Evaluation of E10 and NH3 Co-fuelling in a Modern Spark Ignition Engine

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ABSTRACT

Ammonia (NH_3) is emerging as a potential favoured fuel for longer range decarbonised heavy transport, particularly in the marine sector, predominantly due to highly favourable characteristics as an effective hydrogen carrier. This is despite generally unfavourable combustion and toxicity attributes, restricting end use to applications where robust health and safety protocols can always be upheld. In the currently reported work a spark ignited thermodynamic single cylinder research engine equipped with gasoline (E10) direct injection was upgraded to include gaseous ammonia port injection fuelling, with the aim of understanding maximum viable ammonia substitution ratios across the speed-load operating map. The work was conducted under overall stoichiometric conditions with the spark timing re-optimised for maximum brake torque at all stable logged sites. The experiments included industry standard measurements of combustion, performance and engine-out emissions (including NH_3 "slip"). With a geometric compression ratio of 12.39:1 it was found possible to run the engine on pure ammonia at low engine speeds (1000-1800rpm) at low to moderate engine loads in a fully warmed up state (e.g. linear low load limit line from 1000rpm/6bar net IMEP to 1800rpm/9bar net IMEP). When progressively dropping down below this threshold load limit, an increasing amount of gasoline co-firing was required to avoid engine misfire. All metrics of combustion, efficiency and emissions tend to improve when moving upwards from the threshold load line. Due to the favourable anti-knock characteristics of NH_3, pure ammonia operation was up to 5% more efficient than pure E10 operation under stable operating regions. A maximum net indicated efficiency of 40% was achieved at 1800rpm 16bar IMEPn, with efficiency tending to increase with speed and load. For co-fuelling of E10 and ammonia in a pure ammonia attainable operating region, it was found that addition of E10 improved the combustion, but these improvements were not sufficient to translate into improved thermal efficiency. Emissions of NH_3 slip reduced with increased substitution of E10, albeit with increased NOx. However, the reduction in NH_3 slip is nearly 10 times the increase in NOx emissions. Comparing pure NH_3 and pure E10 operation, NOx reduces by ~60% when switching from pure E10 to pure NH_3 (associated with longer and cooler combustion). Future work will be concerned with detailed breakdown of individual NOx species together with measuring the impact of hydrogen enrichment.

1 INTRODUCTION

The transportation sector is going through a renaissance in response to increasing pressures from global governments and society to reduce emissions of greenhouse gases and other pollutants resulting from the use of fossil fuels for power. While electrification is often the preferred solution to tackle this challenge, relative immaturity of battery technology, combined with associated lack of energy density, make full electric propulsion unsuitable for heavy transport applications such as marine, off-road, rail and freight.

Ammonia (NH_3) has gained significant interest in recent years, both as a decarbonised energy vector and efficient hydrogen carrier. Volumetrically, liquid NH_3 can store ~45% more hydrogen than liquid hydrogen. Furthermore, NH_3 can be inexpensively stored as liquid (at -33°C at 0.1MPa or 0.86MPa at 15°C) and conveniently transported. Such promising characteristics of NH_3 have led many researchers to believe ammonia could become a key fuel for heavy transport provided key challenges around slow combustion and emissions control can be overcome [1,2].

2 RELATED WORK

The concept of using NH_3 as a fuel in internal combustion engines can be traced back nearly a century, where it was used to run buses in Belgium during the 2nd World War [3]. This was followed by extensive research in the mid-1960s, where experiments were carried out in both Compression Ignition (CI) and Spark Ignition (SI) engines. Due to the high auto ignition temperature of NH_3, pure ammonia operation in CI engines is only possible with very high compression ratio (e.g. ~35:1) [4]. As a result, most studies in CI engines focus on "dual fuel" operation, where a pre-mixed ammonia-air mixture is ignited by a pilot fuel of low auto ignition temperature and favourable cetane rating.

The dual fuel approach has been extensively researched with various fuels including diesel, dimethyl ether, kerosene and amyl-nitrate [5–12]. However, the added complexity of an additional fuel circuit, coupled with difficulties in operating the engines under throttled conditions and high carbon content of the pilot fuel, makes this solution less attractive compared to SI engines. Compared to compression ignition, pure ammonia operation can be achieved in SI engines at considerably lower compression ratios as reported by Starkman et al. as early as the 1960s [13]. Pearsall et al. [10] investigated the operation with ammonia in both types of engines and recommended a high compression ratio (e.g. [12-16]) SI engine as an ideal solution.

While better than compression ignition, the relatively poor premixed combustion characteristics of NH_3 (see Table 1) makes it challenging to operate a SI engine with pure NH_3 at low loads. However, several strategies can be considered, such as increasing the effective compression ratio, supercharging (potentially without charge-air cooling), high ignition energy and co-fuelling with a faster burning sustainable fuel (s). Of these solutions, co-fuelling with hydrogen has been more extensively studied due to excellent combustion characteristics combined with the ability to produce the hydrogen onboard via NH_3 "cracking".

Morch et al. [14] investigated the combustion of NH_3 at different hydrogen substitution levels and concluded that ~10% volume substitution yielded maximum thermal efficiency. Further to this, Firgo et al. [15] investigated ammonia-hydrogen co-fuelling at various speed/load conditions and concluded that combustion improvement from hydrogen enrichment had reduced impact on engine speed extension compared to engine load. They further calculated the minimum amount of hydrogen energy required for stable combustion to be roughly ~7% for full load and ~11% for part load conditions. These researchers also investigated the feasibility of using exhaust gas heat to crack NH_3 on board and confirmed that hydrogen can be produced via the solution, however, the higher combustion temperatures required for the cracker resulted in significantly higher NOx emissions [16]. Recently investigations conducted Lhuillier et al. [17] and Mounaïm-Rousselle et al. [18] in modern SI engines also concluded that the combustion of NH_3 can be greatly improved by small amounts of hydrogen (~10% vol) allowing the engine to operate at various loads and engine speeds ranging from 650rpm to 2000rpm.

Table 1 Combustion Characteristics of Ammonia and Hydrogen [19–23]

Species	**Hydrogen**	**Ammonia**	**Gasoline**
Chemical Formula	H_2	NH_3	$C_nH_{1.87n}$
LHV [MJ/kg]	120	18.8	44.5
Laminar Burning Velocity @ λ=1 [m/s]	3.51	0.07	0.58
Auto-ignition Temperature [K]	773-850	930	503
Research Octane Number	>100	130	90-98
Flammability Limit in Air [vol. %]	4.7-75	15-28	0.6-8
Quench Distance [mm]	0.9	22.07	1.98
Absolute Minimum Ignition Energy [mJ]	0.02	8	0.1

Gasoline has also been studied extensively as a combustion promoter for NH_3 in SI engines, notably investigated by the CFR research group. Grannell et al. [24] investigated the fuel limits and efficiency of ammonia-gasoline co-fuelling and concluded that ammonia can replace most of the gasoline energy above 4bar IMEPn, with the amount of gasoline needed reducing with increasing engine load and speed. Interestingly, their work with various compression ratios didn't yield improvements in gasoline displacement or thermal efficiencies. Ryu et al. [25] investigated the direct injection of gaseous NH_3 into a Port Fuel Injected (PFI) gasoline engine and concluded that the long injection times needed for NH_3 negated any benefits of direct injection compared to PFI systems [26]. These researchers further conducted experiments with direct injection of cracked ammonia and found that the exhaust heat can be used to crack NH_3 on board without having significant impact on the performance and emissions of the engine. Haputhanthri et al. [27] studied the combustion of ammonia/gasoline emulsified mixtures and found that ammonia can be dissolved into gasoline using emulsifiers like ethanol and methanol and that the composite fuel was capable of improving the performance of engine at high load conditions.

The currently reported work involved experimental research using a modern spark ignited single cylinder engine operating on NH_3 and gasoline (E10) over a range of speed and load points with the aim of improving understanding of the maximum viable substitution of NH_3 across the operating map. The goal was to undertake a baseline analysis in a modern high performance gasoline engine equipped with a modern combustion chamber layout and durable high energy ignition system designed for highly downsized SI engines (e.g. >30bar IMEP).

3 EXPERIMENTAL SETUP

3.1 ENGINE HARDWARE

The experiments were undertaken in an externally boosted SI research engine which was a single cylinder derivative of the MAHLE Powertrain "DI3" demonstrator engine. The engine was equipped with a central spark plug and side mounted gasoline direct injector located under the intake valves for delivering standard pump grade E10. Ammonia was delivered at the port via an upgraded manifold using a prototype Clean Air Power port fuel injector. The engine was also equipped with hydraulic fully independent variable valve timing to enable optimisation of valve timing and overlap. Set out in Table 2 are the key characteristics of the engine.

Table 2 Engine hardware specifications

Parameters	Value
Engine Type	Four Stroke Single Cylinder Spark Ignition
Displaced Volume [cc]	400
Stroke [mm]	73.9
Bore [mm]	83
Compression Ratio	12.39
Number of Valves	4
Valvetrain	Dual Independent Variable Valve Timing (40°CA Cam Phasing)
Fuel Injection Configuration	• Side DI Gasoline (E10) • PFI Ammonia
Max Fuel Injection Pressure [bar]	175 (gasoline)
Cylinder Head Geometry	Pent-Roof (high tumble port)
Piston Geometry	Pent-Roof with Cut-outs for Valves
Ignition Coil	Single Fire Coil, 100mJ, 30kV
Max Power [kW]	40 (gasoline)
Max IMEPn [bar]	30 (gasoline)
Max In-cylinder Pressure [bar]	120
Max Speed [rpm]	5000
Boost System	External Compressor (Max 4barA)
Control System	MAHLE Flexible ECU
Interface Software	ETAS INCA

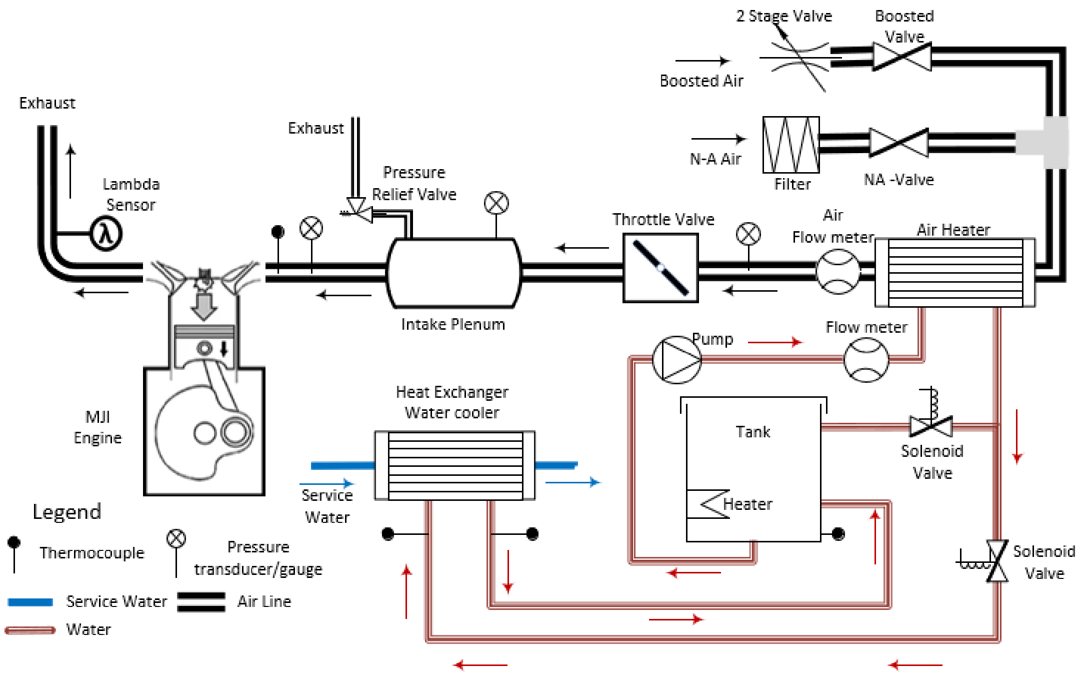

Figure 1 Schematic of the test rig gas path and coolant control

Schematics of the intake air system and the ammonia supply system are shown in Figures 1 and 2 respectively. The engine could be operated as either naturally aspirated or boosted using an external compressor rig providing up to 4barA boost pressure. The temperatures of intake air (45^{0}C), engine coolant (95^{0}C) and oil (95^{0}C) were maintained at a constant value (±1°C) using dedicated conditioning circuits, furthermore, surge tanks were added to both the intake and exhaust to minimise the effects of unwanted gas pressure fluctuations.

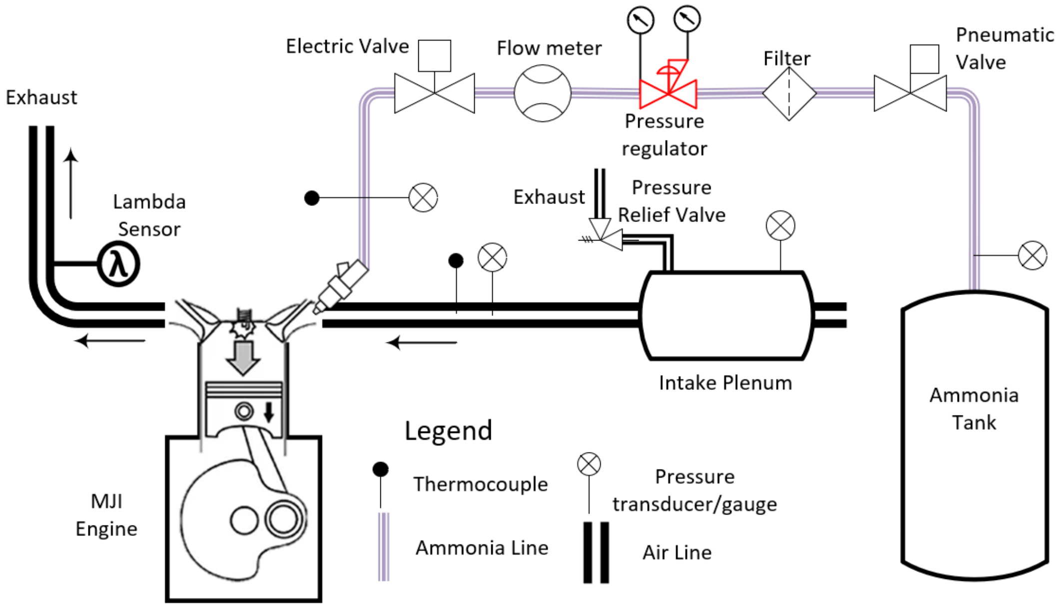

Figure 2 Schematic of the engine fuel supply line

The ammonia was supplied to the engine in gaseous phase using a dedicated port injector supplied by Clean Air Power. The NH_3 was stored in liquid vapour equilibrium via a drum, with the pressure differential between the intake manifold and vapour pressure inside the drum used to drive the supply of ammonia to engine. The flowrate of NH_3 was measured using a Coriolis flowmeter procured from micro-motion (maximum flow rate error of 1% at the minimum flow rates reported). Electrically controlled safety valves and nitrogen based purging were added to the supply line to isolate the ammonia supply in the case of an emergency. For the gasoline supply, an AVL 735 fuel balance unit was used to measure the gasoline (E10) flowrate and condition the gasoline temperature (20°C set point) before being fed to a high pressure fuel pump at constant supply pressure via a fuel regulator.

In-cylinder pressure was measured using a Kistler 6045-B piezo electric pressure transducer working through a AVL Micro-FEM amplifier, and fully calibrated to industry standards via a dead weight tester. The intake and exhaust pressures were also measured using Kistler's 4045A and 4011 piezo resistive transducers. The engine-out emissions were measured using a series of dedicated analysers from the Signal group, in addition to industry standard emissions (NOx, CO_2, CO, THC and O_2) ammonia "slip" emissions (unburned NH_3 in the exhaust) were also measured based on a new Signal unit. The details of the emission analysers are summarised in Table 3. All measurements were recorded and processed using a bespoke National Instruments Data Acquisition system. The data from pressure transducers were recorded at a resolution of 0.2 Crank Angle degrees (CAD) using a Hohner 3232 optical encoder, which was synchronised using an AVL capacitive probe. During testing 300 cycles of pressure data were recorded. Mass fractions burned were evaluated on a qualitative basis using one dimensional heat release analysis. Other "steady state" temperature, pressure and flow measurements were taken at a frequency of 10Hz.

Table 3 Details of the emission analysers

Equipment	Gas	Operating Principle	Dynamic Range	Accuracy / Error(%)
4000 VM	NOx	Chemiluminescence	0-1000 ppm	Better than +1% range or ±0.2 ppm whichever is greater.
8000 M	O2	Dumbbell paramagnetic sensing	0 -5 %, 0 -10 %, 0 -25 %	±0.01 %O2.
S4 Nebula	NH_3	Tuneable Diode laser Spectrometry	1ppm -10,000 ppm	±2% of FDS
3000 HM	THC	Flame ionisation detector	0-10000 ppm	Better than ±1 % range or ±0.2 ppm whichever is greater.
7000 FM	CO, CO2	Infra-red gas filter correlation technique	100-10000 ppm Or 1-100 %	Better than ±1 % of range or ±0.5 ppm whichever is greater.

3.2 TEST PLAN

Since practical applications of ammonia are expected to be in low-to-medium speed heavy duty engines, the test points were selected to cover typical peak power speed ratings for such engines. The tests were conducted at 1000, 1400 and 1800rpm with the engine load varied from 4 to 12bar net Indicated Mean Effective Pressure

(IMEPn). The aim of the tests was to determine the pure ammonia speed-load map and associated impacts upon combustion, performance, fuel economy and emissions with and without co-fuelling. The co-fuelling required was evaluated by undertaking ammonia "displacement sweeps"; with the engine first fired using pure E10 and NH_3 progressively added until an upturn in combustion stability occurred (with repeat logs around this upturn to establish the maximum possible NH_3 substitution and the upper limit set to a coefficient of variation in IMEP of >3%). All logs were obtained under stoichiometric conditions with the spark timing set to Maximum Brake Torque (MBT). In early work it was proposed that slightly rich running might aid NH_3 displacement (due to slightly higher laminar burning velocity) but this was not found to be the case; with the engine misfiring more easily when attempting to operate slightly richer when at the substitution ratio limit due to the relatively low relative air-to-fuel ratio of NH_3 and significant reduction in the ratio of specific heats (and hence gas temperature) "over-ruling" relatively small increases in laminar burning velocity when slightly rich. Such effects were previously insinuated by the chemical modelling work of Kobayashi et al.[28].

The engine settings used for the tests are set out in Table 4. In addition to these settings, the valve timing was fixed for the tests, however, the overlap was adjusted from 37 Crank Angle Degrees (CAD) to 24 CAD for the 1000rpm tests as the slow speed combined with high boost pressure otherwise resulted in significant ammonia slip, due to high apparent cylinder scavenging at this speed.

Table 4 Engine settings for substitution tests

Settings	Values
Operating Temperature (Coolant & Oil) [^{0}C]	95
Spark Timing	Maximum Brake Torque (MBT)
Air-fuel Equivalence ratio	1
E10 Injection Start angle [CAD BTDCf]	310
Ammonia Injection End angle [CAD BTDCf]	400
Inlet air temperature [^{0}C]	45
Ammonia rail pressure [barG]	3-5
Ammonia Feed Temperature [^{0}C]	27 - 30
E10 Temperature [^{0}C]	20

4 RESULTS

4.1 MAXIMUM DISPLACEMENT OF AMMONIA

The results of the maximum ammonia percent displacement at various test points are shown in Figure 3.

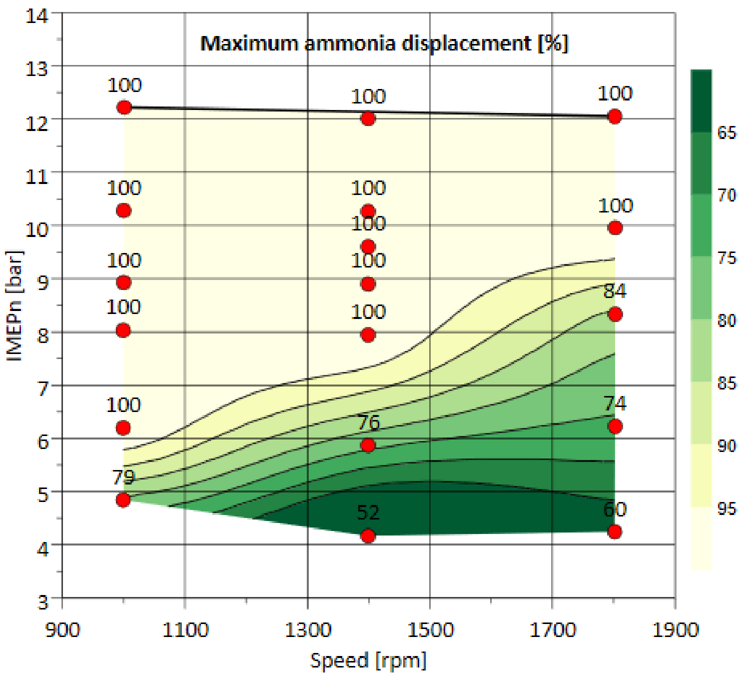

Figure 3 Maximum substitution of ammonia achieved at different load points (λ=1, MBT spark timing)

The engine was capable of operating with pure NH_3 at relatively moderate engine loads. Furthermore, ammonia constituted most of the fuel energy across the map, validating the prior findings of Granell et al. [24]. The 100%

substitution isoline follows a near-linear pattern, with the threshold load required to operate on pure NH_3 increasing by 2bar IMEP for an increase of 400rpm in engine speed. This direct relation of threshold engine load and engine speed was also observed by Mounaïm-Rousselle et al. [18] in their work on ammonia SI engines. This trend is despite increasing gas temperatures at higher speeds and illustrates the dominance in lower speed providing more time for combustion to occur despite the fact the in-cylinder and exhaust gas temperatures usually increase with engine speed (for a given load). The impact of increasing in-cylinder turbulence with higher speed remains unknown and will be qualified in future work.

4.2 GENERAL TRENDS OF PURE AMMONIA OPERATION IN AN SI ENGINE

4.2.1 COMBUSTION

Figure 4 shows the spark timing required to achieve MBT and the corresponding stability of the engine at the tested points. Examining the map it is evident that the engine operation improves considerably as the load increases from the threshold load for pure ammonia operation.

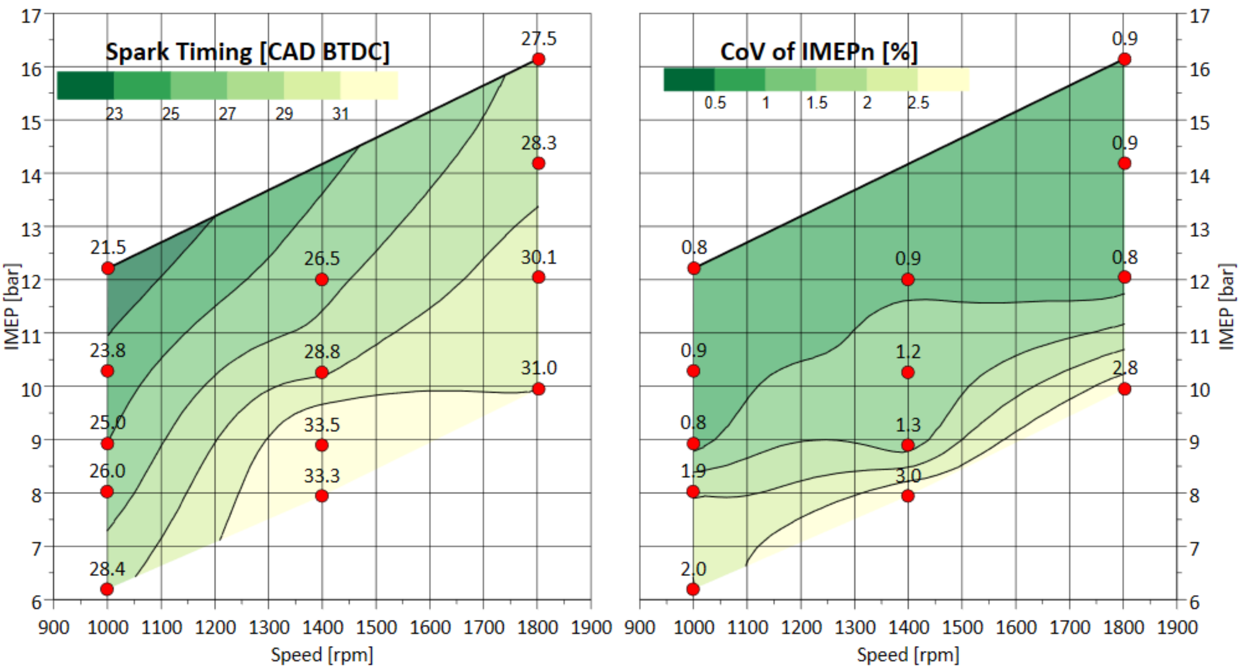

Figure 4 Spark timing and CoV of IMEPn of pure ammonia test points

The spark advance required to achieve MBT reduces with increase in load or reduction in engine speed, similarly the engine operation becomes notably more stable beyond 4bar IMEPn at all engine speeds. The mass fraction burned at the various test points is shown in Figure 5, where the "flame development phase" (0%-10% MFB) followed a similar trend to the spark timing. However, the "combustion phase" (10%-90% MFB) variation was relatively smaller for the test points. Moreover, the flame development phase was similar to the combustion phase at low speeds and became larger than the combustion phase as the speed increased. In other words, nearly 50% or more of the total combustion duration encompasses the flame development phase. The lack of variation in the combustion phase with speed could be a direct result of increased turbulence enabled by a high tumble head used in the study (to be confirmed in future optical and CFD analysis work).

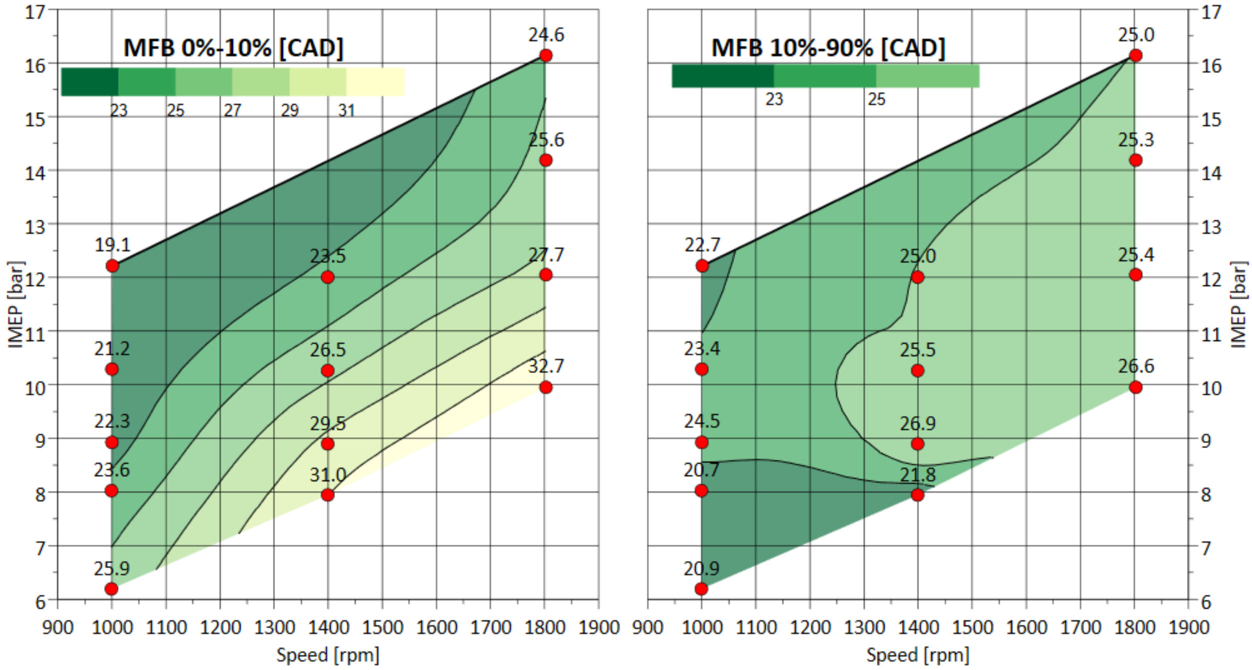

Figure 5 Variation of combustion metrics 0%-10% MFB and 10%-90% MFB for pure ammonia combustion at various speeds and loads

4.2.2 EFFICIENCY

The variation in net Indicated Thermal Efficiency (ITE) in the test region for pure NH_3 operation and pure E10 operation is set out in Figure 6. Pure NH_3 operation is considerably more efficient than E10 in the test region by virtue of ammonia having a high octane rating and low air-fuel ratio, both of which combined enabled the engine to be operated at MBT with high loads, allowing the engine to achieve efficiencies as high as 40% at 1800rpm/16bar IMEPn.

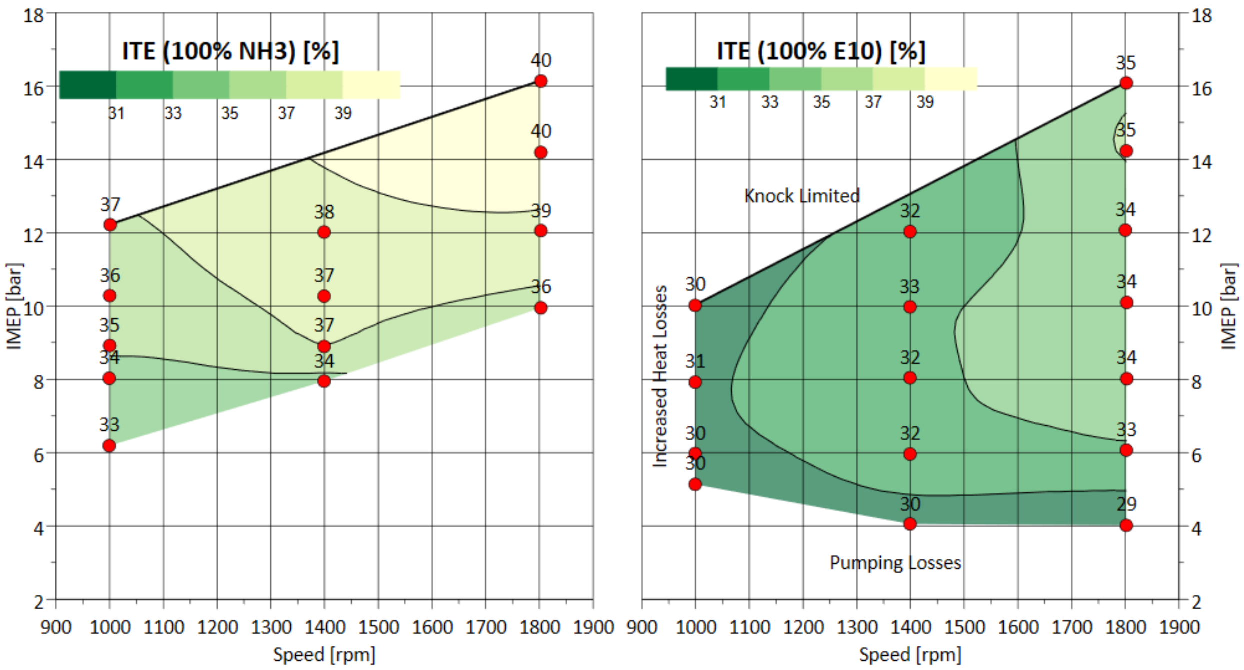

Figure 6 ITE of 100% NH_3 vs 100% E10 operation

Examining the variation of ITE for pure NH_3 operation, the efficiency improves with increase in speed and load, between them the impact of load increase is larger than that of engine speed. This variation suggests losses from increased heat rejection, pumping and knock that govern E10 operation in the test region do not directly apply to pure NH_3 operation, or these factors have minimal impact on the ITE (potentially related to the ability to achieve MBT across the map)

4.2.3 EMISSIONS

The NOx and NH_3 slip emissions from the engine operating on pure NH_3 are set out in Figure 7. NOx emissions remain relatively similar across the tested region, with the values increasing closer to the threshold load points mainly due to the advanced spark timing aiding the NOx formation via increased cylinder temperature. However, the emissions are nearly a third of that produced during pure E10 operation (~3000-4000ppm) under the same conditions.

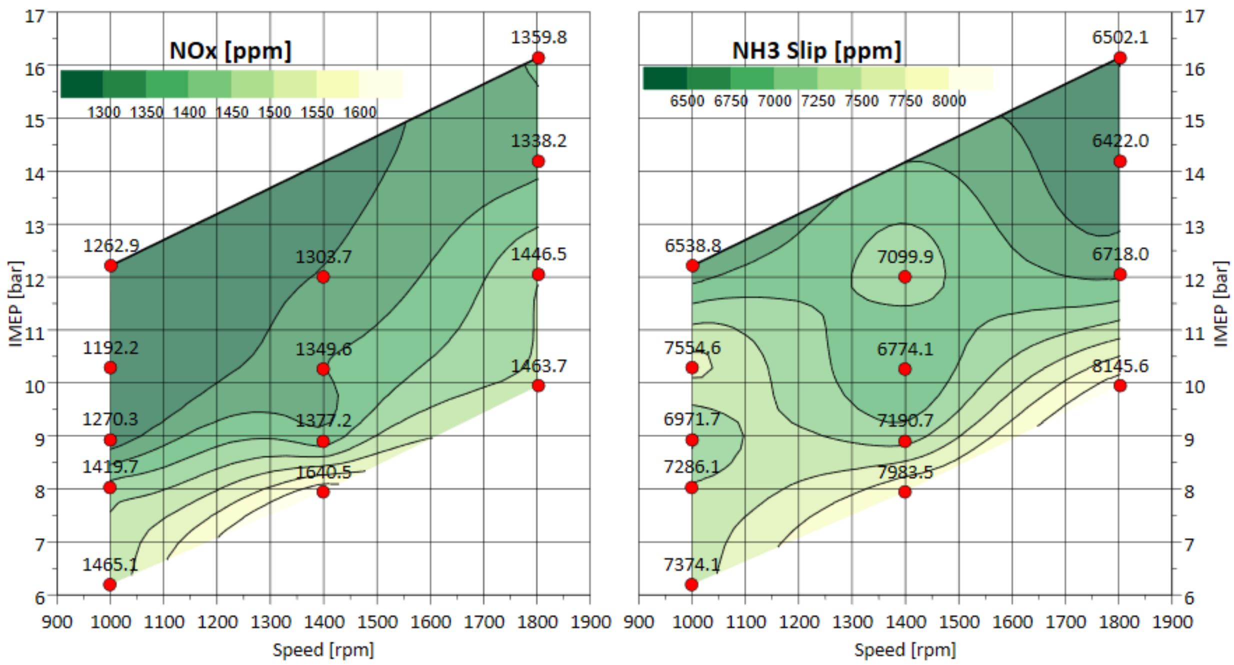

Figure 7 Emissions of NOx and ammonia slip for pure ammonia test region

Similar to NOx, ammonia slip also peaks near the threshold load from the unstable engine operation in those points. While the slip improves with engine stability, there is considerable slip (> 0.5% vol) even in the stable operating points. The recorded NH_3 slip values are comparable to previous studies published by Lhuillier et al and Mounaïm-Rousselle et al [17,29] using similar engines and under similar operating conditions (λ, MBT). The two major causes for the high values of slip are (a) in-cylinder scavenging, pushing part of the injected ammonia in the intake port directly into the exhaust and (b) the incomplete combustion of ammonia trapped in crevice volumes. However, further investigations are necessary to quantify such effects. One of the potential uses of the excessive slip is to clean the NOx via a Selective Catalytic Reduction (SCR) catalyst, potentially eliminating the need for any "AdBlue" (to be confirmed in future work). Moreover, high exhaust gas temperatures could enable the oxidation of excess ammonia within the catalyst as determined by Girard et al [30]. However, the "alpha" ratio (ratio of NH_3 to NOx in ppm) is considerably higher than desired values between 1 and 2, which suggests the need for ammonia scrubber/oxidation catalyst to remove the excess ammonia (with potential trade-offs to be made with N_2O production).

4.3 E10-AMMONIA CO-FUELLING AT HIGH LOAD CONDITIONS

As explained in the previous section, while pure NH_3 operation can be achieved at moderate-to-high load operation, some form of fuel enhancement is needed to stably operate the engine at low loads, idling and cold start. Therefore, additional displacement tests were conducted at a pure NH_3 operational starting point with the aim of understanding if co-fuelling enhances the performance, efficiency or emissions of the engine (despite the fact pure ammonia operation was possible). The tests were conducted at 1400rpm and 10bar IMEPn with the engine settings as previously listed in Table 4.

4.3.1 COMBUSTION

The impact of increased E10 substitution on the stability and spark timing of the engine is shown in Figure 8. Replacing 25% of the energy with E10 improves the stability as well as the spark advance required to achieve MBT. Further substitution of E10, however, did not have any positive impact on the operation of the engine. A similar pattern can also be found with addition of NH_3 to pure E10 operation. the high knock resistance of NH_3 allows the engine to be operated at MBT without knock suppression improving the stability of operation.

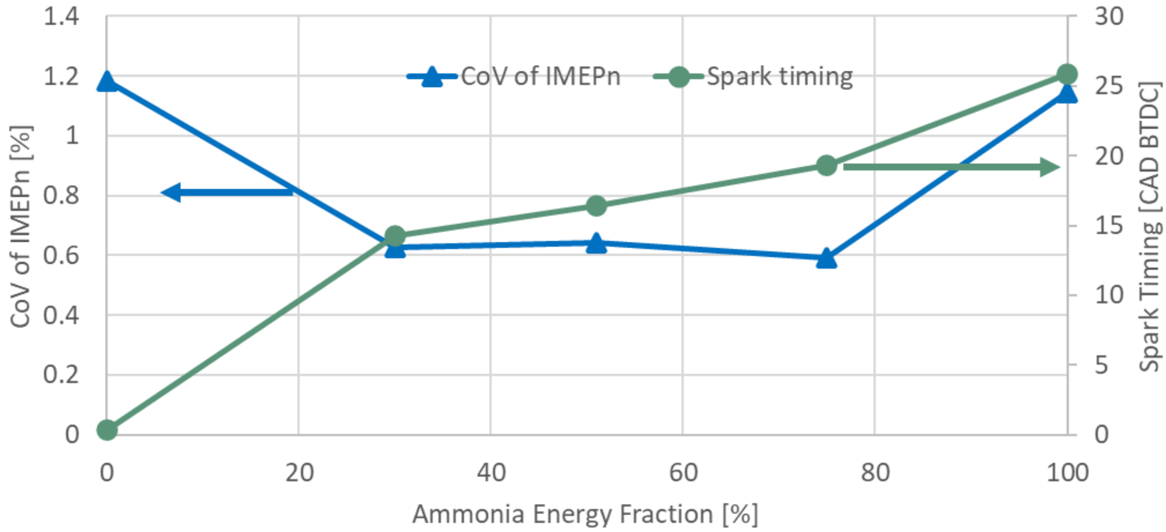

Figure 8 Stability and MBT Spark timing for different levels of E10 substitution

The impact of E10 substitution on mass fraction burned is depicted in Figure 9, where the addition of 25% E10 reduces the flame development phase of combustion by 25%. However, further increase in substitution had reducing impact on the flame development phase. Similar results were also obtained by Mercier et al. [31] in their studies with hydrogen substitution, however, the similar substitution of hydrogen had a bigger impact (~50%) than E10.

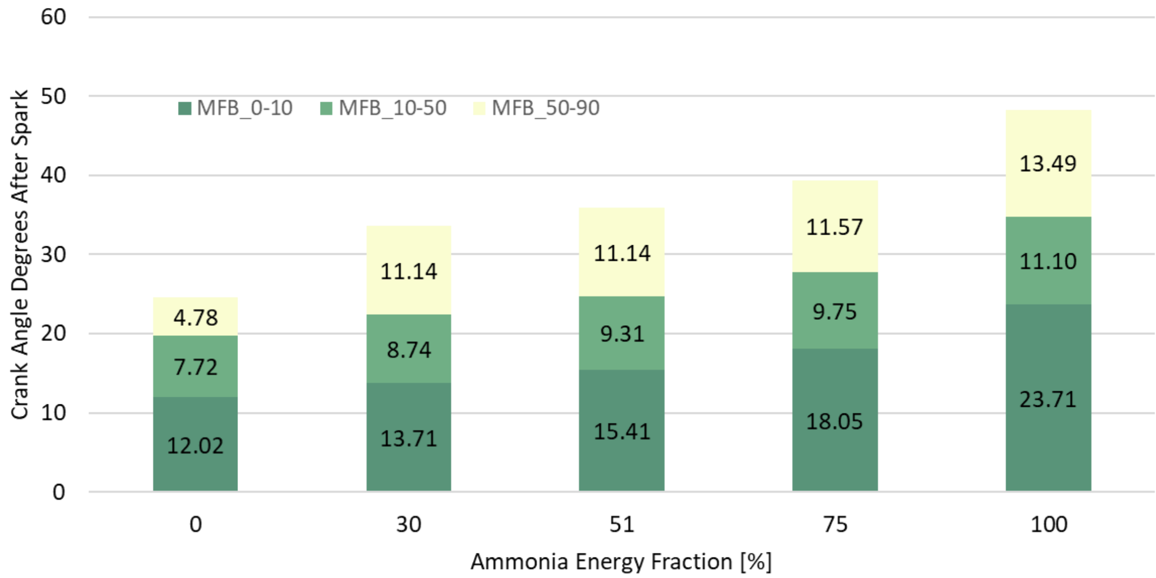

Figure 9 Combustion metrics for different levels of ammonia substitution

Compared to the flame development phase, E10 substitution had minimal impact on the combustion, taking a similar duration as the pure ammonia combustion. Ammonia substitution, however, increases the combustion phase considerably as indicated by increase in values between 0% and 30%. This data showed no benefits in combustion can be achieved by increasing the substitution beyond ~25%.

4.3.2 EFFICIENCY

The values of ITE achieved at different substitution rates are shown in Figure 10, where addition of E10 to the engine reduces the efficiency, however the impact is less than 1% and remains nearly constant in the co-fuelling region. This indicates that the improvements in combustion achieved from E10 substitution increases the ratio of heat losses to work output.

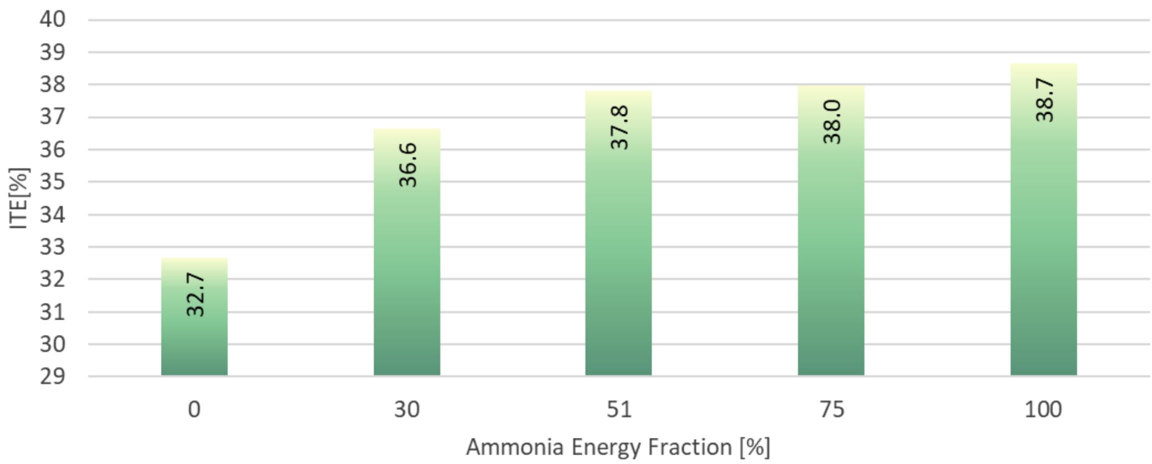

Figure 10 Indicated thermal efficiency achieved at different E10 substitution rates

4.3.3 EMISSIONS

The impact of E10 co-fuelling on NOx and NH_3 slip is shown in Figure 11. While co-fuelling with E10 would add other carbon-based emissions they are not shown here as these emissions simply increased in linear proportion to increased E10 substitution. Compared to pure NH_3 operation, co-fuelled operation decreases the NH_3 slip considerably, partially due to the lower quantities of NH_3 injected and resulting higher cylinder temperatures, as is evident from the increase in NOx values with increased substitution.

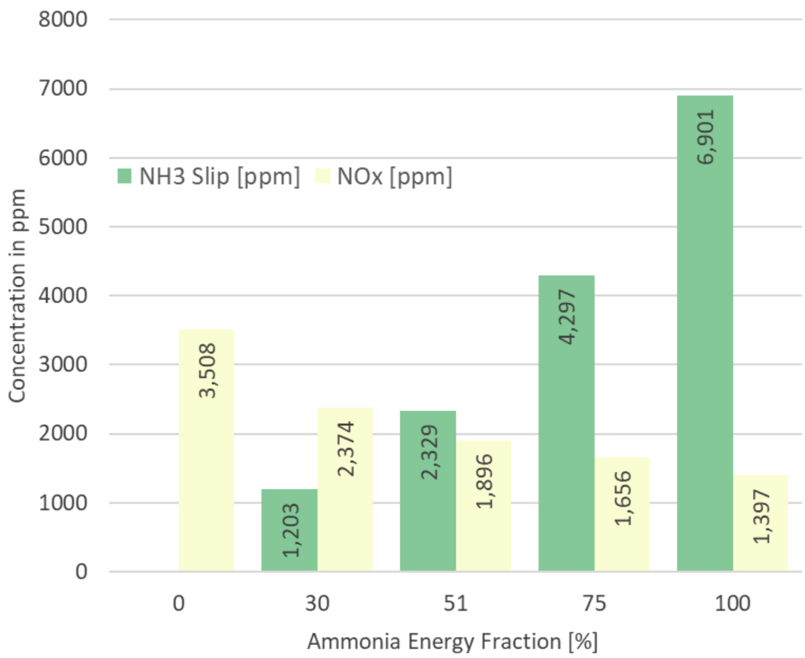

Figure 11 Emissions of NOx and ammonia slip for various E10 substitution rates

Comparing the values of NH_3 slip and NOx emissions, the drop in NH_3 slip emissions is nearly 10 times that the increase of NOx between each level of substitution. Furthermore, comparing pure NH_3 and pure E10 operation, the NOx emissions reduce by nearly 60% from 3500ppm to 1400ppm.

These tests indicate that co-fuelling in the pure NH_3 capable operating region can deliver positive impacts with respect to combustion and emissions without affecting the efficiency considerably. Further investigations with more reactive fuels like hydrogen could yield better results and will be investigated in future work.

5 CONCLUSIONS

This paper detailed experimental work undertaken to assess the feasibility of co-fuelling a modern SI engine with ammonia and E10. The key conclusions for the work can be summarised:

- Under low speeds and a fully warm engine state, the engine can operate efficiently on pure ammonia at low to moderate loads.
- The threshold engine load where pure ammonia operation is achieved reduces with reduction in engine speed, reducing from 10bar IMEPn at 1800rpm to 6bar IMEPn at 1000rpm.
- Stable operation of the engine below the threshold engine load requires co-fuelling with E10, however more than 50% ammonia substitution is achieved at test points above 4bar IMEPn.
- For a given engine speed, the spark advance required to achieve MBT improves as load increases from the threshold load.
- The flame development phase (0-10% MFB) of pure ammonia combustion was identical to, or longer than, the combustion phase (10-90% MFB), with duration reducing with engine load and increasing with engine speed.
- The combustion phase using ammonia has minimal variation with load and speed changes, remaining within 4-5 CAD across the test region.
- The favourable anti-knock characteristics of ammonia enabled higher net indicated thermal efficiency (increased ITE ~5%) under pure ammonia operation compared to pure E10.
- A maximum net indicated thermal efficiency of 40% was achieved at 1800rpm/16bar IMEPn, which could increase with load and speed as heat transfer losses seem to be reduced in the test region due to lower combustion temperatures.
- NOx emissions remained relatively similar (within 500ppm) across the map, with ammonia operation generally resulting in lower NOx emissions (up to 60% reduction compared to pure E10 operation). This potentially indicates remaining significant chemical NOx formation mechanisms, rather than thermal formation mechanisms alone.
- Ammonia slip emissions were relatively high in the tested region (albeit in agreement with reports elsewhere), peaking near the threshold load points due to potentially incomplete combustion. Values for NH_3 emissions remain high (>6000ppm) even at stable operating points.

E10 co-fuelling under the pure ammonia operating region was also investigated in this work, with the following conclusions made:

- The addition of E10 improves the combustion stability, allowing the retardation of spark by ~5 CAD for a substitution of 25% E10 by energy.
- The flame development phase is impacted the most by E10 substitution, reducing by 5 CAD for 25% substitution. The combustion phase however exhibited relatively minimal impact.
- The small improvements in combustion did not translate to improved efficiency, with a decrease in efficiency by of up to1% recorded with increased E10 substitution
- Ammonia slip emissions improve considerably with E10 substitution, partially due to less ammonia being injected and potentially due to higher in-cylinder temperatures.
- Higher in-cylinder temperatures also led to an increase in NOx emissions, even with retarded spark timing.
- The reduction in ammonia slip emissions was approximately 10 times the increase in NOx emissions, reducing from 6900ppm to 4300ppm for an increase in NOx emissions from 1400ppm to 1650ppm for 25% E10 substitution.
- The benefits of E10 co-fuelling reduced beyond 25% substitution, suggesting a maximum limit for fuel enhancers in ammonia combustion.

Immediate future work will focus on co-fuelling with hydrogen, accompanied by detailed breakdown of NOx species (NO, N_2O, NO_2) at varied compression ratios and relative fuel to air ratios. The engine is also being modified to incorporate a higher stroke to bore ratio, better emulating typical heavy duty operation and also enabling higher geometric compression ratios.

6 ACKNOWLEDGEMENTS

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9 DEFINITIONS

CAD : Crank Angle Degree
NH_3 : Ammonia
NOx : Oxides of Nitrogen
SI : Spark Ignition
LHV : Lower heating Value
ITE : Indicated thermal efficiency
MFB : Mass Fraction Burned
CoV : Coefficient of Variance
E10 : Gasoline with 10% ethanol
CI : Compressed Ignition
DI: Direct injection
PFI : Port fuel Injection
BTDC : Before Top dead centre
BTDCf : Before Top dead centre firing
MBT: Maximum Brake Torque
ppm : parts per million
IMEPn : Net Indicated Mean Effective Pressure