Keywords: Ammonia, hydrogen, Dual Fuel combustion engine

Combustion concept for ammonia-fuelled cracker-engine-unit as propulsion system for inland waterway vessels

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Abstract

The CAMPFIRE partner alliance is working intensively on the use of regeneratively produced ammonia as a maritime fuel and energy storage. In addition to topics relating to ammonia bunkering, refueling and storage for marine applications, one project proposal is the development of a pilot propulsion system for inland waterway vessels that operates exclusively on ammonia. The project consortium would like to present the potential but also the challenges of using ammonia in shipping based on the development of an innovative cracker-engine-unit with focus on the energy conversion of the internal combustion engine (ICE).

The paper and presentation will give an overview of the systematic combustion analysis on a SCE. The basis is a combustion concept with ammonia as the main fuel and hydrogen, cracked from Ammonia, as a pilot fuel. As a part of the experimental tests various fuel injection concepts (DI and PFI) for both fuels were investigated. For carrying out the tests the development of a new fuel injector concept was driven ahead. The experimental tests on the engines were supported by optical investigation on mixture formation with liquid ammonia as well 0D/1D and 3D simulation of the combustion chamber and SCE.

The various fuel injection configurations are evaluated for performance and emissions with the focus on maritime application. Resulting from the evaluation of the experimental data a preferred combustion concept was deduced for the planed multi-cylinder engine test at the Campfire Open Innovation Lab (COIL) near to Rostock.

1. Introduction

The goal of the CAMPFIRE1 consortium is to research and develop new energy conversion and storage technologies based on green ammonia for the energy system of the future. A long-term objective is for small and medium-sized enterprises (SMEs) in the region of Northeast Germany to establish pathways for the utilisation of advanced technologies that can be exported to other areas. CAMPFIRE technologies produced in the Northeast will facilitate the development of an energy economy based on green ammonia and a carbon-free, secure energy supply in the Baltic Sea Region, in Europe, and abroad. In the long term, the consortium aims to develop effective and cost-efficient methods to reduce the global carbon dioxide levels in the Earth's atmosphere.

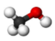
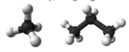
Maritime transport is an essential part of the global trading systems. Although it is the most efficient and climate friendliest transport method, worldwide shipping accounts for 3% of global CO_2 emissions. This demands a respective emissions reduction in parallel to all other polluters. For that reason, the IMO (International Maritime Organisation) has committed to achieve full climate neutrality for sea-going shipping until 2050. [1]

Due to the required operation ranges and limited volumetric and gravimetric storage capacity on board of vessels, pure electrification solution with batteries as energy storage are out of discussion for most shipping applications. As solution the use of alternative fuels based on renewable energies has highest priority. In this respect Ammonia is very interesting due to its possibility as Hydrogen carrier and CO_2-free local combustion.

The typical characteristic values of Ammonia compared to other fuels types are displayed in the following Table 1.

[1] https://wir-campfire.de/

Table 1: Fuel characteristics values

		Hydrogen	Methanol	Ammonia	eFuels (e-Diesel Fuel)	MDO	LNG
Molecular Formula		H_2	CH_3OH 	NH_3 	C_nH_{2n} n = 8 ... 20	C_nH_{2n} n = 8 ... 20	CH_4, C_3H_8 
State of Aggregate @ 0°C, 1,013 bar	-	gaseous	liquid	gaseous	liquid	liquid	gaseous
Density @0°C, 1,013 bar	kg/m³	0,09 @ 15°C, 350 bar: 24 @ 15°C 700 bar: 40 Liquified -253°C: 71	792	0,73	approx. 890 (Diesel Fuel)	890 ... 900	0,72 Liquified -161 °C: 431 ... 464
Storage Conditions		Liquified: -245 ... -250 °C @13 bar Compressed: 350 bar / 700 bar @ ambient temp.	Ambient	Liquified: -34°C @ athm. pressure Gaseous: 10 ... 30 bar @ ambient temp.	ambient	ambient	Liquified: -161 ... -164 °C @ approx. athm. pressure
Boiling Point	°C	-253	64	-33	200	163 ... 399	-161
Lower Heating Value (gravimetric)	KWh/kg MJ/kg	33,3 120	5,5 20	5,2 19	11,9 43	11,8 approx. 42	13,9 50
Min. Ignition Energy	mJ	0,016	0,1	14	0,23	0,23	0,25 ... 0,28

In addition, the required gravimetric and volumetric storage density needs to be considered (see Figure 1). The figure shows that not only the fuel specifics are essential, but also the required containment technology.

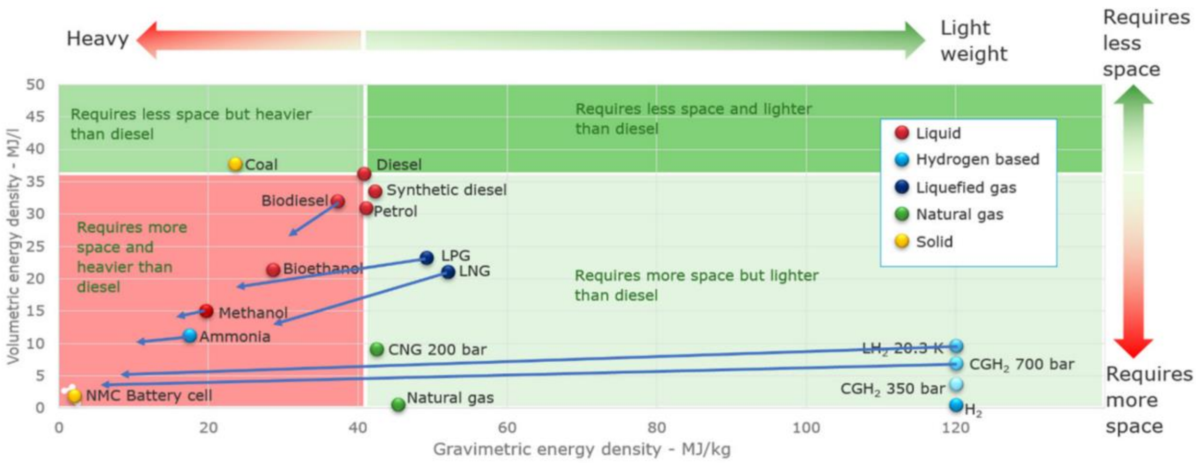


Figure 1: Energy densities for different energy carriers. The arrows represent the impact on density when considering the storage systems for different types of fuel (indicated values only) [2]

With this information plus the application type and operating profile as well as the intended operation range the configuration of a new built vessel fuelled by Ammonia can be executed. In case of a ship's retrofit to Ammonia operation the resulting restrictions can be defined accordingly.

The lower heating value of Ammonia is significantly reduced, compared to traditional fuels. But in comparison to Hydrogen, it offers the advantage that, depending on temperature, Ammonia is already present in liquid state at moderate pressures and temperatures (8,6 bar @ 20 °C). This results in a higher volumetric energy density compared to highly compressed or liquid Hydrogen. [3]

Other challenges with the use of Ammonia are related to material compatibility and interaction with lubrication oil, leading to further required investigation works. The safe handling of Ammonia during bunkering, storage, and usage ashore and on board is another evident topic which needs high attention. This includes crew training concepts as well. [3]

Currently various developments for Ammonia-fuelled engines are ongoing. The first movers can be found in the slow-speed two-stroke segment as main propulsion for large seagoing vessels, followed by respective works on medium-speed four-stroke engines. In all that cases dual-fuel combustion concepts are utilized where the Ammonia is ignited by a Diesel pilot fuel spray and the Diesel operation mode is available as back-up option. The first and very few of these engines are expected to start field operation in 2025.

For high-speed four-stroke engines several developments are ongoing, e. g. for power generation purposes or inland waterway shipping within the CAMPFIRE alliance, but all far from end customer usage. Within these projects both dual-fuel and spark-ignited combustion concepts are under investigation.

Interestingly, recently a consortium in Japan and China claimed to finalize development of a car engine with Ammonia as fuel. [4]

The involvement in Ammonia-fuelled engine development showed, that a concentration on the engine development only is not sufficient. Due to the novelty the complete auxiliary systems around the engines and including the tank and fuelling infrastructure need engineering as this is not available as commodity. In this respect also, safety regulations must be developed and the respective processes need approval by the responsible authorities.

2. Application and Development Process

In this CAMPFIRE research and development project the focus is on inland waterway transport applications. An ammonia tanker (Figure 2) that mainly travels on the Rhine between Ludwigshafen and Rotterdam has been selected as the test vehicle for the field testing of the NH_3 cracker combustion engine.

Currently the ship is powered by an old diesel engine, which is now to be replaced by a hybrid drive consisting of the NH_3 engine, cracker, generator and electric motor connected to the propeller shaft (see Figure 2).

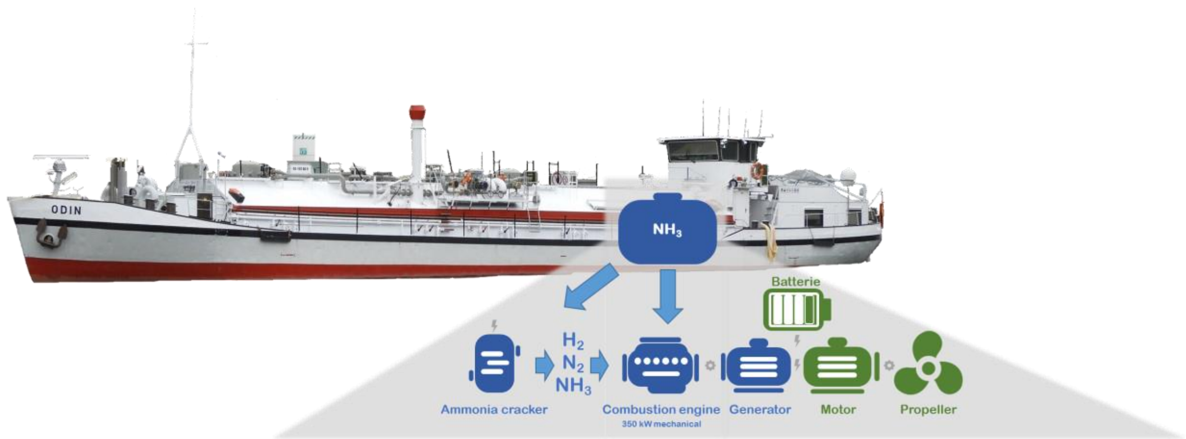


Figure 2: The hybrid drive, based on NH_3-engine, Cracker, generator, battery, e-motor and propeller that will be developed to integrate in inland water vessel MS ODIN.

The hybrid drive was chosen to enable redundancy in the drive and to obtain more flexibility of the drive. Legal safety requirements for an ammonia engine are not yet available. Nevertheless, measures have been defined in coordination with DNV to ensure the safety of the drive. This includes a separate room for the fuel tank and the battery unit and a ventilation concept for the entire area.

Currently, the operation and load profile of the ship is being recorded over a longer period of time using additional measuring technology. The recorded operating data is then used to examine and determine the efficient use of the hybrid drive. The characteristics of the combustion engine and the cracker must be considered, as well as features of the route (including the number of locks, water flow). The optimization of the hybrid drive should therefore contribute to a further increase in efficiency and a reduction in emissions.

However, before the engine is used in the ship, a number of development steps must be considered. A number of partners are involved in this CAMPFIRE research project, who contribute their specialist knowledge to the various tasks and enable the project to progress. In order to be able to evaluate different combustion concepts, determine their potential and test the necessary hardware, a methodology was defined that focuses on tests on the single-cylinder engine, which are accompanied by thermodynamic and CFD simulations (Figure 3).

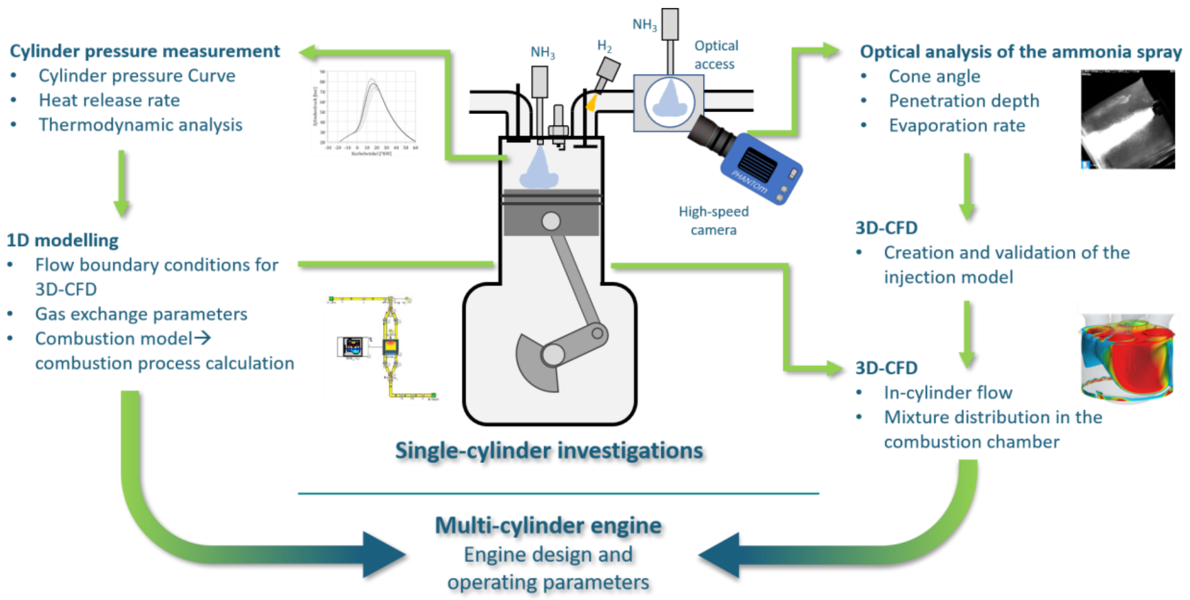


Figure 3: Development methodology for the combustion process

The single-cylinder engine (SCE), which runs on the test bench at KIT in Karlsruhe, was equipped with extensive measuring equipment. Since no knowledge of NH_3 injection was available, one of the first investigations was to visually examine the injection jet geometry and the evaporation of NH_3 with injection tests into the intake manifold. The results were important for the design and optimization of the NH_3 injectors and also for the design of the combustion concept.

Based on the experimental tests on the SCE models for the simulation of ammonia mixture formation and following combustion process are in development. These models represent the basis for setting up a multi-cylinder engine model that is used to derive basic requirements for the design of the full engine with regard to the turbocharger, the cooling system and the parameterization of the engine control unit.

3. Combustion Process Development

3.1. Testbed and Single Cylinder Engine

To allow proper single-cylinder engine monitoring and data acquisition of a wide range of relevant information for engine testing, the experimental activities were carried out in a fully instrumented testbench, as depicted in the scheme in Figure 4. The test engine is a single-cylinder unit based on the Liebherr D966 diesel engine. The cylinder head was modified so that a sleeve with spark plug can be fitted in place of the diesel injector. An injector for direct ammonia injection can be fitted in an additional bore in a lateral position. A specially manufactured intake manifold can accommodate both an injection valve for hydrogen and the ammonia injector. The relevant engine data are listed in Table 2.

The engine was coupled to a dynamometer to control load and speed. The in-cylinder pressure was measured by a Kistler 6045B piezoelectric pressure transducer in conjunction with a Kistler Piezo charge amplifier and referenced by an encoder with a resolution of 0.1 crank-angle degrees. A Kistler absolute pressure sensor was used to measure the intake pressure, exhaust pressure and fuel pressure of ammonia. The ignition settings were controlled by an SEM Multispark Control Unit (Default System). All the sensors above were connected to a Dewetron-800-CA high-speed data acquisition unit, allowing real-time combustion analysis using the Dewesoft software. Quantitative information about the combustion process and its cycle-to-cycle variability was obtained through a heat-release analysis from 100 consecutive cycles.

The fuel mass flow rate was measured through a Endress+Hauser coriolis flowmeter. The air mass was measured by a RMA rotary piston meter. A Bosch wide-band lambda sensor LSU 4.9, conditioned by an ETAS Lambda Meter LA4, determined the exhaust oxygen concentration. This allows the calculation of the air-fuel ratio. The concentration of HC, CO, CO_2, O_2 and NO_x on the exhaust were measured by an AVL AMA 4000. The concentration of NH_3, N_2O and NO_x on the exhaust were measured by a IAG FTIR and the H_2 in the exhaust was measured by a MS4 HSense.

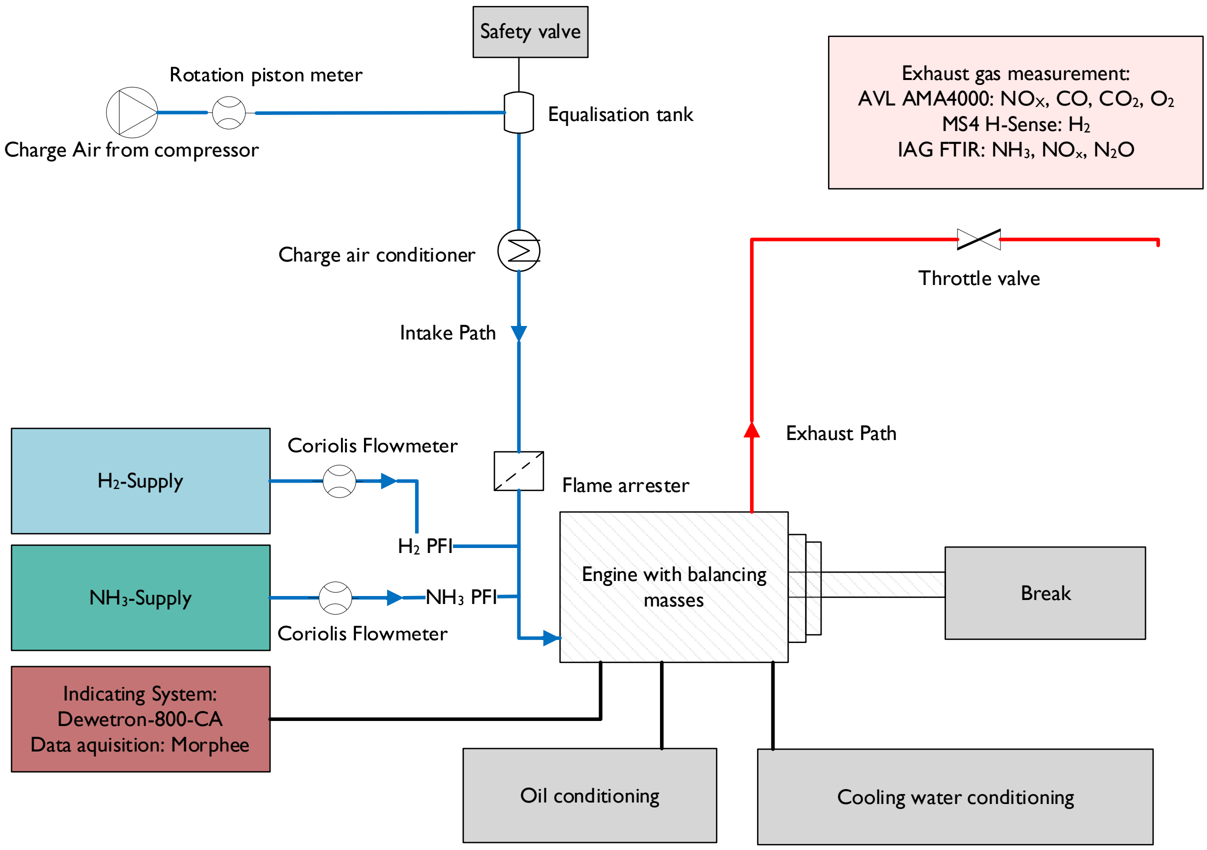


Figure 4: Scheme of the single-cylinder engine test bench

Table 2: Engine data

Cylinder	1
Stroke	157 mm
Bore	135 mm
Max. speed	1900 rpm
Operating principle	4-stroke SI
Displacement	2.24 l
Type	Modified Liebherr Diesel engine based on D966

3.2. Injection System Configuration

Ammonia and hydrogen were supplied from storage bottles. Hydrogen is supplied in gaseous form at a pressure of 300 bar at ambient temperature and can therefore be injected directly into the intake manifold via a pressure reducer using a hydrogen injector from Hörbiger. Most of the ammonia is in liquid form in the ammonia storage bottles. The pressure is a function of the ambient temperature via the vapour pressure in the bottles and is in the range of 2 to 8 bar. To achieve the desired injection pressure the ammonia must be compressed. For this purpose, a double-acting compressor station from Maximator was integrated, which can compress the ammonia up to 60 bar. The desired injection pressure is then set via a pressure reducer. The ammonia injector can be operated up to a pressure of 30 bar. In order to avoid pressure pulsations in the injection path, two Hydac membrane dampers and two calming volumes were integrated into the injection pipes. The basic structure of the ammonia supply system is shown in Figure 5.

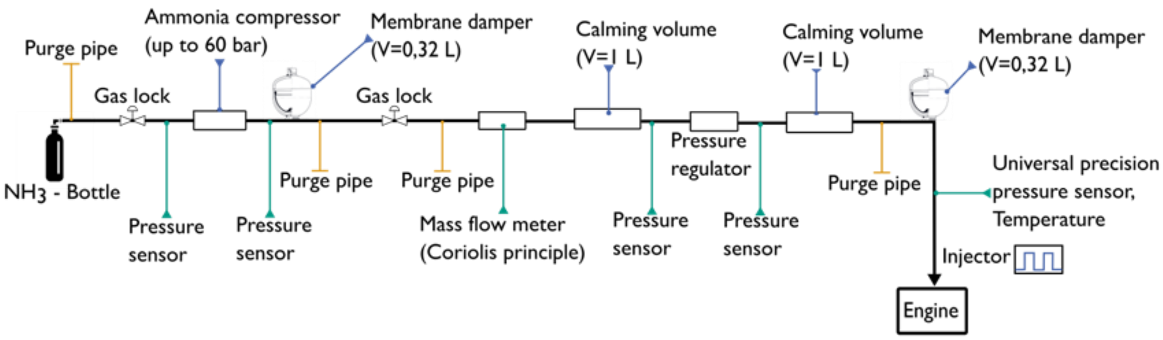


Figure 5: Ammonia infrastructure of the test bench

The DI injector from Liebherr can inject liquid and gaseous ammonia and gaseous hydrogen. The injection configurations shown in Figure 6 can be realised with the existing components. Hydrogen can be injected directly into the combustion chamber or into the intake manifold. In either case, the hydrogen is gaseous. Ammonia can be injected in liquid state into the intake manifold or directly into the combustion chamber by the Liebherr injector.

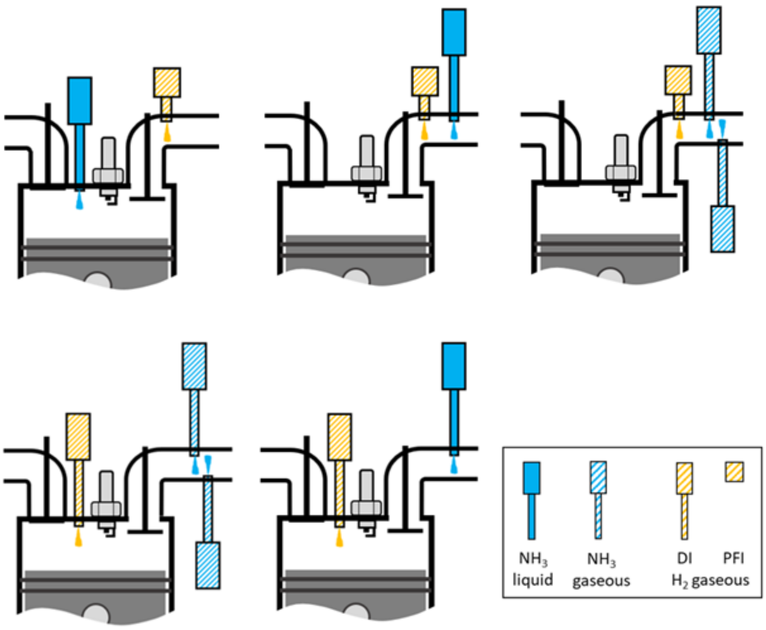


Figure 6: Injection configuration with J-Gap spark plug Combustion

The present study is divided in three sections: (I) the first one explores the behaviour of the combustion process and the emissions for different lambdas. The (II) second one shows a comparison between different compression ratios and the (III) last one shows the impact of a different ignition system.

(I) Lambda Variation

A lambda variation was carried out to determine the optimum operating conditions for the ammonia-hydrogen combustion. The combustion stability was limited to a COV of IMEP of 3%. The IMEP of this measurement was 17 bar at 88% energetic NH_3 share and 1500 rpm. Both fuels were injected in the intake path (PFI) of the single-cylinder engine. The objective during the operation was to find the lambda with the lowest COV of IMEP and the lowest emissions. Regarding a 100% hydrogen combustion a leaner lambda would be preferable because of lower NO_x emissions. In this combined NH_3-H_2-combustion process NO_x can be produced not only by the usual mechanisms such as Zeldovich mechanism but also from the ammonia fuel itself. This makes it important to know what influence lambda has on the generation of emissions during this combustion process.

Figure 7 shows the influence of the mixture composition on the combustion process. With a fuel air ratio of lambda higher than 1, the COV of IMEP increases. This means that the combustion stability decreases. This exacerbated cycle-to-cycle variability in reason of the mixture composition is caused by the deterioration of the flame development on its initial stages, leading to different combustion development, or in some cases, flame quenching. This is also the case when lambda is too rich. In terms of exhaust gas emissions, it is shown how the formation of nitrous oxide (NO_x) is greatly reduced as lambda is decreased, reaching the lowest emissions at lambda 0.9. With lambda smaller than lambda 1, the formation of NO_x decreases (logarithmic scale). The high vaporization enthalpy of ammonia leads to decreasing of the in-cylinder temperature and the lack of sufficient oxygen atoms below lambda 1 for the formation of NO_x lead to decreasing NO_x emissions. At the same time, hydrogen emissions increase, suggesting that ammonia decomposes into hydrogen in air-fuel-mixtures below lambda 1. Which is also shown by the decreasing NH_3 slip. The N_2O emissions decrease with decreasing lambda due to the lack of oxygen below lambda 1. In summary, it can be said that lambda has a major influence on emission formation and the combustion stability. The emissions can be controlled in a very targeted manner, also with regard to exhaust aftertreatment. Based on these results, lambda 1 operation was selected for further measurements. The trade-off between good combustion stability and low emissions is best here.

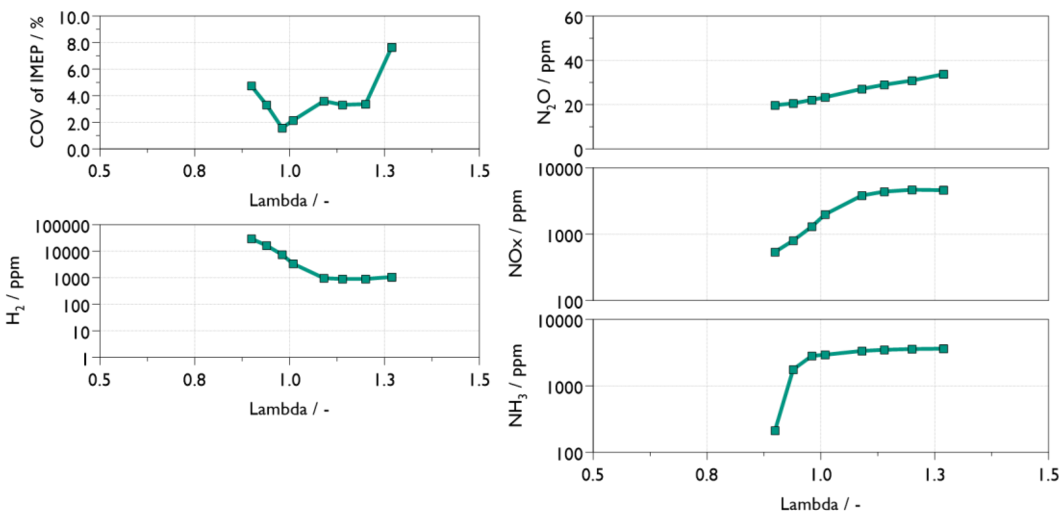


Figure 7: COV of IMEP and Emissions over Lambda at compression ratio 18.5 at 17 bar IMEP, 60 °C charge air temperature

(II) Compression ratio variation

The experiments were conducted by changing the compression ratio at lambda 1 in order to investigate the effect of a higher compression ratio on the maximum energetic NH_3 share. In this study knocking is characterized by a maximum knocking amplitude. The aim of the study was to get the maximum energetic NH_3 share.

In Figure 8 the impact of the COV of IMEP over the energetic NH_3 share for three different load points is shown. With increasing compression ratio, the energetic NH_3 share increases. The combustion process has two different limiting factors. The first limiting factor is the COV of IMEP it must not exceed 3 %. This criterion becomes critical if the ammonia content is too high. When the ammonia content of the mixture is too high it leads to an increase of the COV of IMEP resulting from a poor and late combustion. That means the combustion process gets unstable. The second limiting factor is the knocking criterion. This criterion is reached when too much hydrogen is injected. This behaviour is reinforced with increasing compression ratio.

For CR 14 to 18.5 the engine could be operated at all load points. The ignition timing was always set so that the mfb 50% is in the range 8±1° CA after TDC. A more precise adjustment of the mfb 50% during engine operation is not practicable, as the mfb 50% would be strongly influenced by slight deviations in the NH_3 share when readjusting the load and too many iterations would be necessary. Only at the operating point IMEP = 22 bar with a compression ratio of 18.5 did the mfb 50% have to be set to 12° CA after TDC in order to avoid exceeding the permissible peak pressure. With the highest CR of 22 it was not possible to reach fullload with 22 bar IMEP because the peak pressure was above the limit even with reasonable late adjustment of mfb 50%. The efficiency and energetic NH_3 share predominate for CR 17.

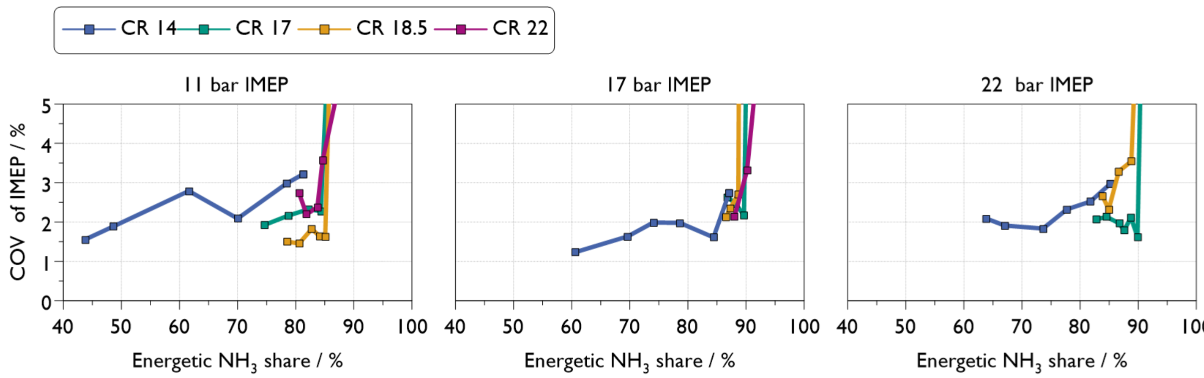


Figure 8: COV of IMEP over the energetic NH_3 share for different compression ratios at Lambda 1, 40 °C charge air temperature

Figure 8 shows cylinder pressure curves at IMEP of 17 bar for the variation of the compression ratios at maximum H_2 content, at maximum NH_3 content and in between at stable combustion behaviour with small COV of IMEP. Each curve is from a single combustion cycle. To illustrate the range of fluctuations, the dashed line shows the cycle with the lowest peak pressure and the solid line the cycle with the highest peak pressure from a measurement with 100 consecutive cycles.

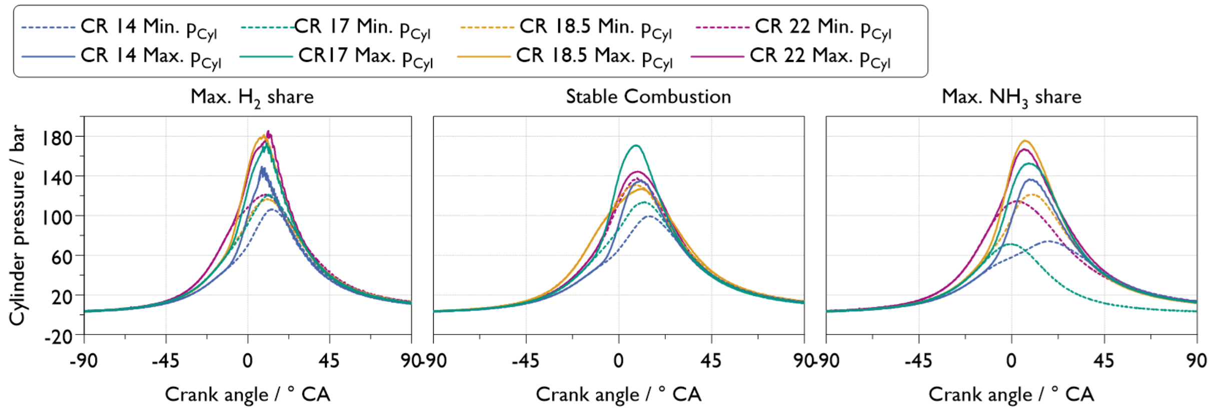


Figure 9: Cylinder pressure over crank angle for different compression ratios at 17 bar IMEP, Lambda 1, 40 °C charge air temperature

The left-hand diagram in Figure 9 shows that in all cases knocking combustion limits the maximum H_2 content. The permissible peak pressure of 220 bar is not reached. The maximum NH_3 content is limited by exceeding the COV of IMEP of 3%. In some cases, very late combustion phasing occurs, such as with CR = 14, or even misfiring, such as with CR = 17 and 22. This is related to the poor flammability and low laminar flame speed of ammonia. The "stable" range in between still has COV of IMEP values significantly greater than 1. For example, the fluctuation range of the peak pressure per cycle at CR = 17 is approx. 55 bar (corresponding to 32% of the maximum peak pressure). A further reduction in the COV of IMEP would be desirable for commercial applications.

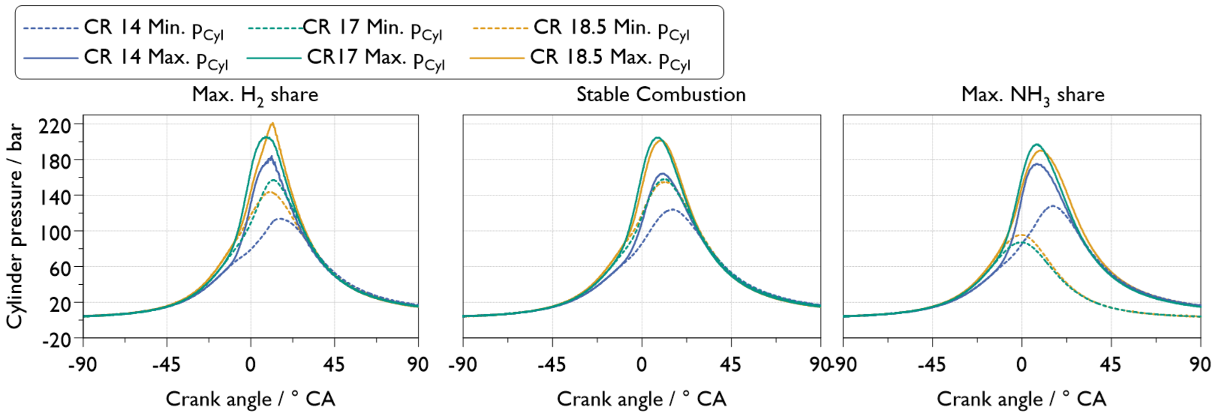


Figure 10: Cylinder pressure over Crank angle for different compression ratios at 22 bar IMEP, Lambda 1, 40 °C charge air temperature

Figure 10 shows the same as in Figure 9 but for the fullload conditions at 22 bar IMEP. There is no significant deviation in the behaviour in the stable range and at the limit of the maximum NH_3 content. With the latter, as with 17 bar IMEP, late combustion phasing and misfires occur, which prevent a further increase in the NH_3 content. At a compression ratio of CR = 14, the limiting factor is also the occurrence of knocking combustion, as in Figure 9. As Figure 11 shows, the maximum knocking amplitude reaches almost 5. At compression ratios 17 and 18.5, the permissible peak pressure of 220 bar is exceeded before knocking of a similar intensity to that at CR = 14 occurs, which is the limiting factor. It is not possible to operate the engine at the compression ratio CR = 22 at 22 bar IMEP due to peak pressures being exceeded in the entire NH_3 range.

The knocking in Figure 11 is calculated out of a high-pass filtered cylinder pressure signal. The software divides the high-pass filtered signal after combustion by the high-pass filtered signal before combustion. When there is no knocking the result of this equation is 1. When the combustion starts to knock the result of this equation increases. The limiting maximum knocking amplitude was set to 4. The limiting maximum knocking amplitude of 4 is not achieved for both CR 17 and CR 18.5. The peak pressure of the engine must not exceed 220 bar. The measurement has to be stopped because of the peak pressure for CR 17 und CR 18.5 which is shown in Figure 11 (right). CR 17 und CR 18 exceeded the peak pressure at around 82% energetic NH_3 share. The maximum knocking amplitude in this point is comparatively low, but it is still sufficient to exceed the peak pressure limit of the engine.

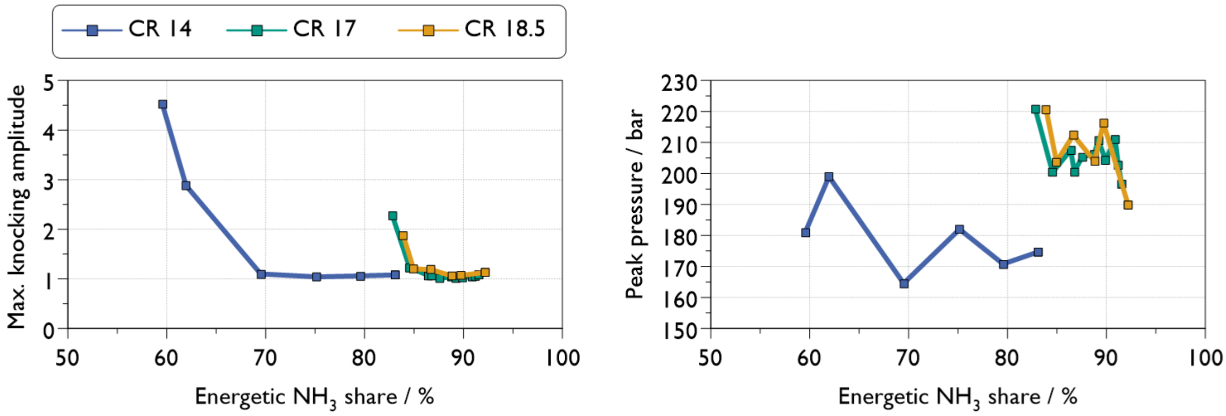


Figure 11: Left) Maximum knocking amplitude over the energetic NH_3 share at 22 bar IMEP, Lambda 1, 40 °C charge air temperature
right) Peak pressure over the energetic NH_3 share at 22 bar IMEP, Lambda 1, 40 °C charge air temperature

Figure 12 shows the total heating value and the net heat release rate for different compression ratios at 17 bar IMEP and 85% energetic NH_3 share. These are the mean values of each measurement with the lowest COV of IMEP to compare the velocity of the combustion. With the CR 18.5, a slightly faster burn-through can be recognised. However, the combustion processes differ only insignificantly. The compression ratio therefore has only a minor influence on combustion rate. The efficiency analysis in Figure 13 comes to the same conclusion, that on the one hand the increase in compression has only little effect on the efficiency and on the other hand the influence on the energetic NH_3 share is small. This means that by increasing the compression ratio not significantly more energetic NH_3 share can be driven. The negative part of CR 22 is a combination of increased wall heat losses and blowby.

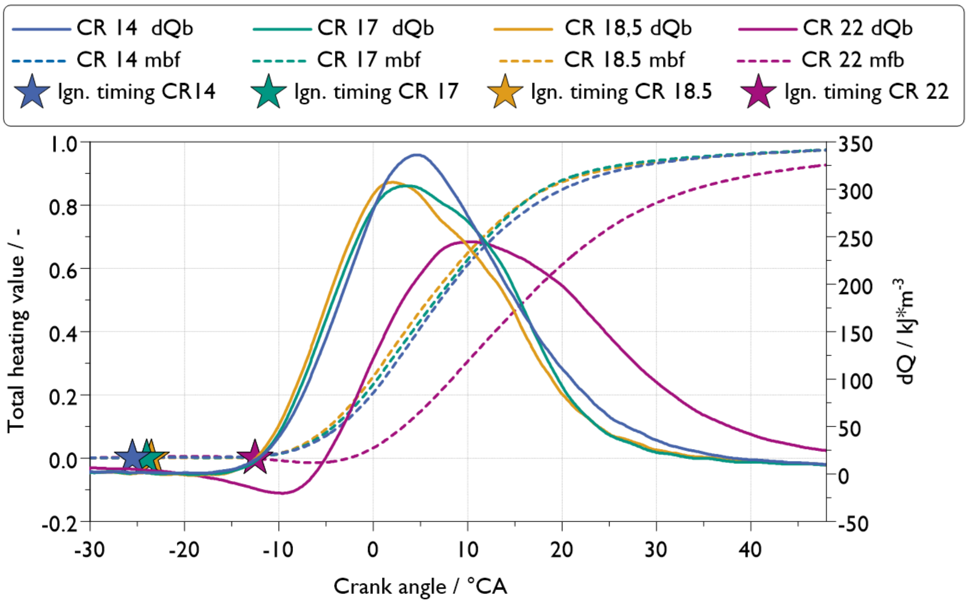


Figure 12: Total heating value, net heat release rate and ignition timing over crank angle for three different compression ratios at 17 bar IMEP, Lambda 1

If one first looks at the Figure 13 on the right, an increase in efficiency with load can be seen. At higher loads, significantly higher pressures and temperatures occur in the combustion chamber, which improve both the ignition and the decomposition of the ammonia and thus the burn-through behaviour. Both lead to the increase shown.

The dependence of the efficiency on the compression ratio is not uniform and not significant at higher loads. The diagram on the right-hand side shows values at maximum NH_3 share. The engine would not be operated directly at this limit. If looked at the full load point of 22 bar IMEP above the NH_3 share, the left-hand diagram is decisive. As shown in Figure 14, the possible NH_3 range at compression ratios 17 and 18.5 (22 is not possible here) is extremely narrow. The centre of the NH_3 range between the two limits is around 87% (red line). The difference in efficiency between the two compression ratios is in the range of the line thickness and can be neglected. As the pressure and temperature load on the engine are somewhat lower at CR = 17 (see Figure 15) and the knock susceptibility is also somewhat more moderate, the lower compression ratio would even be preferable.

As the right-hand side of the limit range is exceeded "more gently" with a compression ratio of 14 and the COV of IMEP does not rise as abruptly as with the higher compression ratios (see Figure 8), the engine could at least be operated very close to this limit. However, the efficiency would be one percentage point lower here and the running smoothness would be somewhat worse in principle. Overall, CR = 14 would therefore be unfavourable.

Somewhat surprisingly, the curve for CR = 22 is the lowest for the drivable load points, although mfb 50% and mfb 90% have the same values as at the other compression ratios. For a conclusive explanation, these operating points would have to be analysed in more detail. However, this was not done, as CR = 22 is out of the question for a real application due to the lack of full load capability.

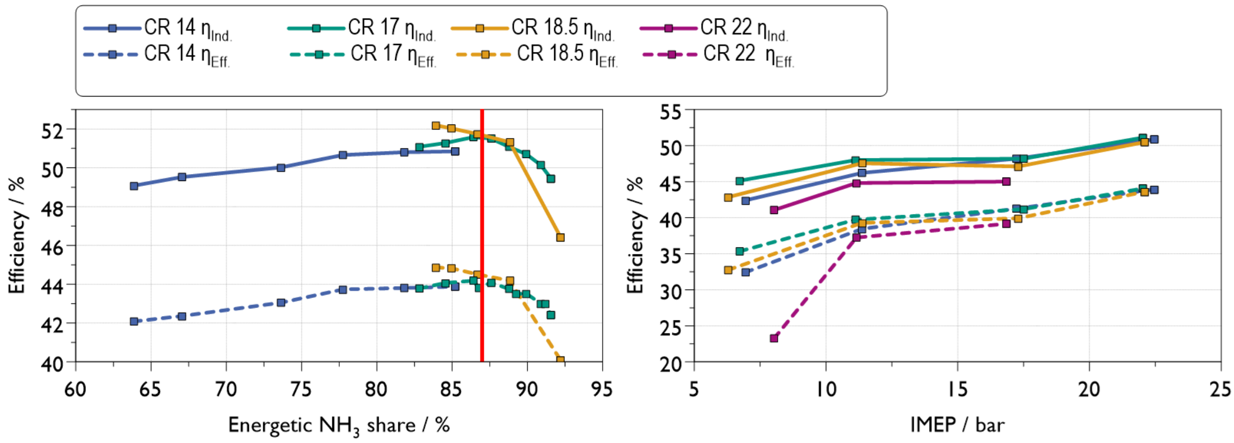


Figure 13: Indicated and effective efficiency a) over the energetic NH_3 share at 22 bar IMEP, Lambda 1 and 40 °C charge air temperature. b) over IMEP for the highest possible energetic ammonia share with a COV below 3%

Figure 13 shows the NH_3 range in which the engine can be operated for various loads and compression ratios. As mentioned, the limiting factors are a max. COV IMEP of 3%, a max. peak pressure of 220 bar, a max. exhaust gas temperature of 650 °C and a max. knock amplitude of 4 bar. The respective termination criteria are discussed in more detail in Figure 15. The basic aim is to operate the engine with the highest possible proportion of NH_3 - ideally without any addition of hydrogen.

The results show that pure ammonia operation is not possible, as the cyclical fluctuations increase sharply or combustion misfires also occur. The left-hand limit of the NH_3 range (max. possible H_2 share) should not actually be approached. However, it is important because it defines the width of the possible operating range. In the application presented in Chapter 2, the hydrogen mass flow is made available via a cracker unit. The actual mixing ratio of hydrogen and ammonia is therefore also dependent on the controllability of the cracker unit. A very narrow operating range with regard to the possible ammonia content places high demands here in order to avoid both misfiring and knocking in real operation at all times. In addition to the highest possible NH_3 content, the secondary objective is therefore also a sufficiently wide operating range.

A significant step in the right-hand limit of the operating range can only be observed for all loads when the compression ratio is increased from 14 to 17. This is most noticeable at the load of IMEP = 11 bar. From a compression ratio of 17, the trend is no longer clear. The shifts in the right-hand limit as a function of the compression ratio are marginal at 11 and 17 bar IMEP and almost within the range of measurement inaccuracy due to the determination of the relatively small hydrogen mass flow. For real operation and the design of the multi-cylinder engine, these dependencies are negligible. A preference as to whether CR = 17 or 18.5 should be selected cannot be derived from these measurements at these loads. At full load, however, the right-hand limit at CR = 17 is at a significantly higher NH_3 share. CR = 14 is unfavourable due to the significantly lower maximum NH_3 content and the lower efficiency. CR = 22 is unsuitable as full load operation is not possible. If looked at the width of the operating range, this is greater for CR = 17 at a load of IMEP = 11 for all loads, and even significantly greater at full load. At IMEP = 17 bar, the range is still larger but the difference is no longer as pronounced. From this point of view, a compression ratio of CR = 17 would therefore be the better choice.

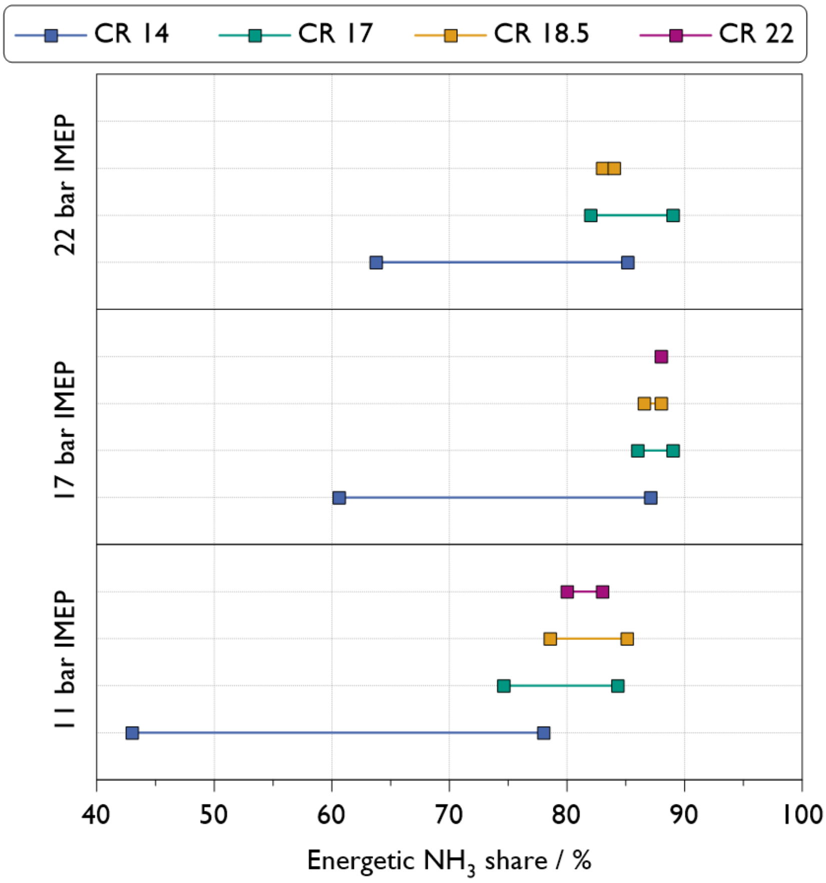


Figure 14: Possible energetic NH_3 range as a function of the compression ratio at different loads

Figure 15 shows the values of the termination criteria at full load relative to their upper limit ($T_{Ex.gas}$ = 650 °C, COV = 3% and Peak Pressure = 220 bar) and the target parameters (Max. NH_3 share = as high as possible, Achieved Load = 100%, Ind. Eff. = as high as possible) at the right operating range limit. It can be seen that the maximum exhaust gas temperature is never reached even at full load at all compression ratios and can therefore be categorised as uncritical. The peak pressure also has no limiting effect. In all cases, the maximum NH_3 content is defined by exceeding the limit value of 3% for the COV IMEP. The highest possible NH_3 content is reached at CR = 17 with 89%. At CR = 18.5, the indicated efficiency is one percentage point higher than with the two lower compression ratios. However, the better values with regard to the range of the possible NH_3 share at CR = 17 outweigh this efficiency advantage. The full load of 22 bar IMEP was also achieved for all compression ratios.

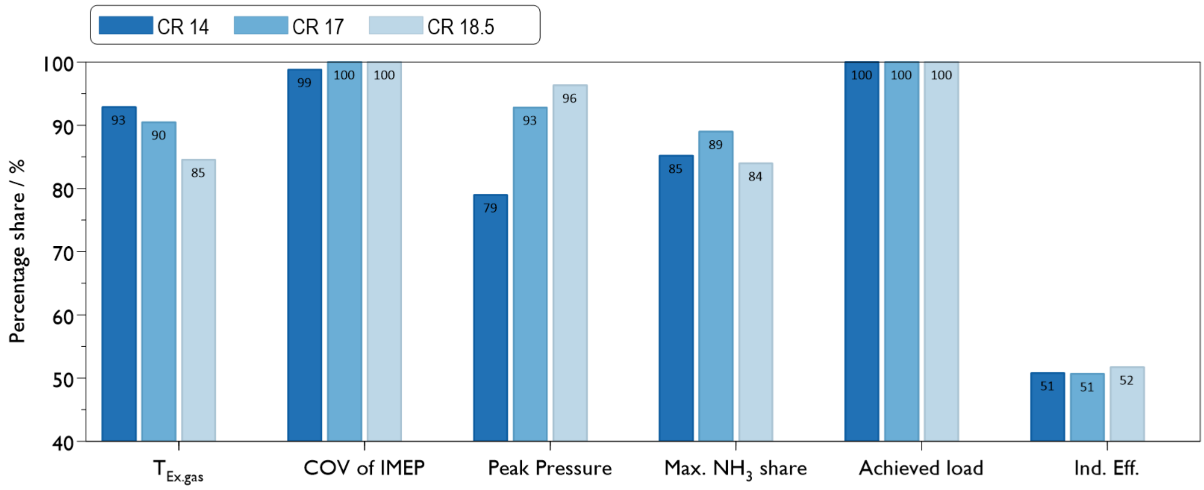


Figure 15: Termination criteria (relative to their limits: maximum exhaust gas temperature: 650 °C, maximum COV of IMEP: 3%, maximum peak pressure: 220 bar) and target parameters (max. NH_3 share and efficiency: as high as possible, load: 100% = 22 bar IMEP) for different compression ratios at 22 bar IMEP

In Figure 16, the emissions are plotted against the NH_3 share. The values for ammonia are at a relatively high level and initially increase significantly for CR = 14 from low to high NH_3 share in the fuel. In the possible operating range for CR = 17 of 82 to 89% NH_3 share determined according to Figure 14, the emissions are at an almost constant level of around 8 g/kWh. A drop in NH_3 slip immediately after exceeding the right-hand operating range limit must be assessed critically. If the sharp increase in cyclical fluctuations in reality leads to a sharp increase in NH_3 slip and the measuring device (FTIR) exceeds the measuring range limit, this measured value is taken into account in the internal averaging of the FTIR with zero and only leads to an apparent decrease in NH_3 slip. With the favoured compression ratio of CR = 17, the NO_x emissions are also at a similar level, albeit somewhat higher with values of 9 to 12 g/kWh. Exhaust gas aftertreatment to achieve permissible emission values is therefore essential.

However, the similar level of NH_3 and NO_x is an advantage. The ammonia contained in the exhaust gas can be used directly as a reducing agent in a SCR system. For a maximum conversion rate, a ratio of ammonia to nitrogen oxides of 1:1 should be aimed for [5]. The raw emissions are at least close to this ratio. In further planned tests with a real exhaust aftertreatment system, it is to be investigated whether and how much additional reducing agent must be injected in order to achieve the NO_x target values. Another open question is the cold-start efficiency of the exhaust gas aftertreatment system. Laughing gas is also clearly measurable as a raw emission in the exhaust gas with values of 0.07 to 0.15 g/kWh. These values can increase further in a downstream SCR system. This aspect must also be taken into account when designing the exhaust gas aftertreatment system. Hydrogen emissions are dependent on the compression ratio. Hydrogen in the exhaust gas is not only present as slip from the hydrogen share supplied, but also through the unburnt portion from the decomposition of ammonia. This takes place more efficiently at higher temperatures and therefore at higher compression ratios and leads to the observed increase with the compression ratio.

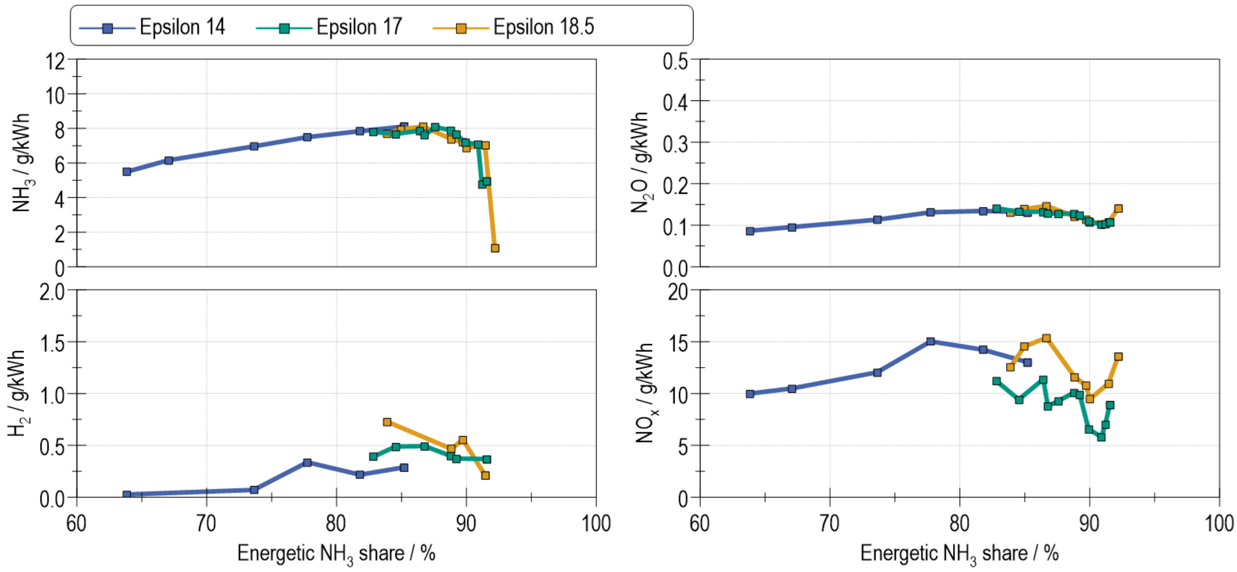


Figure 16: Emissions over the energetic NH_3 share for different compression ratios at 22 bar IMEP, Lambda 1 and 40 °C charge air temperature

(III) Variation of the ignition system

Another measure to increase the possible NH_3 content can be the selection of a suitable ignition system. The ignition energy and ignition phase are decisive factors for good combustion with low cyclical fluctuations. Independent of the ignition parameters themselves, combustion can be significantly improved by the mixture formation. Firstly, the correct local lambda must be present at the spark plug; this can be achieved either by good homogenization or targeted stratification. Secondly, cyclical fluctuations of the local lambda at the spark plug must be prevented. If a constant lambda cannot be guaranteed, then the ignition energy has to be further increased so that a reliable ignition occurs even in the worst case. The measurement presented in this chapter shows the influence of a higher ignition energy on the ammonia-hydrogen combustion process. A new ignition system from the company SEM was used for this purpose. This made it possible to adjust the spark duration and the spark current variably. The longer duration of the spark increases the probability that ignitable mixture will flow past the spark plug and ignite. The configuration of the engine for this measurement was ammonia and hydrogen via port fuel injection, ignited by a J-Gap spark plug at compression ratio 18.5.

The ignition current was measured on the primary side using a current clamp. The standard ignition system produces a single current peak for a single ignition spark (Figure 17 right). With the SEM Flexispark ignition system, it is possible to generate a longer ignition spark over several degrees of cranking angle (Figure 17 left). This increases the probability that an ignitable mixture comes close to the spark plug and can be ignited. With the capacitive Flexispark system, the ignition spark voltage, the ignition spark burning time and the ignition spark holding current can be variably adjusted. The Flexispark ignition system recognizes when a spark fails, then the capacitor discharge is switched off and the current is regulated by the plasma to a target value until it is switched off after a set time interval. This always triggers a spark and eliminates the excess current, which reduces spark plug wear [6]. The current is set in the closed control loop to values in 50 to 200 mA and the duration in 40 to 3000 μ s, which leads to spark energies in 2 to 150 mJ. This capacitive ignition system does not necessarily lead to increased wear of the spark plug. With a suitable choice of current and control duration, the ignition spark can be maintained in the low-wear glow discharge.

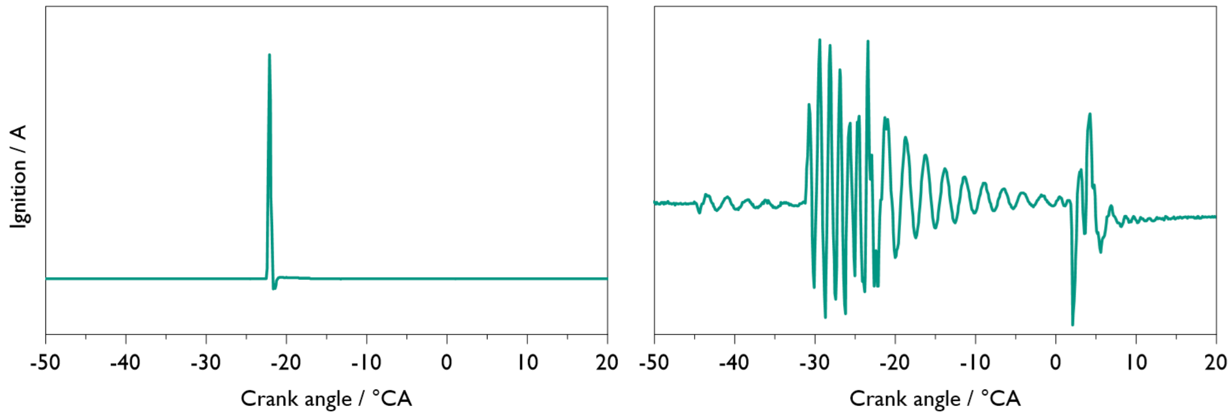


Figure 17: Ignition signal measured on the primary side with a current clamp.
left) Default inductive system with a single spark right) Capacitive Flexispark system with a multi spark ignition

At a compression ratio of 18.5, a full load point and a part load point were compared. The measurements were taken at Lambda 1 and an intake temperature of 40 °C. In Figure 17, top left, the COV of the IMEP is plotted against the energetic NH_3 share. At 11 bar IMEP, the limit was around 85% energetic NH_3 share; at 22 bar IMEP, 91% energetic NH_3 share could be realised. With the Flexispark ignition system, it was possible to completely eliminate the hydrogen and realise a 100% ammonia combustion process with a COV of IMEP below 3%. In general, combustion stability has been improved with the Flexispark ignition system.

Figure 18 shows the stabilization of combustion and its associated consequences. In the medium load range at 11 bar IMEP, the same COV of IMEP is achieved within the previously permissible range of the NH_3 share. However, there is no longer a limit on the maximum NH_3 share. Up to pure ammonia operation, the COV of IMEP only slightly increases from 1.6 to 2%. At full load at 22 bar, the COV of IMEP is even significantly better across the entire operating range compared to the previous ignition system. A 100% ammonia operation is also possible. The efficiency does not change at either load and continues to show a slight downward trend up to pure ammonia operation.

Initially, it may be surprising that at full load the efficiency does not increase despite the significant reduction in cyclic variations. The reason is evident from the diagrams on the ignition timing, the mfb50, and the maximum cylinder pressure (Figure 19). The improved combustion process would lead to an exceedance of the permissible peak pressure in at least some cycles if the mfb 50% remained the same. Therefore, for safety and comparability reasons, the peak pressure for both ignition systems were kept approximately constant. This required delaying the combustion timing. The center of combustion was shifted by about 2 to 3° CA, which is thermodynamically disadvantageous in principle. Thus, the efficiency remains "only" constant, even though the cyclic variations could be reduced.

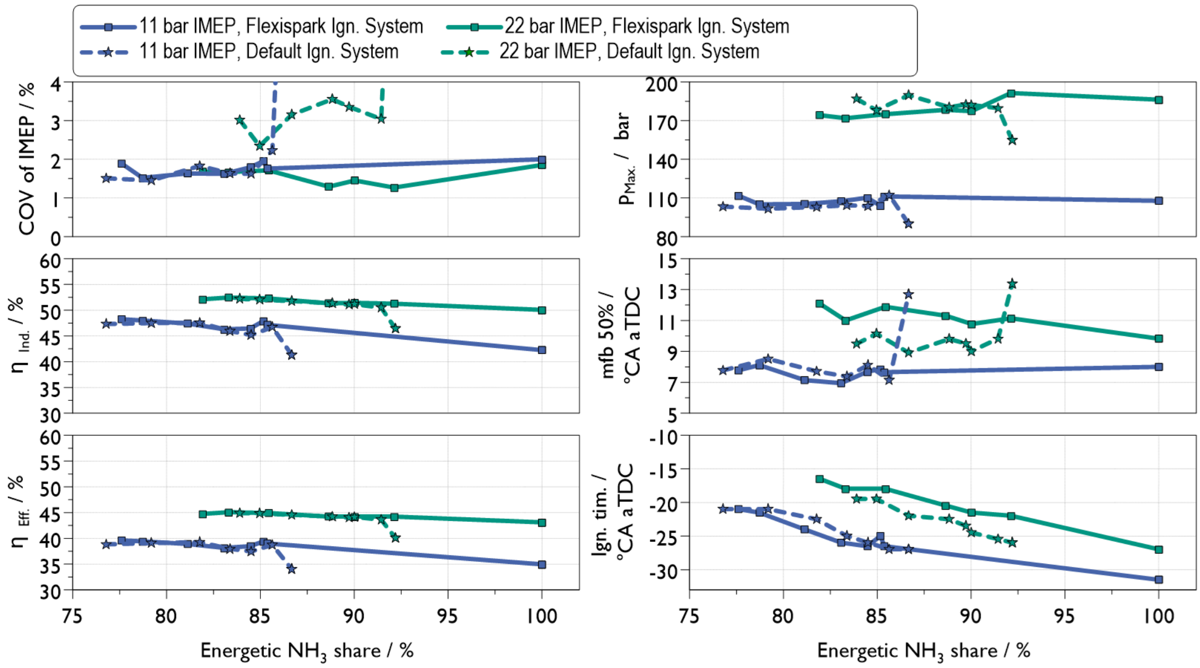


Figure 18: Comparison of the ignition systems. COV of IMEP, indicated efficiency, effective efficiency, peak pressure, mfb 50 % and ignition timing over the energetic NH_3 share for 50% and 100% load at CR 18,5, Lambda 1 and 40 °C charge air temperature

The following Figure 19 and Figure 20 describe the improvement in combustion behaviour. Figure 19 shows 100 consecutive combustion cycles for the original and the Flexispark ignition system vs. the load. Ignition timing and start of combustion (mfb 10%), center of combustion (mfb 50%) and end of combustion (mfb 90%) are shown. The ignition timing is constant for all 100 cycles. To demonstrate the advantage of the Flexispark ignition system, operating points were plotted with an NH_3 share at which significant cyclical fluctuations already occurred with the conventional ignition system.

The ignition points are set in such a way that, in the stable operating range, combustion centers of approx. 8° CA a. TDC occurred. With the Flexispark ignition system, the mfb 50 % could be kept constant. A significantly smaller combustion delay can also be recognised, which, with the exception of 22 bar IMEP, also exhibits smaller fluctuation ranges. This also applies to mfb 50% at all loads and in particular to the end of Combustion at mfb 90%. The difference in the fluctuation range is most pronounced here. The stabilisation of the ignition phase therefore has a positive influence on the entire combustion process and is ultimately responsible for the fact that operation with pure ammonia is possible.

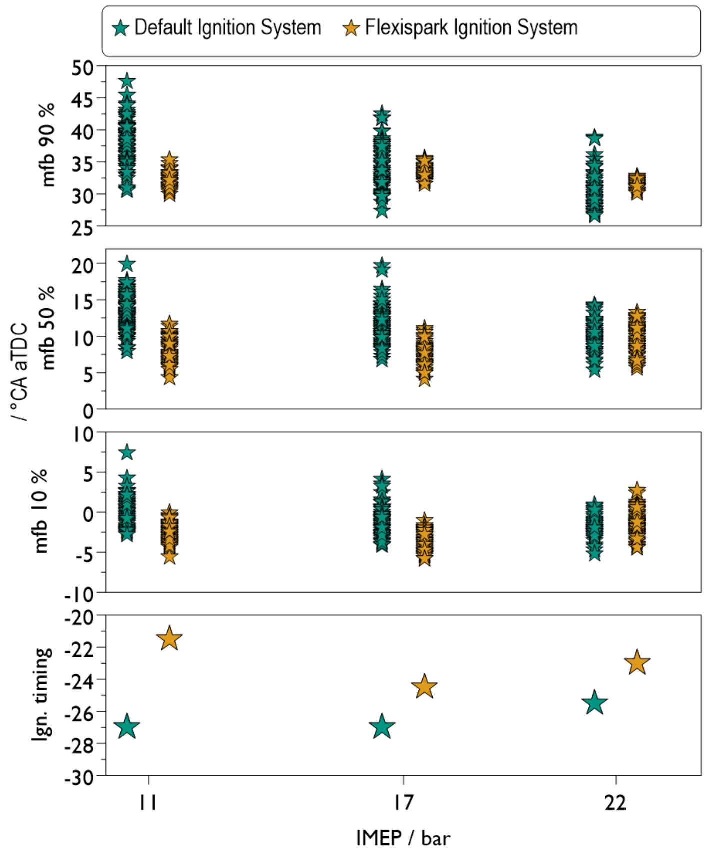


Figure 19: Comparison of the ignition timing, mfb 10%, mfb 50%, mfb 90% over the IMEP for the default und Flexispark ignition system at Lambda 1.

Figure 20 shows the operating point IMEP = 17 bar from Figure 19 in more detail. On the left, the mfb 90% is shown above the mfb 10%. As already described, the fluctuation range in both variables is generally smaller with the Flexispark system. However, a comparison of the two values also shows that with the Flexispark system, later mfb 10% does not lead to a later end of combustion to the same extent. Overall, the combustion process is much more stable and less subject to cyclical fluctuations. With the conventional ignition system, the green dots show a clear tendency for a late end of combustion to correlate with a large combustion delay. This means that difficulties with ignition continue throughout the entire combustion process. The combustion process has a stable overall shape, but shifts analogue to the start of combustion. The overall slower combustion process can be clearly seen in the right-hand diagram in Figure 20. The average value over all 100 cycles is shown here.

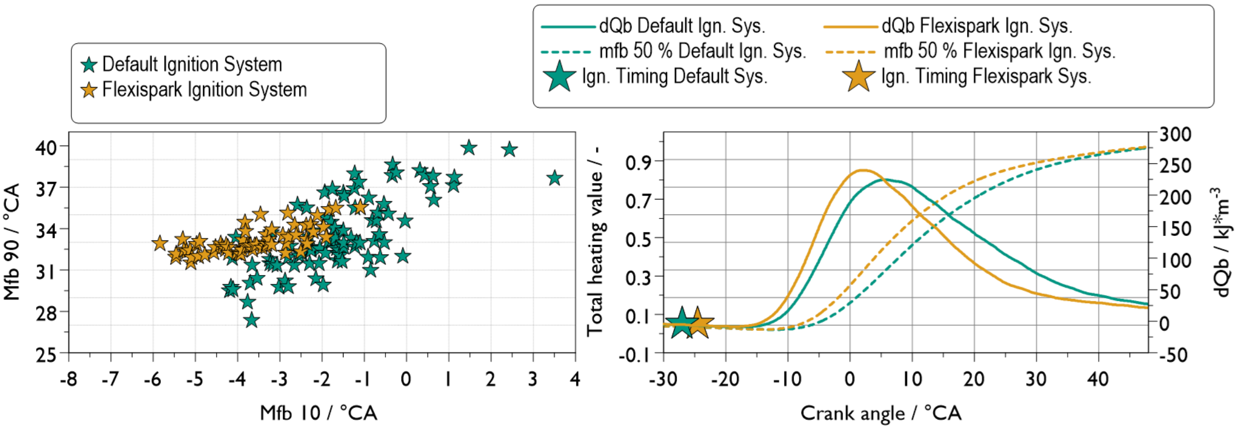


Figure 20: a) Comparison of mfb 90% over mfb 10% for the default and Flexispark ignition system at 17 bar IMEP, Lambda 1 and 92% energetic NH_3 share b) Total heating value and net heat release rate over crank angle for the default and Flexispark ignition system

4. Summary and outlook

The urgent evidence to reduce CO_2 emissions in all areas demand the quick realization of respective solutions for the worldwide shipping branch both for sea-going and inland waterway vessels. Therefore, several options are viable, most of them based on green e-fuels. Ammonia, produced on basis of renewable energies e. g. solar and wind power, looks as a promising candidate for shipping as worldwide trading and handling are already in place which eases the establishment of a fuel supply chain.

The presented project handles the development of a maritime propulsion system based on a cracker-engine-unit for inland waterway vessel application. Within this presentation the technological aspects for establishing such system concept were discussed. Special focus is set here on the combustion concept development on a single-cylinder research engine as basis for the upcoming multi-cylinder engine setup.

As ammonia requires a very high ignition energy compared to conventional fuels, which cannot be readily provided by most conventional ignition systems, measures to improve ignition are necessary. In this case, this is achieved by adding hydrogen to the ammonia. The hydrogen can be ignited with little energy, whereby the energy released from the hydrogen combustion leads to the decomposition of ammonia and its ignition. The supply of hydrogen via the aforementioned cracker unit costs money and energy. The primary aim of the investigations was therefore to define engine boundary conditions under which as little hydrogen as possible is required. Ideally, the combustion process should work with pure ammonia. The influence of injection timing, injection pressure, charge air temperature and direct injection of ammonia have already been investigated in initial studies [7]. The investigations in this paper focused on the variation of the air/fuel ratio, the compression ratio, and the ignition energy.

A lambda value of 1 proved to be the best solution for the air/fuel ratio. The combustion process reacts sensitively to lambda changes. A very slight deviation into the rich range could possibly stabilise the ignition behaviour somewhat, but the hydrogen emissions then increase exponentially. The nitrogen-based emissions could be reduced. Significant leaning beyond approx. lambda = 1.3 is not possible, as the cyclical fluctuations then increase abruptly. The resulting air thinning does not yet contribute to a reduction in nitrogen oxides.

The increase in the compression ratio was based on the idea that the higher pressures and temperatures have a positive effect on the ignition conditions and thus enable higher maximum ammonia contents. This showed that an increase does not always bring advantages. At CR = 22, full-load operation is not possible due to peak pressures being exceeded. This compression ratio can therefore be excluded. Compared to the basic compression ratio of CR = 14, the increases to 17 and 18.5 were able to achieve advantages in terms of both the maximum possible NH_3 content and efficiency. Significant in both cases, however, is the much higher susceptibility to knocking, which severely limits the range of possible NH_3 content in the fuel. However, the differences in emissions and efficiency between these two compression ratios are small. Based on these findings, CR = 17.5 is preferable.

Last but not Least, the conventional ignition system was replaced by a Flexispark ignition system from the company SEM. The system provides an extended ignition spark with increased ignition energy. Thanks to the improved ignition behaviour, the combustion was stabilised overall, i.e. the cyclical fluctuations were reduced. It was also possible to operate the engine with pure ammonia without adding any hydrogen. Here, too, the COV of the IMEP fell below the set limit value of 3% at all times. It remains to be investigated whether the new ignition system significantly reduces the service life of the spark plug. It also remains to be seen whether another ignition strategy, such as conventional spark ignition in a passive pre-chamber, could achieve a similar result. Both are already the subject of ongoing research.

In summary, it can be stated that a stable combustion process could be realised in a diesel-based engine with intake manifold injection of hydrogen and ammonia at a compression ratio of CR = 17. The required hydrogen admixture was low at approx. 10 to 15% and could even be completely avoided with a potent ignition system. The full load of the diesel engine could be achieved. The effective efficiency of approx. 44% is at a very good level. The raw nitrogen oxide emissions are high, as are the NH_3 emissions. However, due to the favourable concentration ratio of the latter two, even a passive SCR system could be sufficient to comply with existing limit values. This aspect is also the subject of future investigations.

After technology transfer from the single-cylinder engine, the multi-cylinder engine will be containerized and tested at the CAMPFIRE Open Innovation Lab (COIL) near to Rostock/Germany. There the marriage with the Ammonia cracker will happen as well, allowing the complete system to be initially tested and calibrated, followed by an endurance run under application typical conditions.

Acknowledgments

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