CIMAC CONGRESS 23
BUSAN, JUNE 12 – 16, 2023

2023 | 105

Assessment of combustion concepts and operational limits of net-zero carbon fuels

New Engine Developments - Alternative Fuels & Other New Engine Concepts

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ABSTRACT

In order to reduce greenhouse gas emissions, zero carbon and net-zero carbon fuel alternatives such as hydrogen, ammonia, methanol and synthetic LNG will play a crucial role, mainly for shipping, power generation and certain off-road applications. Especially in high power systems, combustion engines fueled by net-zero carbon fuels offer a number of practical advantages such as the use of existing technologies as a basis for robust solutions at acceptable costs.

This paper describes different engine concepts and combustion technologies for net-zero carbon fuels covering high and medium speed 4 stroke engines. The comparison considers mixture formation and working cycle analysis.

Selected single cylinder engine test results in Otto-cycle operation with port gas admission and gas-fed pre-chamber with different ratios of Hydrogen and Natural Gas will be discussed. The results include operational limits (misfire, single stage turbocharging), abnormal combustion phenomena and achieved engine performance. Heat release rates will be compared for further explanation. The impact of increased hydrogen admixing rates to the turbocharger requirements like as boost pressure for constant and low NOx operation and decreased exhaust gas enthalphy will be discussed.

The impact of hydrogen mixing to natural gas and also pure hydrogen onto the measured gaseous emissions will be presented, with focus on standard emissions such as nitrogen oxides, hydrocarbons and carbon monoxide. Potentials of green house gas reduction will be assessed.

Specific emission results like formaldehyde CH2O and nitrous oxide N2O are included in the analysis. Hydrogen slip and/or ammonia slip will be quantified as well. The emission values depending on different engine operation boundaries like A/F ratio, hydrogen content and intake temperature will be discussed.

Finally, the latest performed measurement results with net-zero carbon fuels and combustion concepts will be reviewed. The paper will conclude with the basic requirements for the exhaust gas after treatment systems and further engine development steps. An outlook on the potential of future technologies such as low pressure and high pressure direct hydrogen injection as well as direct injection of ammonia and methanol is given.

1 INTRODUCTION

In order to reduce greenhouse gas emissions, zero carbon and net-zero carbon fuels such as hydrogen, ammonia, methanol and synthetic liquefied natural gas will play a crucial role. The further use of engines for high power applications such as shipping, power generation and certain off-road applications, fueled by carbon neutral fuels offers practical advantages and allows a carry-over of existing base technology. If only minor adaptations are required, robust solutions at acceptable costs are depictable.

The focus of this paper will be the comparison and ranking of technology options for hydrogen and ammonia as suitable for both, High-Speed and Medium-Speed large bore engines.

Experimental results from a pre-chamber spark ignited Medium Speed single cylinder engine with port gas admission are described in detail, covering a variation of the ratio of hydrogen and natural gas mixed upstream of the engine.

Initial medium speed engine test results covering diesel ignited ammonia operation are discussed and compared with results from corresponding tests as performed on a High-Speed engine.

Operational restrictions are analyzed, specific challenges are described, and the corresponding further development needs are highlighted.

2 AMBITIONS OF GREENHOUSE GAS REDUCTION

Decarbonization and carbon dioxide (CO_2) / greenhouse gas reduction (GHG) in all energy sectors is currently one of the most important and discussed topics to limit the worldwide average temperature increase.

The blending of hydrogen with natural gas is currently seen as a relatively straightforward, short-term solution to reduce greenhouse gas (GHG) emissions from power plants that run on gas engines. In the mid-term future, engines powered solely by hydrogen will fully realize their zero-emission potential as they do not emit any methane, hydrocarbons, or CO_2. The availability of hydrogen-fueled engines in the energy sector is on the rise [1].

Another promising carbon-free fuel is ammonia. Ammonia is a superior hydrogen carrier compared to liquid hydrogen, making it a highly viable fuel in the context of ongoing searches for suitable Power-to-X energy carriers across various energy production sectors. The maritime industry is particularly interested in ammonia as a future fuel, with a strong potential for use in deep-sea shipping. Development of ammonia engines in both 2-stroke and 4-stroke categories is underway, with the first field engines expected to be available by 2024, for example [2] and [3].

The ambitions for greenhouse gas reduction and carbon neutrality energy sources are defined, for example, in the Paris agreement, IMO and EPA ambitions or local, country specific strategies. IMO is focusing on a greenhouse reduction of 50% by 2050 compared to 2008 values. The reduction of carbon intensity is targeted with 70% in 2050. The projected pathway of IMO is presented in Figure 1 [4]. For the year 2100 a zero-emission target is discussed.

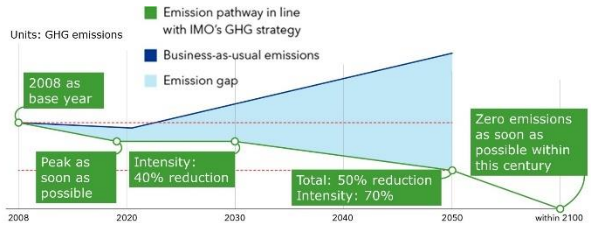


Figure 1. DNV Emission reduction scenario until 2100; [4]

To achieve the reduction goals liquefied natural gas, hydrogen, ammonia and methanol must be introduced into the market. However, until today, no 100% clear picture of future fuel mix in marine or other energy sectors is available. As an example, American Bureau of Shipping (ABS) is expecting following marine fuel mix up to 2050 is shown in Figure 2 [5].

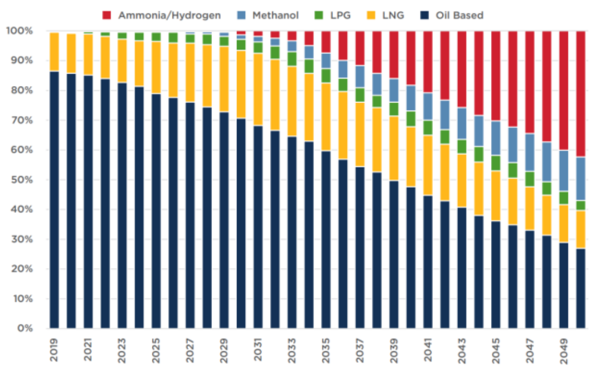


Figure 2. Projected marine fuel use to 2050; Source: ABS zero Carbon outlook (2022) [5]

According to Figure 2, the number of ships fueled by liquefied natural gas (LNG) or capable of using LNG is projected to increase until 2035, after which a slight decrease is expected. It is estimated that LNG will power approximately 12% of ships in 2050. The use of liquefied petroleum gas (LPG) and methanol is also expected to increase, albeit to a lesser extent, until 2050, with a combined market share of 15%. The highest market share is still expected to be held by oil-based fuels, as well as engines fueled by ammonia and hydrogen, with 28% and 42% respectively.

For the usage of future carbon neutral fuels in internal combustion engines and fuel cells some open questions must be answered in parallel.

- Definition of classification rules and guidelines e.g. hydrogen or ammonia (as hydrogen carrier) fueled engines
- Build-up of infrastructure / storage / bunkering capacity in a large scale and in harbours worldwide
- Attractiveness for capital expenditures and operational expenditures (CAPEX / OPEX)
- Minimization of production costs of future fuels (sufficient renewable energy for green produced hydrogen, ammonia, methanol and synthetic fuels)
- Advertising for market acceptance (plant or ship owner) related to new fuels
- Adaption of vehicle design (ship, power plant, locomotive or mining truck/ excavator): larger tanks due to lower volumetric energy in comparison to Diesel fuel; 2nd or 3rd fuel system versus transport capacity – economic point of view of the operators

To demonstrate the applicability of the discussed different fuels, AVL investigated in research and development projects the performance in internal combustion engines.

3 COMBUSTION CONCEPTS

In the last years and months AVL performed extensively measurement campaigns with a variety of combustion systems as well as with various alternative fuels on single cylinder engine test beds [6], [7], [8] and [9]. Figure 3 shows in simplified schemes the investigated combustion systems for future applications with hydrogen, ammonia and methanol.

This paper deals with concept 1 and 2. Both concepts were investigated on a medium speed single cylinder engine. Detailed results with concept 2 and 3 will be presented in paper number 667 [10], investigated on the new AVL high speed single cylinder engine platform [11], [12] and [13]. For concept 4 some details of the measurement results with high pressure direct injection of methanol are collected in paper number 139 [14].

Concept 1 is a typical configuration for stationary medium speed engines used for electric power generation. For safety and engine protection issues, cylinder individual port gas admission is standard. For smaller high speed engines an open-chamber concept is also feasible and available on the market.

Concept 2 is a relatively simple solution for retrofitting to existing Diesel engines. For multi cylinder engine development an ammonia port gas admission valve is recommended. Port gas admission valves can minimize the ammonia slip during the scavenging process.

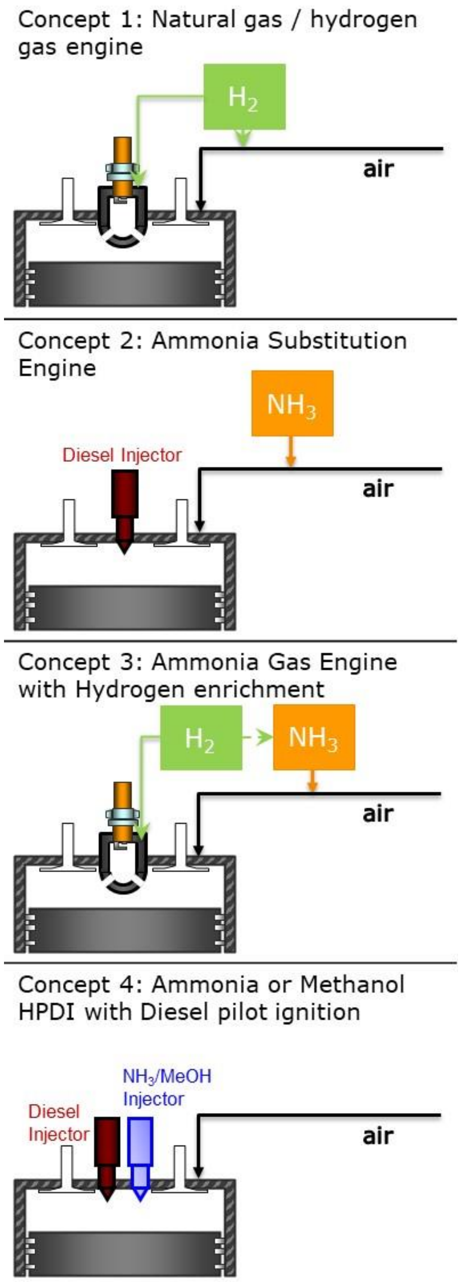


Figure 3. Investigated combustion concepts for alternative fuels

4 TEST RESULTS WITH NATURAL GAS-HYDROGEN ENGINE OPERATION

The test campaigns with mixtures of natural gas, propane and hydrogen, and also pure hydrogen were performed on a medium speed single cylinder engine with a cylinder displacement of approximately 30 liters per cylinder. The main parameters are summarized in Table 1.

Table 1. Engine data – operated with natural gas / hydrogen with spark ignition

Parameter (unit)	Value or remark
Bore (mm)	300
Stroke (mm)	430
Rated speed (rpm)	750
Rated power (kW)	573
Compression ratio	10…>15 Adaptable with shims and/or piston crowns
Valve timings	Miller timing, adaptable with separate intake and exhaust cam segments

A cross section view of the engine is shown in Figure 4 showing also the intermediate bearing and tandem electrical and water dynamometers. Due to the large bore diameter, the selected combustion concept was a gas-fed pre-chamber combustion concept, ignited with a spark plug. With the pre-chamber concept, a strong ignition of the very lean mixture in the main combustion chamber is more reliable. The gas supply to the main chamber was realized with a solenoid activated gas admission valve located close to the intake port. A constant speed setpoint of 750 rpm for genset operation was defined.

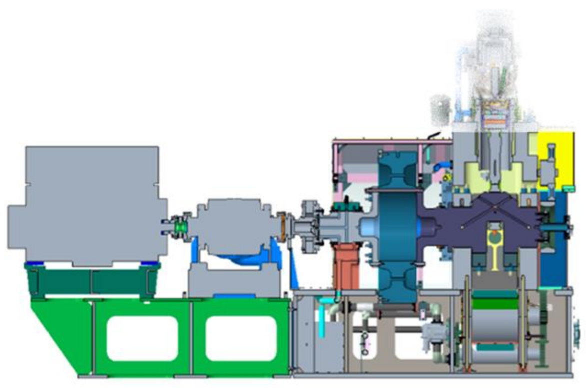


Figure 4. Cross section of the single cylinder engine including flywheel, intermediate bearing, water brake and electrical dynamometer

For the admixing of hydrogen to natural gas, two specifications of natural gas were used. Natural gas with high methane number of approximately 90 and a natural gas propane blend with a moderate methane number of 65.

The demonstrated measurement results are based on geometrical compression ratio (CR) of 10.3:1. Investigations with a higher CR of 11.9:1 were also performed. With the higher compression ratio, engine operation with 100% hydrogen was further limited in load, due to the restrictions given by several combustion anomalies.

Figure 5 displays the measured engine-out emission values at a constant load and speed. At 100% hydrogen operation, very low or no carbon dioxide was detected. Only small amounts of carbon monoxide and total hydrocarbons due to oil in the combustion process were detected. At hydrogen blending rates higher than 40 vol%, nitrogen oxide (NOx) values were below the limit of 100 mg/m³ @ 5% O_2, as referenced to the 44. BImSchV. Zero NOx was even achievable at pure hydrogen operation. Approximately 80 vol% or 50 mass% of hydrogen is required to halve the CO_2 emissions compared to pure natural gas operation. The fraction of unburned hydrogen in the exhaust gas increases with higher blending rates. At pure hydrogen operation, values of 2.5 g/kWh (or ~1800 ppm) were measured with a separate hydrogen emissions measurement device. Carbon monoxide emissions remained constant up to a hydrogen blending rate of 90 vol%, above which the emissions decrease. Total hydrocarbon emissions remain constant up to a 60 vol% blending rate before then decreasing.

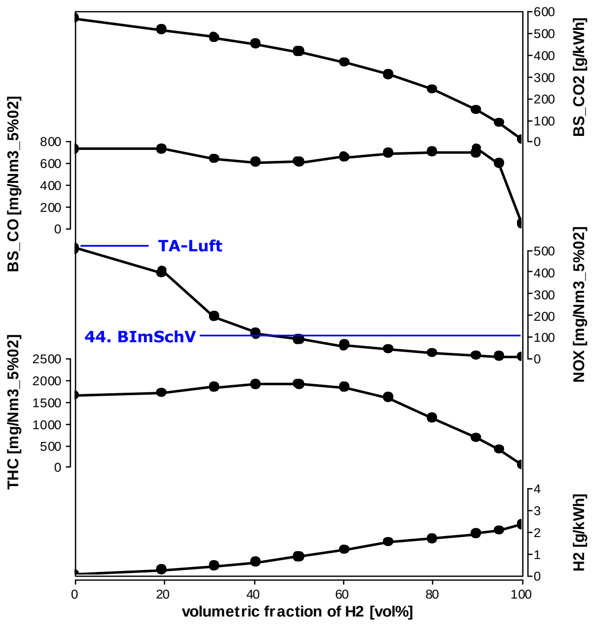


Figure 5. Gas Engine Emissions when increasing hydrogen fraction

As shown in the upper part of the diagram of Figure 6., the indicated engine efficiency increased by 2% points, when comparing operation with pure hydrogen and pure natural gas. The scale in the lower part of the diagram shows the measured low-pressure cycle efficiency (pumping losses). The scavenging loss deteriorates due to the reduction of exhaust enthalpy (decreasing of exhaust temperature). It is expected that pumping losses will improve in the future as newly developed turbochargers with higher efficiencies become available.

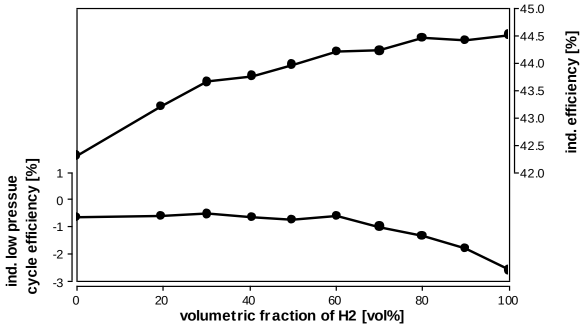


Figure 6. Efficiency and pumping loss development depending on hydrogen rate into natural gas

An example of how the combustion is affected with increased hydrogen rates up to 100% hydrogen is shown in Figure 7. All four blend ratios were measured at IMEP 11 bar. For stable engine operation on pure hydrogen, a significant increase of boost pressure (lambda) is required.

The combustion duration was shorter in comparison to pure natural gas operation due to the superior combustibility and reactivity of hydrogen even in extreme lean conditions.

The more compact heat release rates in Figure 7 go along with a reduction in NOx emissions, following the NOx trace from left to right in Figure 5. For the reduction of the combustion temperature and the reduction of nitrogen oxides formation in pure hydrogen operation the air excess ratio must be increased in comparison to natural gas. This higher lambda value can be easily realized with increased boost pressure on the single cylinder engine test bed. For multi cylinder applications, the available turbochargers, as well as the charging concept itself (single or two stage), can limit the achievable upper lambda levels, and subsequent power output. In addition, to avoid the high pressure rises within the combustion process (dp/dt [bar/°CA]) the control of lambda with increasing of the boost pressure must be considered.

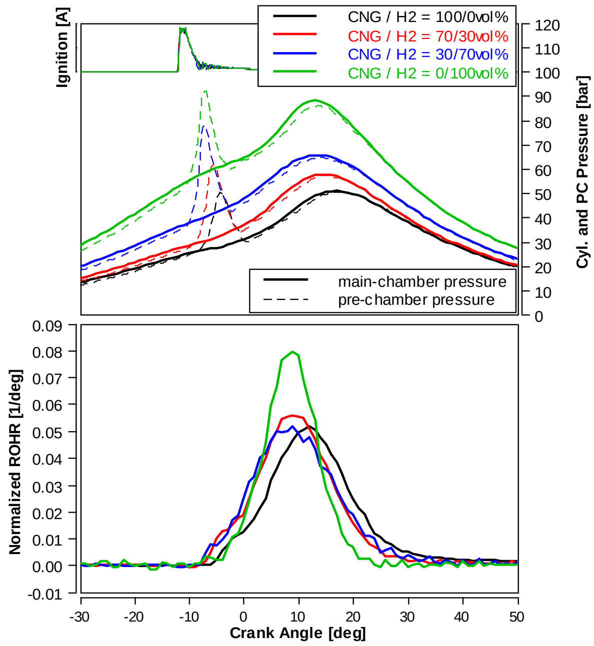


Figure 7. Gas quality variation at constant load (IMEP = 11 bar)

In Figures 8, 9 and 10 below, the measured exhaust gas emissions of carbon dioxide CO_2, hydrogen H_2 and formaldehyde CH_2O are shown. For CO_2 reduction, it can be seen there is a clear benefit of hydrogen to natural gas, as shown in Figure 8.

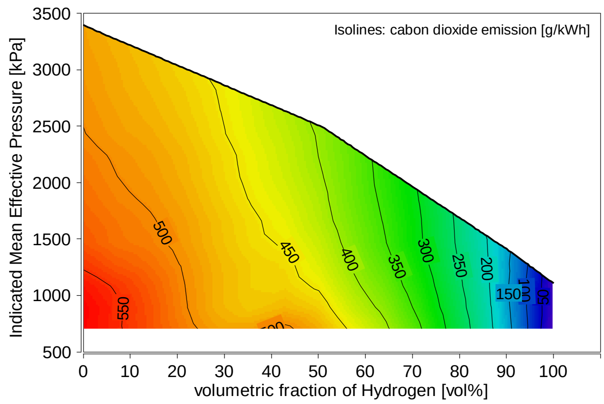


Figure 8. Engine map hydrogen – IMEP – carbon dioxide emissions

Due to the Otto cycle concept, unburned hydrogen emissions were measured in the exhaust gas. The maximum values in the range of 1600 to 1800ppm were measured at 90 to 100 vol% hydrogen, Figure 9.

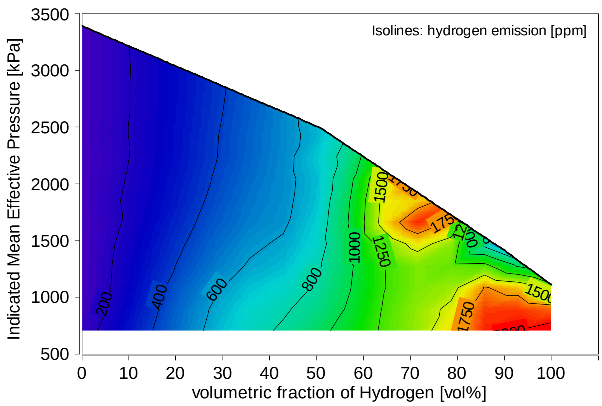


Figure 9. Engine map hydrogen – IMEP – hydrogen emissions

The formaldehyde emissions (CH_2O) in the engine map are shown in Figure 10. Significant reduction of the carcinogenic CH_2O can be achieved when small fraction of hydrogen is admixed to natural gas. In the range of 70 vol% to pure hydrogen, values below 60 mg/mn³ at 5% residual oxygen are achievable.

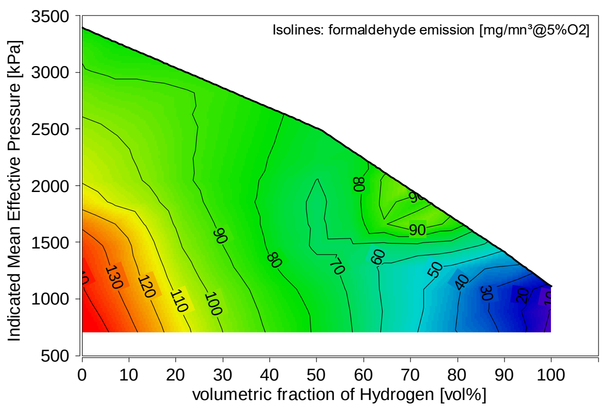


Figure 10. Engine map hydrogen – IMEP – formaldehyde emissions

4.1 Engine retrofits for natural gas / hydrogen operation

For retrofit solutions as well the discussed approach to mix hydrogen into the public natural gas grid, small engine maps of two different hydrogen rates to natural gas with MN65 were measured, as shown in Figure 11. Results indicated, that the operating window with 30vol% and 50vol% gets significantly smaller to the lambda range compared to pure natural gas engine operation. Due to rapid increasing of cylinder pressure gradients, an engine operation at 500 mg/mn³ @ 5% O_2 was not feasible with 30vol% and 50vol% of H_2 due to very high rates of cylinder pressure rise and with high combustion noise although, due to rich conditions, no knocking was measured. Though, the advantage with hydrogen is clearly visible for stable engine operation below NOx values below 100 mg/mn³ @ 5% O_2.

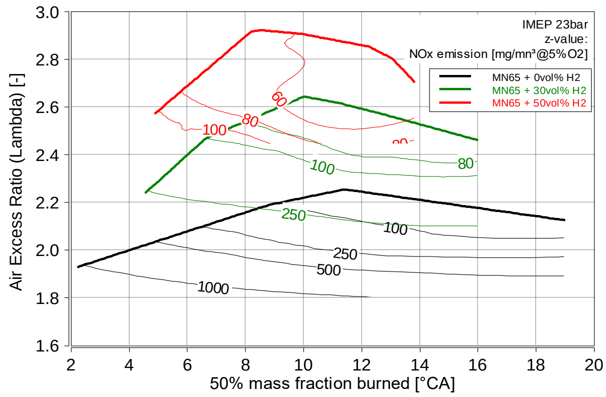


Figure 11. Operating range with 0/30/50vol% hydrogen admixing to natural gas at IMEP 23 bar

Figure 12 shows the measured pressure traces and heat release rates at different loads at constant mixing ratio of natural gas / hydrogen = 70/30vol%. Safe and stable engine operation higher than IMEP 25 bar is feasible. The NOx value of 100 mg/mn³ @ 5% O_2 was kept constant. An efficiency optimized 50% mass fraction burned point of appr. 8°CA aTDC is possible.

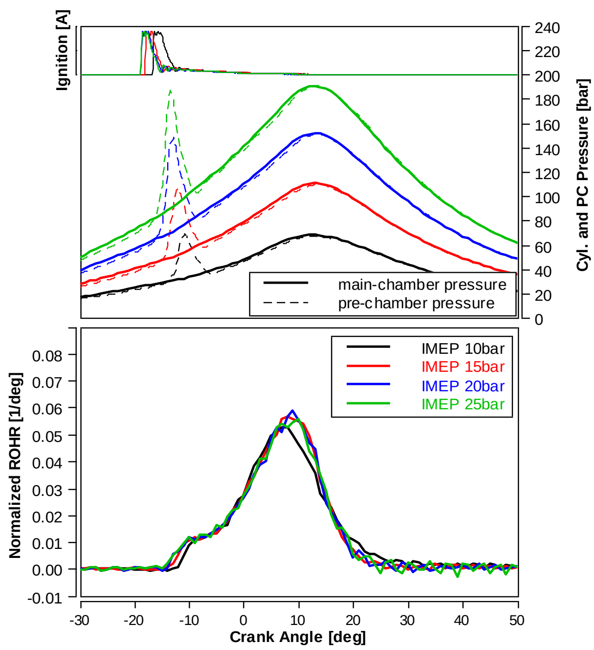


Figure 12. Load variation at natural gas / hydrogen 70 / 30 vol%

4.2 Operational engine challenges mixing natural gas with hydrogen

4.2.1 Combustion anomalies

To achieve higher power outputs in pure hydrogen operation, the different detected events with combustion anomalies must be handled in a safe manner.

Two detected issues related to combustion anomalies are presented in Figure 13 and Figure 14. Figure 13 shows a backfire event into the intake piping system when the intake valves were open (early self-ignition before IVC).

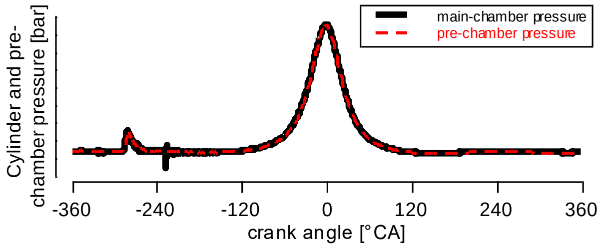


Figure 13. Combustion anomaly – early self-ignition before IVC (backfiring)

Possible reasons for backfiring are hot surface temperatures on e.g. spark plug or exhaust valve, hot residual gases, burning deposits or burning oil droplets.

Figure 14 shows an HCCI-like auto-ignition initiated combustion process. This means that the combustion in main and pre-chamber initiates and continues in parallel. The typical pre-chamber pressure peak was not measured. The combustion was not controllable with adaptation of ignition timing. Both combustion events in the main and pre-chamber follow same pressure rise characteristics. From the author's perspective, the reason for this phenomenon can be hot surfaces or burning/self-ignition of the lubrication oil.

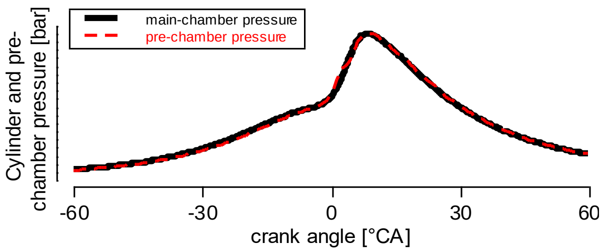


Figure 14. Combustion anomaly – HCCI like combustion

For the detection of combustion anomalies, the crank angle-based pressure measurement in the main- and pre-chamber as well in the intake and the exhaust port was considered. Combined with parallel and intelligent online evaluation on the test bed the different effects in anomalies were monitored in a safe manner.

4.2.2 Turbocharger Challenges

For the thermodynamic engine layout, the charging system becomes a challenge. For pure hydrogen engine operation, a significant increase of the air excess ratio (lambda) is required compared to pure natural gas operation. A high lambda is mandatory for low NOx emissions and for the limitation of the pressure rise in the cylinder. The requirement of high air mass flow was realized with increased boost pressure. In contrast, the exhaust gas temperature drops very quickly at higher volumetric fractions of hydrogen. Based on simulation results and experience it is evident, that a large compressor and a small turbine must be laid out. Figure 15 shows the increase of the boost pressure and the decrease in exhaust gas temperatures when increasing the volumetric fraction of hydrogen.

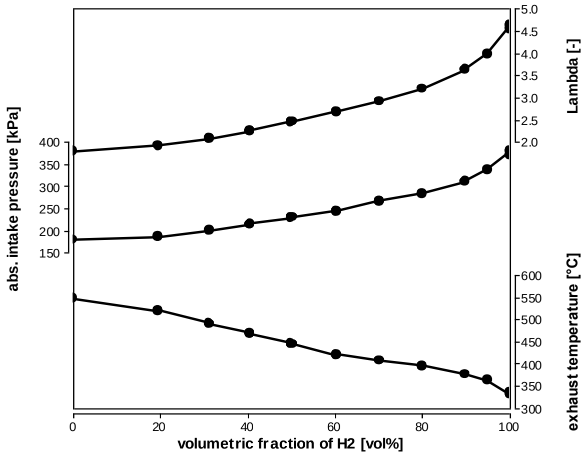


Figure 15. Lambda, intake pressure and exhaust temperature development depending on hydrogen rate into natural gas (constant load)

In case of retrofitting, the existing size of the turbocharger must be checked carefully to see whether adaptation with nozzle rings is possible or not.

Finally, in Figure 16, the measured absolute boost pressure in the engine map is shown.

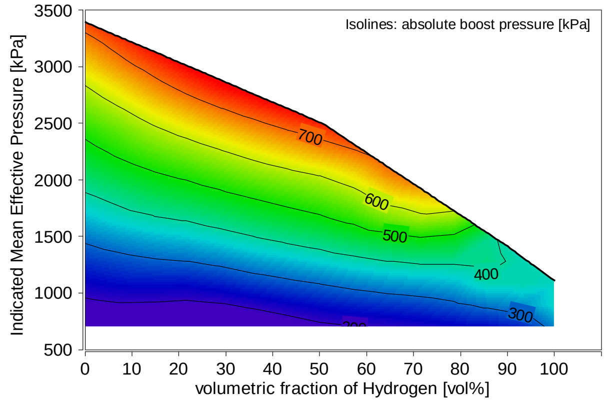


Figure 16. Engine map hydrogen – IMEP – absolute boost pressure

For moderate hydrogen admixing and when targeting high power output, two-stage turbocharging must be considered. In pure hydrogen engine operation, a single stage turbocharging is feasible but with the challenge of a combination of a large compressor together with a small turbine.

5 TEST RESULTS WITH AMMONIA-DIESEL ENGINE OPERATION

5.1 Initial Tests with Ammonia

Chapter number 5 deals with initial engine tests with different ratios of ammonia and Diesel. The combustion development and hardware as well the calibration optimization are still ongoing at the time of paper submittal.

The ammonia diesel investigations were performed on a single cylinder engine with 250 mm bore diameter. The diesel injector was positioned in a typical central location, the NH_3 supply was done with a venturi mixer unit to intake air. Investigations with a port gas admission valve close to the intake ports were not feasible due to limited hardware availability and space for installation of a port gas admission valve. The engine configuration is summarized in Table 2 below. The installation setting was comparable to Figure 4.

Table 2. Engine data – operated in ammonia substitution concept with Diesel ignition

Parameter (unit)	Value or remark
Bore (mm)	250
Stroke (mm)	320
Rated speed (rpm)	750
Rated power (kW)	200
Compression ratio (-)	17.0
Valve timings (-)	Miller 520

Figure 17 depicts the variation of the Diesel energy ratio. A Diesel fuel energy ratio of 1 represents pure Diesel operation. Engine load, engine speed, as well the air excess ratio were kept constant at a BMEP of 20.3 bar and 750 rpm. The main benefit of Diesel substitution with ammonia was clearly demonstrated in the significant level of CO_2 reduction compared to Diesel combustion. Limitations of the investigated combustion system, however, were identified as decreased combustion stability (measured by covariance of IMEP), which leads to a loss of combustion efficiency due to increased unburned fuel in the exhaust gas.

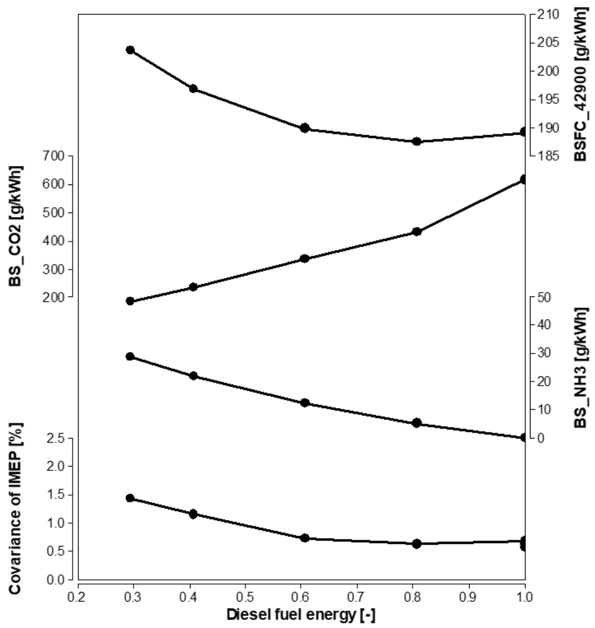


Figure 17. Diesel energy variation (BMEP 20.3 bar)

Reducing unburned ammonia (NH_3) in the exhaust gas in the entire engine map is one of the biggest challenges in ammonia combustion with diesel ignition. Figure 18 shows brake-specific engine-out NH_3 emissions at BMEP 20.3 bar. The excess air ratio (lambda) has a significant impact on specific ammonia emissions. In lean engine operations with a ratio greater than 2.0, a high fraction of unburned fuel was measured due to increased combustion instability. Approximately 12% of the supplied ammonia fuel energy was found to remain unburned in the exhaust gas flow. Decreasing the excess air ratio significantly improves the situation. The impact of injection timing and the 50% mass fraction burned point is relatively minor compared to the dominant impact of the excess air ratio.

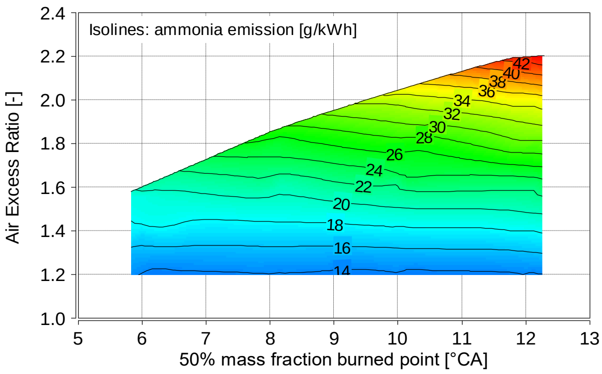


Figure 18. Specific ammonia emission in engine map at constant load (BMEP 20.3 bar) and speed (750 rpm)

Similar trend can be seen on N_2O (laughing gas) emissions over the entire engine map. The minimization of N_2O will be a key development target for the ongoing development steps of ammonia combustion engines. This is of utmost importance since the global warming potential (GWP) of nitrous oxide (N_2O) is about 265 times that of carbon dioxide (CO_2) over a 100-year time horizon [16]. A minimization of N_2O emissions can be achieved by means of engine calibration or consideration of adequate exhaust aftertreatment system. The values referenced are engine-out position and were measured by means of an AVL FTIR Sesam II system.

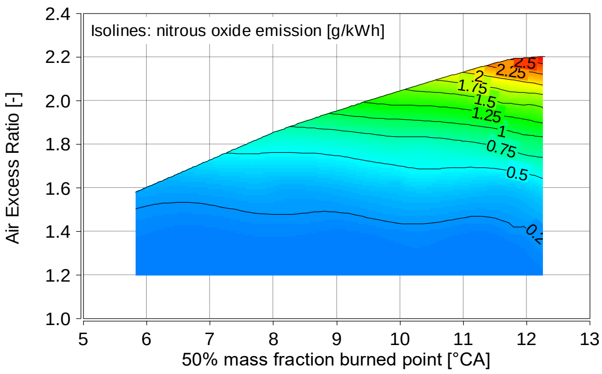


Figure 19. Specific nitrous oxide emission (N_2O) in engine map at constant load (BMEP 20.3 bar) and speed (750 rpm)

However, the engine operation in rich conditions is limited due to measured carbon monoxide emissions in the exhaust gas as well as the measured exhaust gas temperature. The target should be to avoid an additional CO oxidation catalyst, meaning that only a SCR system with ammonia slip catalyst is required. In Figure 20 the trend of the increasing of CO is plotted depending on the excess air ratio (lambda). In the described setting, the limit where the CO emissions are acceptable is approximately 1.4. Below that air excess ratio, the CO emissions increased rapidly.

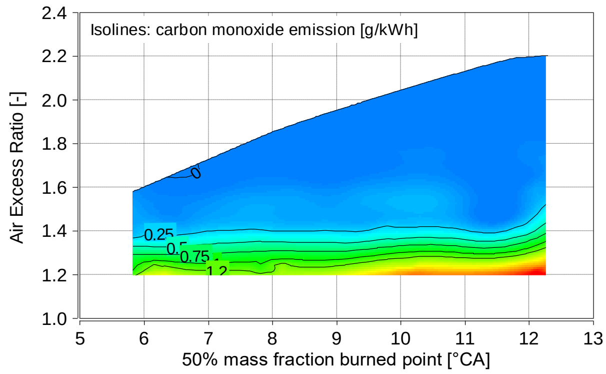


Figure 20. Specific carbon monoxide emission in engine map at constant load (BMEP 20.3 bar) and speed (750 rpm)

On the other hand, the design and type of turbocharger are influenced by the calibrated air-to-fuel ratio and the combustion phasing. For medium-speed engines, temperatures at an air-to-fuel ratio of 1.4 are considered acceptable. However, if the air-to-fuel ratio falls below 1.4, the temperatures can reach the limit of the turbine inlet temperature.

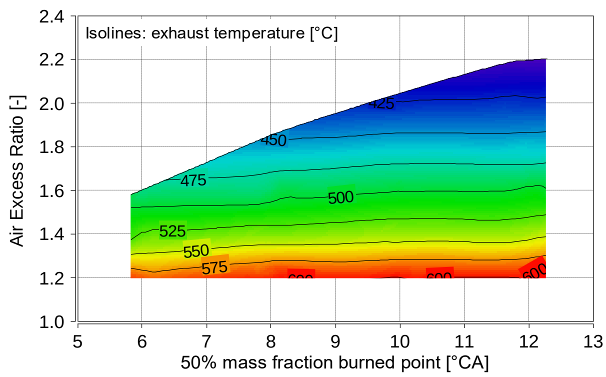


Figure 21. Exhaust temperature in engine map at constant load (BMEP 20.3 bar) and speed (750 rpm)

From an efficiency perspective, the highest values in a single cylinder engine were recorded in the range of air-to-fuel ratios between 1.4 and 1.6 and with advanced combustion timing, with 50% of the mass being burned at 6 to 8°CA after top dead center (aTDC). However, in lean operation and with retarded timing, efficiency decreases by roughly

2% or more due to the substantial increase of unburned ammonia in the exhaust gas.

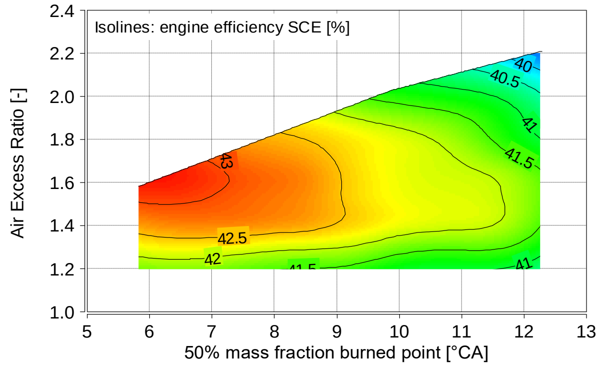


Figure 22. Engine efficiency in engine map at constant load (BMEP 20.3 bar) and speed (750 rpm)

For an accurate comparison of alternative fuels to diesel or gas the overall GHG emissions from combustion have to be assessed as CO_2 equivalent. For the natural gas engines this means to also include the methane slip (CH_4). As analogy, for the ammonia engine, the N_2O emissions have to be considered for the global warming factor considerations, assuming that unburned NH_3 is fully oxidized in the catalyst. CO_2 therefore is considered the baseline with a global warming factor of 1, whereas for N_2O a global warming factor of 265 taken into considerations [16]. This assessment follows the following equation:

$$CO_{2,equ.} = CO_2\ [g/kWh] + N_2O\ [g/kWh] \cdot 265 \quad (1)$$

The achievable CO_2 reduction goes hand-in-hand with the energy fraction of ammonia substituting diesel, although the engine efficiency deteriorates due to unburned NH_3 and lower combustion speed. Figure 23 shows the CO_2 reduction up to 75% (upper right corner), as a function of the air excess ratio and MFB 50% at BMEP 20.3 bar, 750 rpm and an ammonia energy fraction of 70%.

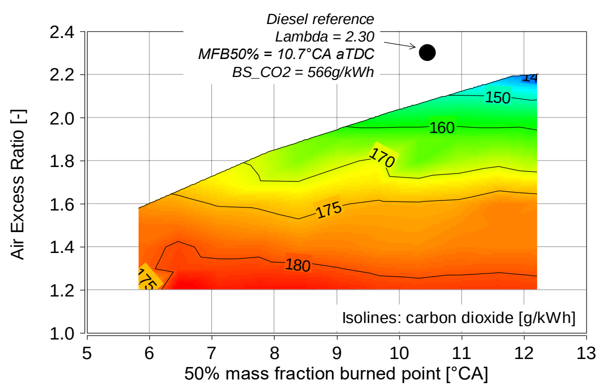


Figure 23. Specific carbon dioxide emissions in engine map at constant load (BMEP 20.3 bar) and speed (750 rpm)

Figure 24 below presents the CO_2 equivalent emission value, including CO_2 and N_2O. The results of a pure diesel measurement are plotted as a reference, with a CO_2 equivalent value of 566 g/kWh.

If the CO_2 equivalent as per equation (1) is considered, the upper right corner is even worse compared to the Diesel reference. The reason is the over-proportionally high emissions of N_2O in the lean burn regime (air excess ratio >1.9).

It can be concluded that a reduction of lambda and an advance of the combustion phasing is beneficial to reduce N_2O.

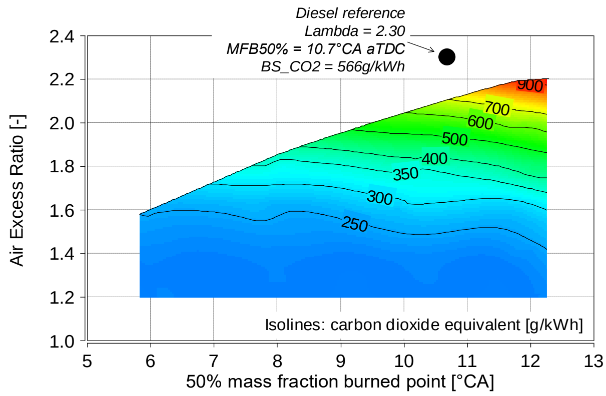


Figure 24: Specific carbon dioxide equivalent emissions (CO_2 + N_2O) in engine map at constant load (BMEP 20.3 bar) and speed (750 rpm)

The heat release evaluation with different air excess ratios with Diesel-ammonia ratio of 30/70 energy percent are shown in Figure 25. The Diesel injection pressure was kept constant at 160 MPa. It is obvious, that the first phase is dominated by the Diesel combustion with the characteristic pre-mix peak. Afterwards, the remaining ammonia is burned. A long burn out phase was measured in lean operation and presents, as mentioned, the highest values of unburned fuel in the exhaust gas. Rich engine conditions can shorten the combustion duration and reduce the engine-out emissions.

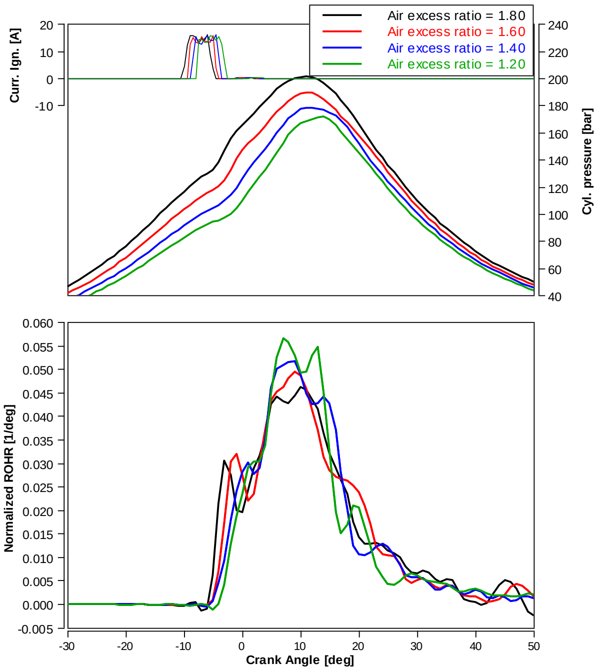


Figure 25. Pressure curves and rate of heat releases with different air excess ratios at BMEP 20.3 bar

5.2 Evaluation of Ammonia combustion concepts

Initial engines test showed that the specific focus has to be set on ammonia and N_2O slip. First priority will be to reduce unburned ammonia and N_2O production with engine internal measures. Compared to other test performed by AVL in parallel studies it was found, that H_2 enrichment can massively improve N_2O engine out emissions. As described in [10], a spark ignited gas engine concept shows significantly lower N_2O emission compared to the ammonia substitution concept described above. Whereas both combustion concepts were investigated by first screening tests on different engine sizes, a thorough combustion development is necessary to fully evaluate their full potential.

Table 3 summarizes the discussed future ammonia combustion concepts. If the systems are classified on the basis of the overall applicability of the generic engine conversion concept, the ranking of the criteria is as follows:

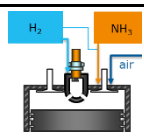
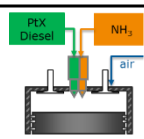
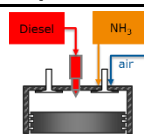
- Easy retrofit
- Net Carbon reduction approach
- Zero Carbon approach

On a more detailed level the ranking is also taking following criteria into account:

- CO_2 reduction potential
- GHG reduction potential
- NOx Compliance engine-out (IMO II)

It has been demonstrated that hydrogen enrichment is mandatory in certain applications, such as in the spark ignition of a hydrogen-fed prechamber to generate strong flame torches. In other cases, hydrogen can be used optionally for the purpose of enhancing flames.

Table 3. Ammonia combustion concepts

	NH_3 Gas Engine	NH_3 HPDI Engine	NH_3 Substitution Engine
			
Generic engine conversion concept	Easy retrofit	Net Carbon reduction approach	Zero Carbon approach
NH_3 mixture formation	Port gas admission	HPDI	Port gas admission
Ignition	Pre-chamber spark ignition	DI Diesel	DI Diesel
Ignition timing	variable	variable	variable
H_2 enrichment	up to 15vol% needed	beneficial	beneficial
CO_2 reduction	+ + +	+ +	+
GHG reduction	+ + +	+ +	+
IMO T2 NOx compliance	-	- -	-
Conversion complexity	- -	- - -	0

As a result of a first Ammonia testing campaign hydrogen enrichment was found to be a beneficial measure for both natural gas and ammonia engines. It supports flame enhancement and burnout, reducing the amount of unburned fuel and its associated byproducts. A promising optimization task will be to find the right balance of combustion calibration versus requirements for exhaust aftertreatment system. This can be reached as example with rich combustion conditions or additional hydrogen supply into the main chamber. These results are described and summarized in paper 667 [10].

5.3 Emission abatement

An inherent disadvantage of highly premixed combustion is the presence of unburned fuel, which is caused by flame quenching and fuel remaining in the crevices in the combustion chamber, leading to undesired combustion byproducts. This phenomenon of "methane slip" is well known in natural gas engines, where unburned fuel in the form of methane is emitted. The most undesirable combustion byproduct of this process is formaldehyde (CH_2O).

Considering analogy for the ammonia engine, the unburned fuel is NH_3 and the most undesired combustion byproduct is nitrous oxide (N_2O). In both cases leaner burn operation increases the amount of unburned fuel and the amount of combustion by-products furthermore.

In the context of exhaust gas aftertreatment, N_2O decomposition requires more than 500..600°C [17], [18], while the CH_2O conversion to CO_2 and H_2O requires a significantly lower exhaust gas temperature of about 350°C [19].

Figure 26 reports exemplary the N_2O conversion with a ruthenium (Ru) catalyst doped with different oxide materials.

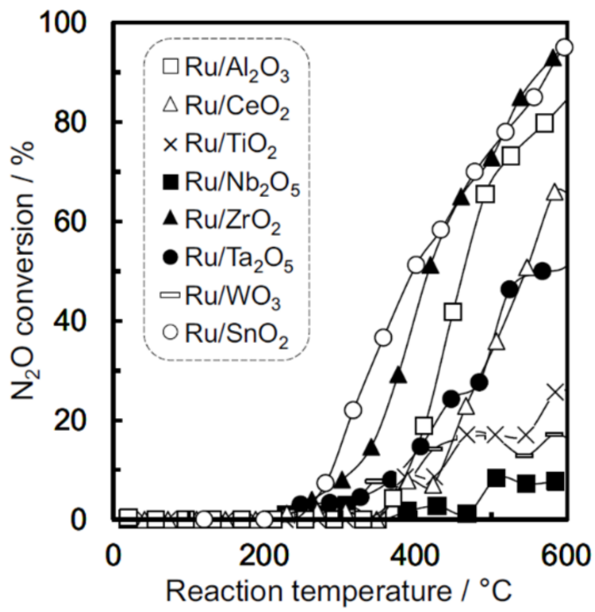


Figure 26. N_2O conversion depending on coating and temperature [17]

Furthermore, attention must be paid to the conversion of residual ammonia exhaust gas emissions. To achieve ammonia conversion efficiency close to 100%, the temperature upstream of the exhaust aftertreatment system must be higher than 250°C. Figure 27 shows an example from a single-cylinder engine test bed where close to zero ammonia concentration was measured downstream of the EAS. The inlet temperature of the EAS was consistently above 250°C. The graph is based on BMEP 20.3bar and the lowest feasible diesel energy ratio was approximately 25% (meaning 75% of the fuel energy was converted to gaseous ammonia).

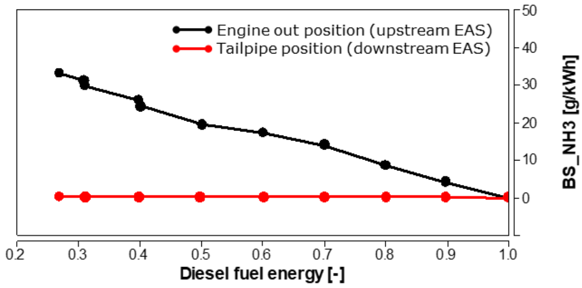


Figure 27. Ammonia conversion in aftertreatment system

Given the boundary conditions of high ammonia emissions during testing as described before, the installation of an exhaust aftertreatment system for the development of ammonia combustion in both single and multi-cylinder engine test beds, is essential. Attention must be paid to avoid ammonia and N_2O emission not only in a later engine application but also during engine development and testing.

Based on the findings described in this paper an important part of future R&D will focus on the characterization and development of the emission aftertreatment systems for minimization NH_3 and N_2O in tailpipe position.

Currently in AVL the ammonia projects are still ongoing, results will be presented in the future.

6 CONCLUSIONS

The paper summarizes the potential of ammonia and hydrogen fueled large internal combustion engines, supporting the required reduction of greenhouse gas of High Power Systems until 2050.

Pure hydrogen operation on a large medium speed gas engine requires a reduction of engine power due to increased numbers of cycles with combustion anomalies.

It was demonstrated that the power of the base gas engine can be kept and a hydrogen tolerance of 50vol% can be achieved.

As such this would be a short-term solution for already installed engines or retrofit projects, as the technical modifications on the engines are relatively small. Main focus for future developments is the increase of the power output in pure hydrogen operation requiring more intensive measures such as improved “hot spot cooling” and reduced lube oil consumption.

The Diesel – ammonia – combustion is successfully demonstrated on a medium speed single cylinder engine. The order of magnitude for a reduction of carbon dioxide is analyzed considering the CO_2 equivalent of N_2O. These first screening tests prove, that further optimization and development is needed to reduce the ammonia slip and N_2O emissions in the near future.

Future investigations with net-zero carbon fuels such as hydrogen, ammonia or methanol are planned on a high and a medium speed single cylinder engine. The aim is to refine engine performance and to define the required exhaust aftertreatment systems by characterization of substrate samples exposed to real exhaust gas.

Different mixture formation concepts, covering the low pressure port gas admission or Methanol injection, the low pressure direct injection into the compression stroke and high-pressure direct injection close to top dead center, are planned depending on the technical maturity and availability of injection - and gas admission components. Simulation work will support all experimental investigations on single and multi-cylinder engine tests. The results of the ongoing ammonia CFD simulations will be presented in 4th quarter of 2023.

7 ABBREVIATIONS

aTDC: after top dead center

AVL: AVL List GmbH

BMEP: Brake mean effective pressure

CFD: Computational fluid dynamics

CR: Compression ratio

DNV: Det Norske Veritas

EAS: Exhaust aftertreatment system

FTIR: Fourier-transform infrared spectroscopy

GHG: Greenhouse gas

HCCI: Homogeneous Charge Compression Ignition

HPDI: High Pressure Direct Injection

IMEP: Indicated mean effective pressure

IMO: International Maritime Organization

IVC: Intake valve closing

MFB 50%: mass fraction burned point 50%

MN: Methane number

O_2: Oxygen

ROHR: Rate of heat release

R&D: Research and development

U.S. EPA: U.S. Environmental Protection Agency

44. BImSchV: 44th Federal Immission Control Ordinance

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9 ACKNOWLEDGMENTS

For the ammonia test program, the authors would like to acknowledge the financial support of the "COMET - Competence Centers for Excellent Technologies" Programme of the Austrian Federal Ministry for Climate Action, Environment, Energy, Mobility, Innovation and Technology (BMK) and the Federal Ministry for Digital and Economic Affairs (BMDW) and the Provinces of Styria, Tyrol and Vienna for the COMET Centre (K1) LEC EvoLET. The COMET Programme is managed by the Austrian Research Promotion Agency (FFG).

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