premixed combustion. Beyond this range, combustion decomposition results in higher NO and N_2O emissions, ammonia slip, and ignition delay, emphasizing a crucial trade-off between pollutants generation and carbon reduction objectives. Although emissions still exist, optimization techniques include improving injection timing, raising injection pressure, and managing combustion modes allowing the use of larger ammonia ratios without sacrificing performance. Overall, ammonia-diesel dual-fuel systems have significant potential for heavy-duty applications that want to reduce carbon emissions, as long as the combustion chambers and exhaust aftertreatment systems are well-designed.

Introduction

Ammonia is gaining popularity as a carbon-free fuel for heavy-duty compression-ignition (diesel) engines due to its low carbon content and ability to be stored and transported as a liquid at lower pressures. In theory, this makes ammonia a desirable option for decarbonizing industries that are challenging to electrify, especially high-power off-road machinery, heavy-duty transportation, and marine engines. However, ammonia is not a direct substitute for diesel since its low flame speed, high required ignition energy, and limited chemical reactivity provide significant combustion problems when used alone in compression-ignition engines. These characteristics lead to prolonged ignition delay, reduced combustion stability, and incomplete combustion, particularly under operating conditions involving lean mixtures or poor mixing.[1,2,3,4]

Because of this, the majority of current research focuses on dual-fuel ammonia-diesel operation, in which ammonia supplies much of the energy input while diesel serves as the ignition source. Although this approach can significantly reduce emissions of carbon dioxide and carbon monoxide and reduce soot, it also introduces new difficulties, such as the possibility of increased nitrogen-containing pollutants like NO and N_2O, the difficulty of maintaining combustion stability at high ammonia substitution rates, and the potential for significant emissions of unburned ammonia (ammonia slip). Therefore, the essential question is not whether ammonia could be helpful in lowering carbon emissions, but rather how well the combustion system can be adjusted to maintain performance while minimizing emissions that are harmful to the environment. This review examines combustion behavior, engine performance, emission characteristics, and optimization strategies discussed in recent studies, with particular focus on the effects of the ammonia substitution ratio, fuel injection strategies, combustion phasing, and operating conditions on overall engine performance.[3,5]

Discussion

Recent studies by Nadimi et al. and Zhang et al. provide the basis for evaluating ammonia-diesel combustion and optimization in heavy-duty compression-ignition engines. Nadimi et al. investigated the effects of increasing ammonia energy share (AES) on combustion, engine performance, and emissions under 1200 rpm and full-load operation. Zhang et al. focused on optimizing injection parameters to achieve higher ammonia substitution while maintaining combustion stability and reducing emissions.

Figures 1 to 4 show the experimental and modeling results provided by Nadimi et al. for a single-cylinder ammonia-diesel dual-fuel CI engine operating at 1200 rpm and full load, while table 1 highlights the optimization results from a separate high-ammonia-substitution investigation by Zhang et al.

Nadimi et al. conducted an experimental and modeling study on an ammonia-diesel dual-fuel compression-ignition engine to understand how substituting diesel with ammonia impacts combustion mode, engine performance, and nitrogen-containing pollutants. Their effort aims to push diesel replacement to the greatest achievable level under full-load, 1200 rpm operation while maintaining stable running, and to evaluate how AES affects ignition delay, combustion phasing, combustion length, unburned ammonia, and N_2O emissions. A 1D model was developed and used alongside engine experiments to generate combustion indicators such as SOC, CA50, and heat release behavior for various ammonia-diesel ratios. Nadimi et al.'s research uses the dataset to highlight the operating window, combustion behavior, and emissions trade-offs for high AES in heavy-duty CI engines.

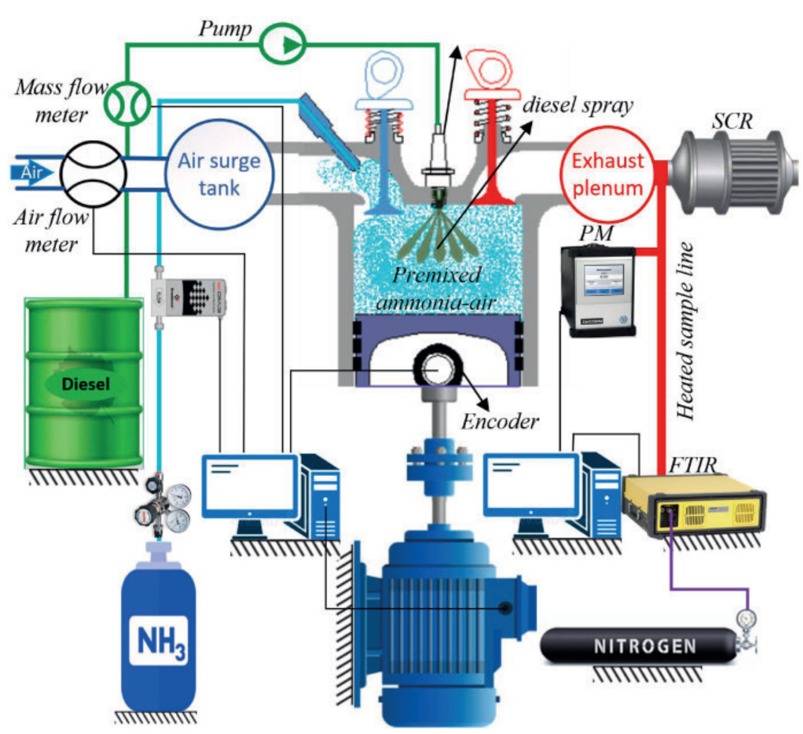
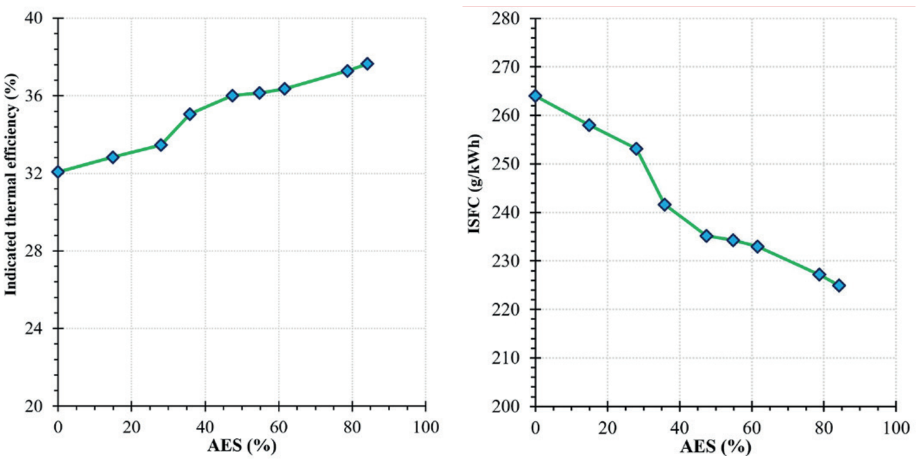


Figure 1: Ammonia-diesel engine test setup[3]

Nadimi et al. used a single-cylinder compression-ignition engine with dual-fuel delivery setup to establish the most stable ammonia substitute for diesel. Diesel was injected directly into the cylinder, while gaseous ammonia was aspirated near the intake valve. The ammonia energy share (AES) was varied by reducing diesel mass flow and increasing ammonia flow while keeping the engine at full load, 1200 rpm, and maintaining the same output power as the pure-diesel reference condition. The experiment's goal was to determine how much ammonia could replace diesel while keeping the engine running at full load at 1200 rpm. AES was used as main control variable for creating the operational map. The engine was able to provide power comparable to pure diesel operation by lowering the diesel mass flow rate at each operational position and adding gaseous ammonia close to the intake valve.

Figure 1 shows that increasing the AES raises the ammonia contribution while reducing the air mass flow rate. This lowers the stoichiometric air-fuel ratio (AFR) of the mixture, as ammonia has much lower stoichiometric AFR (6.0) than diesel. This suggests that even when the engine runs at maximum load, operational conditions become more limited as the AES increases. As a result, this experiment provides the top limit for stable replacement at a given speed and load, going beyond a straightforward fuel comparison. This operating range also explains why full-load operation and 1200 rpm were chosen for the investigation. Low-speed, full-load operation offers the best circumstances for maintaining combustion stability as the replacement ratio rises because of ammonia's low burning velocity and high minimum ignition energy.



(a) Indicated efficiency of the ammonia/diesel-fueled engine for different ammonia energy share.

(b) Equivalent indicated specific fuel consumption of the ammonia/diesel-fueled engine.

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Figure 2. AES vs ITE and $ISFC_{eq}$ for performance trends.[3]

Indicated thermal efficiency (ITE) in Fig. 2(a) and equivalent indicated specific fuel consumption ($ISFC_{eq}$) in Fig.2(b) is two related performances. ITE represents the fraction of the chemical energy supplied by fuels that is converted into indicated work inside the cylinder in Eq.(1):

$$ITE = \frac{P_i}{\dot{m}_f * LHV_f + \dot{m}_{NH_3} * LHV_{NH_3}} \qquad (1)^3$$

Where Pi represents power.

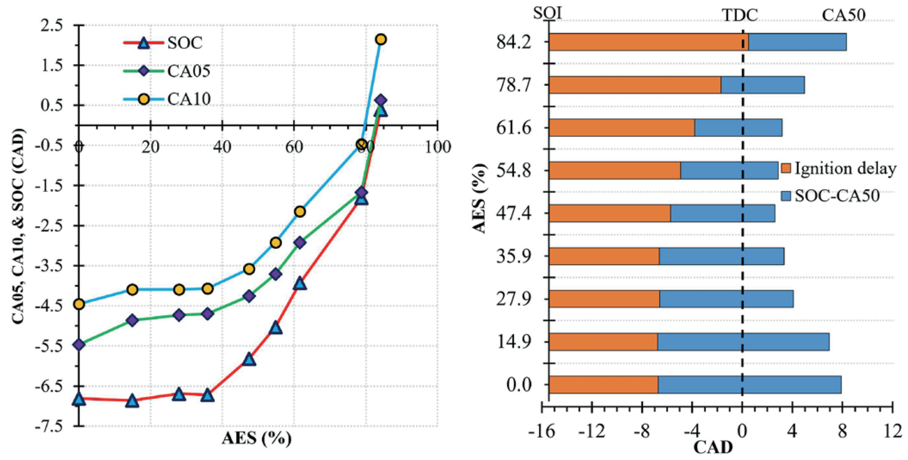
$ISFC_{eq}$ is defined in Eq. (2) in the dual fuel engine.

$$ISFC_{eq} = \frac{\dot{m}_f + \dot{m}_{NH_3} \frac{LHV_{NH_3}}{LHV_f}}{P_i} \qquad (2)^3$$

According to the trends, raising the AES lowers fuel consumption, expressed as $ISFC_{eq}$ in g/kWh, and increases thermal efficiency (ITE), which represents the fraction of fuel's chemical energy converted into indicated engine work, but only within a stable working range. ITE is lowest while running on pure diesel; it rises later when ammonia replaces some of the diesel energy. At the same time, $ISFC_{eq}$ falls, meaning that less total fuel energy is needed to produce the same quality of usable power. A change in combustion mode is reflected in the efficiency gain. The engine switches from a mostly diffusion-controlled diesel combustion mechanism to a dual-fuel operation that leans toward premixed combustion as AES increases. This change reduces energy losses and enhances the heat release phase while $ISFC_{eq}$ decreases from 264 g/kWh to 224.9 g/kWh and ITE increases from 32.0% to 37.6%. As long as the pilot diesel charge is powerful enough to maintain prompt ignition, this simultaneous shift shows that ammonia replacement improves efficiency.

This focused heat release feature is directly related to the thermodynamic mechanism causing the observed rise in ITE. The peak in-cylinder pressure is increased, expansion work is maximized, and relative heat loss to the coolant is decreased throughout the crank angle then most of the fuel mass is burned close to top dead center (TDC). When overall fuel mass consumption is taken into account, there is still a considerable performance disadvantage. $ISFC_{eq}$ increases gradually with the AES because liquid ammonia has a gravimetric energy density that is less than half that of ordinary diesel (about 18.6 MJ/kg versus 42.4 MJ/kg). This suggests that the operating needs of dual-fuel systems demand much larger fuel mass flow rates and bigger fuel storage capacity on vehicle platforms, even while high thermal efficiency may be maintained or enhanced.

This general pattern shows an optimization range where performance keeps getting better until combustion stability starts to decline close to the replacement limit, which is the point when cycle-to-cycle variation and ignition delay become important considerations. As a result, figure 2 supports the idea that ammonia-diesel dual-fuel operating mode can improve efficiency and lower equivalent fuel consumption. However, this advantage is dependent upon steady ignition and suitable combustion phasing.



(a) Start of combustion, CA05, and CA10.

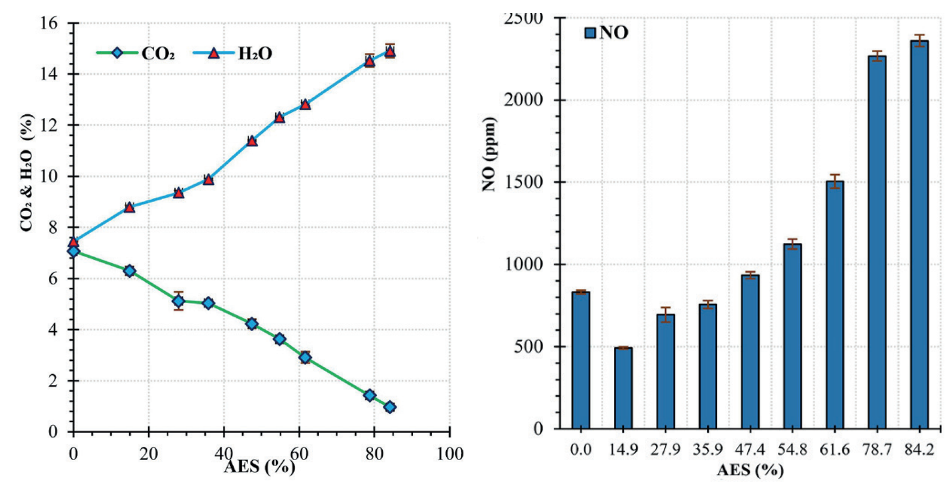
(b) Ignition delay (SOI-SOC), combustion phasing period (SOC-CA50), and location of CA50.

Figure 3. Ammonia/diesel combustion characteristics indicators for different ammonia energy shares.[3]

Figure 3 illustrates the combustion characteristics, reflecting the overall trends in ignition delay, combustion duration, in-cylinder pressure, and the rate of pressure rise. The key observed pattern is that the combustion process slows as the ammonia has a high minimum ignition energy demand and low chemical reactivity, the ignition delay ranges from 8.7 to 15.9 CAD. Conversely, the combustion duration decreases from 56 CAD to 24 CAD, indicating that, once ignition begins, a larger fraction of the fuel-air mixture burns rapidly through premixed combustion. Peak in-cylinder pressure increases from 76.3 bar (diesel-only operation) to 86.2 bar at a moderate AES level of around 61.6%. This suggests that while excessive ammonia replacement delays ignition and moves the combustion phase away from the ideal timing window, moderate substitution improves combustion intensity and timing.

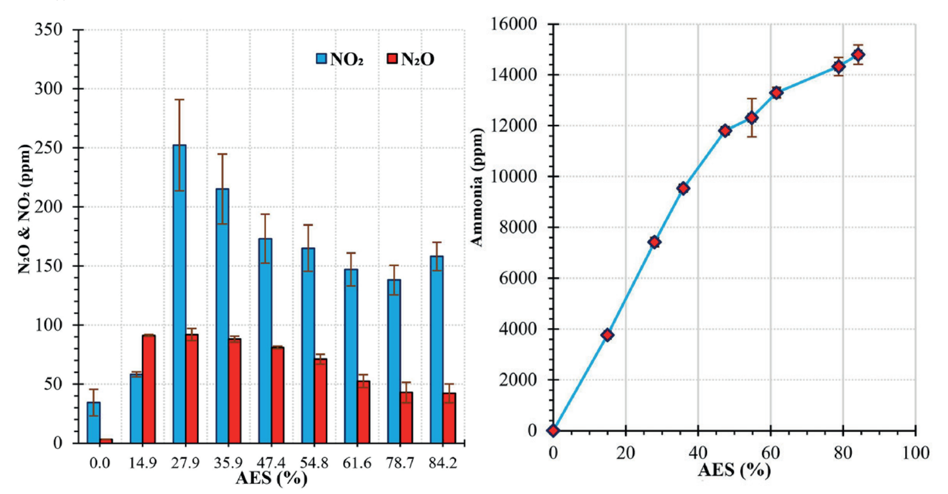
To better understand and fundamental changes in combustion characteristics, in-cylinder pressure and heat release rate (HRR) plots show a shift from the traditional two-stage diesel injection mode characterized by premixed combustion followed by prolonged diffusion combustion to a mode dominated by a high-intensity premixed flame front. The cooling impact of ammonia injection causes the start of combustion (SOC) to be delayed as the AES ratio rises. However, the premixed ammonia-air mixture reacts quickly after being ignited by pilot diesel spray, producing a narrow, highly concentrated heat release rate (HRR) peak roughly at top dead center (TDC). The peak pressure rise rate (PRR_{max}) is greatly increased despite the fact that this quick heat release reduces the entire combustion period (from 56 CAD to 24 CAD). High HRR values associated with high AES ratios pose operational challenges, such as knocking, structural vibration, and noise. Therefore, extending the stable operating range at high AES ratios requires strategies such as split pilot injection or injection timing optimization.

Specifically, the increase in ignition delay from 8.7 to 15.9 CAD at higher AES levels, despite the reduction in combustion duration from 56 to 24 CAD, shows that the upper replacement limit is governed by ignition and combustion stability, rather than by ammonia fuel supply capacity. Ammonia can supply most of the energy, but when there is not enough pilot diesel to maintain timely ignition, combustion quality starts to decline.



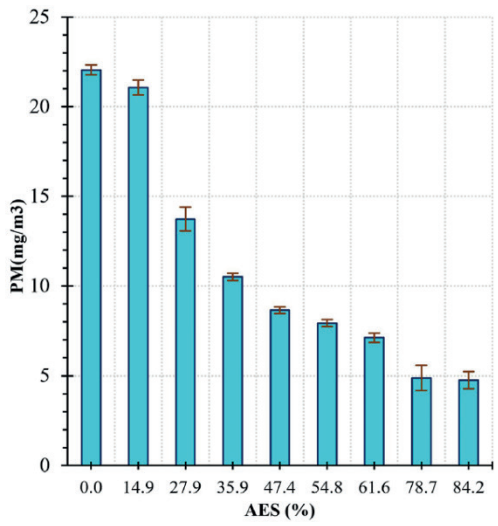
(a) CO_2 and H2O emissions of ammonia-fueled diesel engine for various ammonia energy shares.

(b) Effects of ammonia NO emissions in the ammonia-fueled diesel engine.



(c). NO_2 and N_2O emissions in the ammonia dual fuel diesel engine.

(d). Unburnt ammonia for various ammonia/diesel ratios in the ammonia-fueled diesel engine.



(e). Particulate matter emission for different ammonia energy shares.

Figure 4. Emissions trade-off by ammonia energy share[3]

The emission trade-offs are shown in figure 4, where the general trend reflects both the advantages and disadvantages of ammonia replacement. As the AES ratio increased, carbon-related emission indicators significantly decreased. Additionally, particulate matter emissions significantly decreased, CO levels declined from 7592 ppm to 140 ppm, and CO_2 concentrations dropped from 7.0% to 0.9%. These modifications prove the efficiency of ammonia-diesel co-combustion in reducing emissions of soot and carbon. The emission advantages were especially noticeable during the early stages of this trend.

On the other hand, problems with emissions based on nitrogen were discovered in the next stage. At low AES levels, NO concentration first dropped from 831 ppm to 491 ppm. However, when AES ratio climbed further, NO concentration went up to 2359 ppm. Unburned ammonia emissions increased from 7 ppm to 14800 ppm at the maximum AES level, while N_2O concentrations increased from about 3 ppm in diesel-only mode to over 90 ppm at the highest AES level. The reduction in carbon-based emissions is attributed to the increased use of AES, replacing carbon containing diesel-fuel with carbon-free ammonia. However, the rise in NO, N_2O, and unburned ammonia at high AES levels reflects the more complex nitrogen conversion pathways and incomplete ammonia oxidation processes associated with ammonia combustion processes.

Table 1. Comparison of the engine performance before and after optimization.[5]

Metric	Baseline	Optimized
ITE (%)	~46	50.3
NH_3 emissions (g/kWh)	~36	14.5
N_2O emissions (g/kWh)	~0.40	0.17
GHG (g/kWh)	~600	195

Table 1 compares the baseline operation mode with the optimized mode, which has a high ammonia usage rate. The overall pattern suggests that optimization allows the engine to go from a limit-constrained operating state to a more useful mode with a greater rate of fuel substitution. In comparison to diesel-only operation, the optimized scenario produced a peak ITE of 50.3% with an ammonia energy share of 70%, NH3 emissions of 14.5 g/kWh, N_2O emissions of 0.17 g/kW.h, and a 67.5% decrease in greenhouse gas (GHG) emissions. These findings show that maintaining the operational viability of engines powered by ammonia-diesel mixtures requires modification of parameters, injection timing, and intake conditions.

High diesel injection pressure and early injection timing allow for improved pre-mixing and a shorter ignition delay, increasing efficiency and lowering the net GHG effect. However, table 1 shows that even after optimization, ammonia slips and N_2O emissions are still important limiting factors and are not eliminated. The general implication from table 1 is that ammonia is the most successful when the dual-fuel combustion process is actively optimized, as compared to just functioning as a replacement for traditional fuel.

Advanced combustion strategies for high AES operation

Recent studies indicate a shift toward combustion strategies such as premixed charge compression ignition (PCCI), reactivity-controlled compression ignition (RCCI), diesel pilot injection, ammonia stratification, and direct ammonia injection to enhance operational performance at high fuel requirement ratios. These methods aim to maintain the ignition sensitivity required for ammonia-air mixtures while avoiding excessive mixing that leads to uncontrolled ammonia slip.[2]

Looking more closely, some recent research in the direct injection of liquid ammonia into the combustion chamber reveals that this injection strategy can significantly expand the achievable AES while maintaining combustion stability at low loads. Mi et al. found that controlling the timing and volume of high-pressure liquid ammonia injection in a direct-injection ammonia-diesel dual-fuel engine resulted in larger ammonia energy shares and more stable performance than low-pressure port injection approaches. This technique affects NOx and N_2O production pathways. These findings indicate that for heavy-duty CI engines with configurations comparable to those studied by Nadimi et al., the next step is to research a high-pressure dual-injection strategy that combines direct in-cylinder ammonia delivery with port injection. This arrangement allows for accurate control over ammonia delivery time, local equivalence ratios, and cooling effects. It also evaluates if stability limitations at high AES and N_2O emission concerns may be relocated to more favorable operating conditions.[6]

Several investigations have also examined ammonia-diesel operation in RCCI mode as a route to lower-temperature combustion and reduced NOx emissions. While ignition delay and combustion duration are still sensitive to ammonia fraction and phasing, numerous studies of single-cylinder heavy-duty engines running with port-injected ammonia as the low-reactivity fuel and in-cylinder diesel injection as high-reactivity pilot show that RCCI combustion can achieve significant CO_2 and NOx reductions while maintaining an acceptable load range. An extension pathway is to apply similar AES values in a RCCI framework, using split diesel injection and controlled manifold ammonia supply to shape the reactivity distribution, since the current dual-fuel results already quantify the effects of AES on ignition delay, combustion phasing, and nitrogen-containing emissions. This could help determine whether advanced combustion phasing control can further improve the observed emissions trade-off between CO_2 reduction and NH3/N_2O formation. It would also enable direct comparison between conventional dual-fuel and RCCI operation at comparable ammonia substitution ratios.[7]

Research by Zhang et al. demonstrated that combining high diesel injection pressure with early injection timing improves pre-mixing, shortens ignition delay, and boosts ITE. This results in a more optimized engine operating map, allowing ammonia to supply the majority of the energy without compromising combustion stability[5].

First, to achieve reliable ignition, the amount of pilot diesel and injection timing must be regulated using a systems approach. Second, because these conditions lead to ammonia slip, the ammonia delivery mechanism must prevent excessive cylinder wall wetness or fuel entrapment in cracks. Third, as an ideal configuration at one operating point may result in instability at another, engine operating maps should be assessed throughout ranges of loads, speeds, and equivalence ratios rather than at a single circumstance. Lastly, as raw engine exhaust concentrations run the danger of significantly exceeding allowable limits, post-treatment or exhaust control measures may be necessary if ammonia slip stays high.

Conclusion

For heavy-duty CI (compression-ignition/diesel) engines, ammonia is a viable carbon-free fuel option as long as it is used in a regulated dual-fuel strategy. According to research, ammonia can dramatically reduce CO_2 and particulate matter (PM) emissions while also improving thermal efficiency. To prevent excessive ignition delay, ammonia slip, NO and N_2O emissions, the engine must be precisely adjusted.

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