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Assessment of methanol and hydrogen combustion in large engines

Retrofit Solutions

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ABSTRACT

The interest in switching from fossil fuels to a broad variety of climate-neutral alternatives is growing in the large engine industries. Based on the specific requirements of the application and the specific properties of alternative fuels, an appropriate selection of fuel and combustion concept must be considered.

Typical applications of large engines on which this paper puts its focus are marine applications and stationary power generation applications. Hydrogen is considered on one hand a potential energy carrier to bridge and balance across sectors and to fill the gap between power generation and consumption in terms of time and place. In view of the very low volumetric energy density of hydrogen, on the other hand, methanol is regarded as one of the most promising alternative fuels in marine applications.
This paper performs a comprehensive review of several conceivable combustion concepts to run engines on alternative fuels and discusses their advantages and disadvantages. Both of quick-to-market solutions based on retrofitting of existing diesel or gas engines and solutions dedicated for the alternative fuels are covered in the assessment.

For stationary applications, the spark-ignited and diesel-ignited pre-mixed hydrogen combustion concepts are compared in detail based on the single cylinder engine test results. Their operational limitations and observed combustion anomalies are summarized and possible countermeasures are discussed.

For marine applications, a diesel-ignited pre-mixed methanol combustion is compared with a diffusive methanol combustion with the high pressure methanol direct injection and diesel pilot injection based on the single cylinder engine test results. Their performance in terms of engine efficiency and exhaust emissions as well as their operational limitations are discussed.

The paper finally assesses the potential and challenges of the combustion concepts demonstrated on the single cylinder engine and outline the further development steps in the future.

1 INTRODUCTION

Decarbonization and greenhouse gas (GHG) reduction in all energy sectors is currently one of the most important and debated topics in addressing the global challenge of climate change. Large engines are no exception from this global movement, and they would have to be either replaced by the other CO_2 neutral prime movers like fuel cells (when they are fueled by green hydrogen) or they must be able to run on CO_2 neutral alternative fuels. 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.

The International Energy Agency (IEA) prospects that the share of fossil fuels in the final consumption starts to fall towards 2050 from 66% today to 55% with the stated policy scenario (STEPS) [1], as shown in the Figure 1. In the announced pledges scenario (APS) and net zero emission by 2050 scenario (NZE) the declines are steeper, and the share of fossil fuels falls to 35% and less than 20%, respectively.

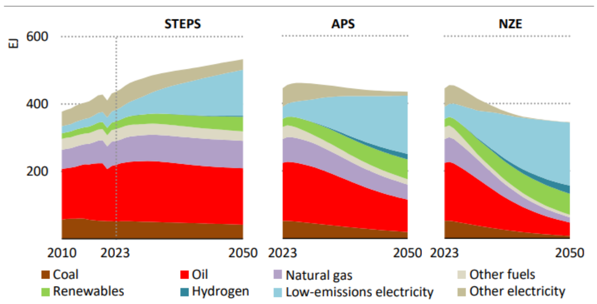


Figure 1. Total final energy consumption by fuel and scenario for 2010-2050, IEA [1]

The electrification of end-uses together with the decarbonisation of power generation is the main contributor for the decline of the fossil fuels share. The IEA expects that the share of renewable energy source in the electricity generation, led by soler PV and wind, increases significantly towards 2050. As the share of renewable energies increases, however, the need for flexibility to balance the gap between the supply of renewable energies and the demand for electricity in terms of time and location will increase rapidly.

Hydrogen is considered as one of the potential energy carriers to fill such gaps as well as to bridge and balance across sectors. The blending of hydrogen with natural gas to fuel gas engine power plants is seen as a relatively straightforward, short-term solution that could meet such flexibility needs. In the mid-term future, engines powered solely by hydrogen will fully realize their zero-emission potential.

On the other hand, there are sectors where the electrification is restricted, at least currently, by technical and commercial limitations such as aviation and shipping. Marine transportation is a critical facilitator of the global economy, but its contribution to total GHG emissions has grown [2]. In response, the Marine Environment Protection Committee (MEPC 80) in the International Maritime Organization (IMO) adopted a revised GHG Strategy in July 2023 with ambitious emissions reduction targets. These targets include a 20% reduction by 2030, a 70% reduction by 2040 (compared to 2008 levels), and ultimately, achieving net-zero emissions by 2050, as shown in Figure 2, [3].

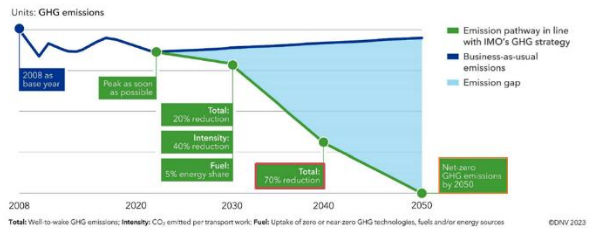


Figure 2. Revised IMO GHG strategy, DNV [3]

As the electrification of marine propulsion system or a replace of internal combustion engines (ICE) with the other prime movers like fuel cells face technical and economical challenges, it is considered as realistic solutions to run the ICEs on low emission fuels such as biofuels and hydrogen-based fuels. In view of the very low volumetric energy density of the hydrogen, ammonia and methanol are regarded as ones of the most promising alternative fuels for marine applications compared to the conventional marine diesel oil.

This paper performs a comprehensive review of several conceivable combustion concepts to run engines on alternative fuels and discusses their advantages and disadvantages. Both of quick-to-market solutions based on retrofitting of existing diesel or gas engines and solutions dedicated for the alternative fuels are covered in the assessment. For stationary applications the focus is put on the pre-mixed hydrogen combustion considering the retrofitting of existing diesel and gas engine power plants whereas the diesel-ignited pre-mixed and diffusive combustions of methanol are discussed in detail considering the fuel redundancy of dual fuel engines preferred for marine applications.

Engine concepts for ammonia combustion are treated in a separated paper also submitted to the present CIMAC congress, paper No. 291 [4].

2 COMBUSTION CONCEPTS

The characteristics of different gas and dual fuel combustion concepts have already been discussed in previous papers [5] and [6]. Their tolerance for the utilization of hydrogen, ammonia and methanol as alternative fuels are summarized in Table 1.

2.1 Pre-mixed combustion with spark ignition

One possible and perhaps easier path is to go for a premixed combustion concept, in which the alternative fuel can be mixed with the air either upfront of the combustion chamber, e.g. in the intake port via gas admission valves/injectors (port fuel injection, PFI) or directly in the combustion chamber using what is known as low pressure direct injection (LPDI). Detailed comparison of the PFI and LPDI will not be discussed in the present paper but a proper gas supply concept must be selected depending on the characteristics of the alternative fuel to be applied, fuel injection equipment (FIE) availability, and cylinder head design compatibility. The pre-mixed air-fuel mixture will then be ignited either by a spark plug or by injecting diesel fuel.

The majority of large natural gas engines on the market today employs spark-ignited pre-mixed combustion concept and apply either an open chamber concept (open chamber spark ignition, OCSI) or a pre-chamber combustion concept (pre-chamber spark ignition, PCSI). In this paper, only the fuel-fed pre-chamber – sometimes also called as active pre-chamber – is included in the PCSI and the pre-chamber spark plug – also known as passive pre-chamber – is included in the OCSI due to its similarity of the engine configuration i.e., no need for the second fuel supply to the pre-chamber.

The main market of the current large natural gas engines is stationary power generation application. The conversion of such engines to alternative fuel probably will favor the application of hydrogen in several steps. The hydrogen must be produced using an excessive power generated by renewable energy sources and converted again to power when and where the power demand cannot be covered by the renewable energy sources. Thus, any further fuel conversion that would consume additional power leads to poor total efficiency and more cost. The first steps of the hydrogen utilization will be an engine operation with the mixed fuel of natural gas and hydrogen. Eventually, the portion of the natural gas might be reduced until the engine runs on the hydrogen only depending on the availability of the hydrogen from the excessive power.

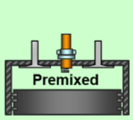
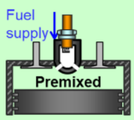
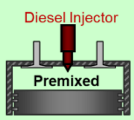
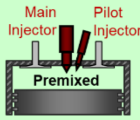
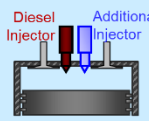
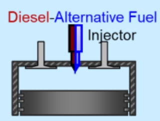
2.2 Pre-mixed combustion with diesel ignition

By adding or adapting the gas supply system, it is feasible in a relatively simple manner to burn alternative fuels, at least partly, on diesel or dual fuel engines. One of the biggest advantages of such diesel substitution or dual fuel engines is the redundancy of engine operation in diesel mode. It is possible to continue the engine operation independently of the availability of the alternative fuels or even in case of troubles of subsystems related to the alternative fuels. The fuel redundancy of this concept would be highly appreciated especially in marine applications.

Dual fuel (DF) large engines typically have a dedicated diesel pilot injector that can inject a small quantity of diesel fuel enabling a high substitution ratio and thus, offers a potential for low carbon emissions. The conventional diesel engines with either mechanical or common rail injection systems usually have only one diesel injector which is designed for the rated power. Therefore, the capability of stable injection at low injection quantities can be limited and thus, the possible substitution rate may be limited.

Furthermore, DF engines usually have a moderate compression ratio to assure a safe natural gas combustion, which is also favorable for combustion of pre-mixed hydrogen or methanol. The typical

Table 1. Combustion Concept for Alternative Fuel

Strategy	**Quick Time to Market Approach**				**Dedicated Concept**	
Mixture Formation	**Port Fuel Injection**				**Direct Injection**	
Combustion and Ignition Concept	**Pre-mixed combustion Spark ignition**		**Pre-mixed combustion Diesel pilot injection**		**Diffusive combustion Diesel pilot injection**	
						
Substitution rate	**100%**		**30 ~ 90%**	**> 95%**	**~ 95%**	
Diesel back-up capability	**N.A.**		**100%**	**100%**	**30 ~ 100%**	

compression ratio of the conventional diesel engines, however, may be too high for the pre-mixed combustions and the possible engine power output may be limited by abnormal combustions such as pre-ignitions and knocking combustions.

2.3 Diffusive combustion with diesel pilot injection

The other option is a diffusive combustion concept, in which the alternative fuel is injected directly into the combustion chamber (high pressure direct injection, HPDI) at a high pressure of around 500 bar and higher. The injected alternative fuel is then ignited by injecting the pilot diesel.

While the pre-mixed combustion concept could be realized in a relatively simple manner by adapting the gas supply system, the associated challenges such as combustion anomalies and exhaust gas emissions pose a limitation on the achievable substitution rate and potential reduction of GHG gas emissions. Diffusive combustion concept could offer a good solution to these known issues of the pre-mixed combustions at a cost of the significantly increased complexity of the engine and subsystems with the high-pressure fuel injection systems for two fuels. Another challenge is the availability, reliability and durability of high-pressure injectors and high-pressure pumps that are compatible for the selected alternative fuel, which are not fully proven yet.

When the baseline engine is a dual fuel engine with a dedicated diesel pilot injector, the main diesel injector could be replaced by an injector for the alternative fuel. The separate dedicated pilot injector is able to provide a small quantity of diesel as an ignition source and thus, the concept can achieve a quite high substitution rate of around 95% or higher. In this case, however, the fuel redundancy is not available or quite limited since the diesel can only be injected by the pilot injector.

To keep the fuel redundancy up to rated power in diesel operation, the injector would have to be a multi-needle injector that can inject both diesel and alternative fuel. There would not be sufficient space available to install a full-load capable diesel injector and a full-load capable alternative fuel injector in a cylinder head of typical high-speed and medium-speed 4-stroke large engines.

3 ASSESSMENT OF METHANOL AND HYDROGEN COMBUSTION

The characteristics and challenges of each combustion concept mentioned in the previous section when combined with hydrogen or methanol are described in this section. The authors reported some of the former test results also in the previous publications [7], [8]. The discussion bases on the single cylinder engine test results when available.

The test results and discussions when combined with ammonia are published separately at the present CIMAC Congress with the paper No. 291, [4]. Furthermore, the authors published ammonia test results in former publications such as [9], [10], [11].

3.1 Test Engines

Figure 3 depicts the series of AVL’s modular large-bore single cylinder engines (SCE) that are characterized by a high level of flexibility enabled by following design features [12].

- Common base frame for engine, support bearing and dynamometer
- Oil sump and main auxiliary system of pumps, pipes, valves and flowmeters are integrated into base frame
- Modular engine design consisting of rigid crankcase and customer specific cylinder block enabling easy reconfigurations to different bore, stroke and power unit set-up
- Easily accessible crank train and valve train for inspection or implementation of telemetric measurement system
- Flexible camshaft for simple adjustable valve timing
- 300+ bar PFP available for specific bore ranges

Figure 3. AVL’s Modular Large-Bore Single Cylinder Engines

The SCE is a cost-effective substitute for a typical first-generation prototype engine for combustion development and therefore, one of the most versatile development tools to shorten development time and reduce development cost.

- Enables hardware definition of fuel injection equipment, combustion system and power unit components
- Majority of engine calibration work can be performed at attractive cost on SCE for all Diesel, Gas and Dual Fuel combustion systems
- Ideal tool for advanced combustion research enabling a maximum optical access for combustion visualization
- Mechanical development and validation of complete power unit

The test results shown in this section of the paper were measured on the single-cylinder test engines based on AVL SCE260 and SCE350 with different power units and different combustion system configurations.

3.2 Hydrogen pre-mixed combustion with spark ignition

Both combustion concepts with and without pre-chamber can tolerate hydrogen well to a certain extent although the high reactivity of hydrogen poses the challenge of managing abnormal combustions. Often, the power output may also have to be reduced compared to the baseline natural gas engine to suppress combustion anomalies. Proper mixing of hydrogen with air including reasonable setting of excess air ratio (EAR) and/or EGR rate, appropriate scavenging of the residual gas, cooling of combustion chamber components, and minimization of oil consumption are ones of the most important development topics to realize a safe and reliable hydrogen combustion. On the other hand, the high EAR or high EGR rate that is required to reduce the combustion speed results in very low NOx emissions providing a possibility to avoid any exhaust aftertreatment system.

Authors performed some initial tests using an SCE350-based single cylinder test engine to examine the combustion characteristics of the hydrogen as well as the mixture of the natural gas and hydrogen. The baseline power unit was equipped with a fuel-fed pre-chamber and spark plug. The fuel gas, either the mixture of natural gas and hydrogen or the pure hydrogen, was mixed with the air through a gas admission valve installed in the intake manifold close to the cylinder head. The compression ratio of the engine was reduced by 2 units from the baseline gas engine.

Figure 4 and 5 shows an example of the test results where the volumetric fraction of the hydrogen in the fuel gas was varied at an indicated mean effective pressure (IMEP) of 11 bar. 0% in the horizontal axis means that the engine was operated with the pure natural gas and 100% means that the engine ran with the pure hydrogen.

Figure 4 illustrates the cylinder pressures in the main combustion chamber and in the pre-chamber as well as the rate of heat release. The black curves demonstrate the measurement results with the natural gas at methane number 65 whereas the red and blue ones are measured with 30% and 70% volumetric fraction of hydrogen, respectively, and green ones are the results with the 100% hydrogen. During this measurement the ignition timing was kept constant and the air fuel ratio was adjusted aiming to control the combustion speed as similar as possible to that of the natural gas combustion.

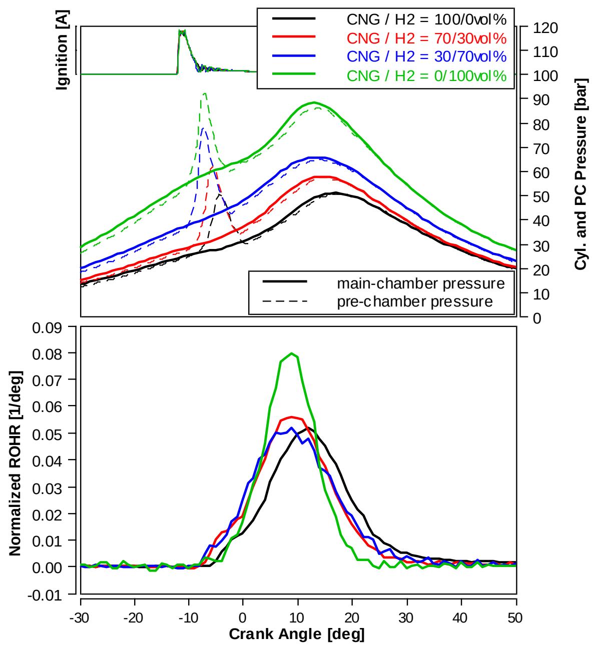


Figure 4. Cylinder pressures in the main and pre-combustion chamber and rate of heat release at different volumetric fraction of hydrogen in the fuel gas

An overview of the engine performance over the volumetric fraction of the hydrogen in the fuel gas is provided by the Figure 5. It can be confirmed that the air fuel ratio was increased consistently. The combustion speed could be well controlled by an adjustment of the air fuel ratio up to 80 to 90% volumetric fraction of hydrogen. At the 100% hydrogen, however, the combustion was quite fast even with the excess air ratio of 4.5. The carbon dioxide emission decreases with the increased hydrogen portion down to nearly zero at 100% hydrogen. It does not hit zero because there is a small quantity of carbon dioxide emission due to oil consumption. The NOx was also reduced to nearly zero at hydrogen ratio of 40% and higher due to extremely lean combustion. The THC emission slightly increased towards 50% hydrogen fraction due to increased excess air ratio and then decreased down to close to zero at 100% hydrogen. The unburned hydrogen increased with the hydrogen admixing ratio reaching around 1600 ppm. The brake thermal efficiency was increased by approximately 2% points mainly due to the fast combustion and minimized losses of unburned fuel. Good combustion stability was maintained in the whole operation range tested.

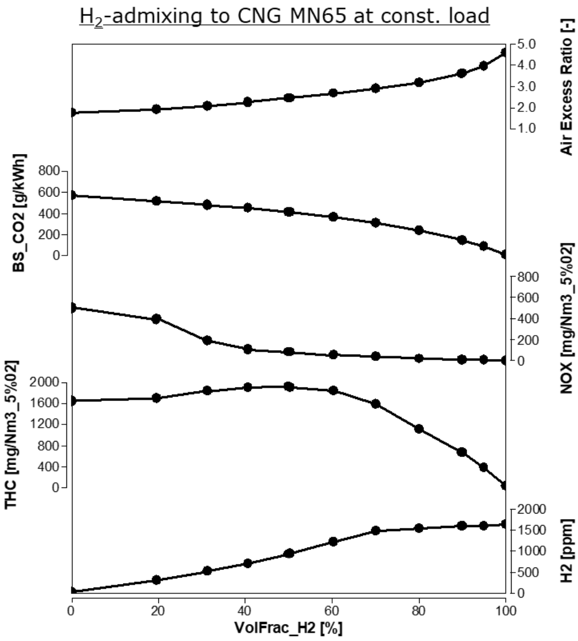


Figure 5. Engine performance over the volumetric fraction of the hydrogen in the fuel gas

Figure 6 shows the distribution of the NOx emissions over the entire engine operating range that could be run without combustion anomalies. The operation conditions are summarized by the indicated mean effective pressure in the vertical axis and the hydrogen volumetric fraction in the horizontal axis.

As mentioned earlier, the excess air ratio was increased to control the combustion speed when the volumetric fraction of hydrogen was increased, and as a result the NOx emission came down to below 100 mg/Nm³ when the hydrogen volumetric fractions was above 50%.

The knocking combustion was not observed in the entire operating area. The reasons that were limiting the further increase of the IMEP differs depending on the hydrogen volumetric fractions indicated by the circles depicted in the Figure 6.

In the range of hydrogen ratio below 50% indicated by the green circle, no combustion anomaly was observed, and the IMEP was limited only due to the boost pressure capability of the test bed.

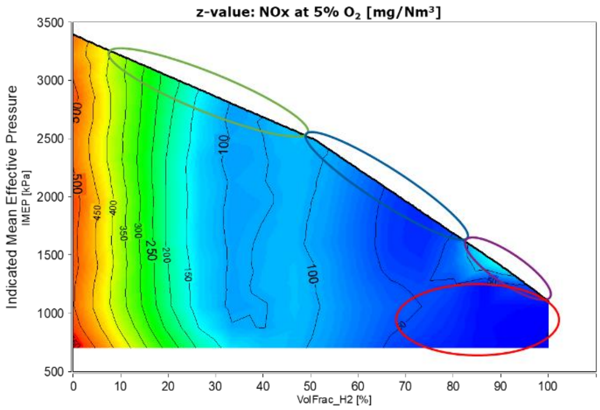


Figure 6. Distribution of the NOx emissions over the entire engine operating range of IMEP vs. hydrogen volumetric fraction

Above 50% hydrogen indicated by the blue circle in the Figure 6, the engine started showing the HCCI-like combustions as shown in the Figure 7. The black curve in the diagram is the main chamber pressure and the red dotted line is the pressure in the pre-chamber. As can be seen, the combustion started almost at the same time in both chambers and the start of combustion could not be controlled anymore by the ignition timing. The combustion continued stably even when the ignition was cut off, indicating that the mixture might be self-igniting. This type of combustion was quite fast with a combustion duration of 10 degrees crank angle or shorter. However, the combustion itself was quite stable over time. It may have been possible to run the engine as it is, but the testing was interrupted when such an HCCI-like combustion started happening.

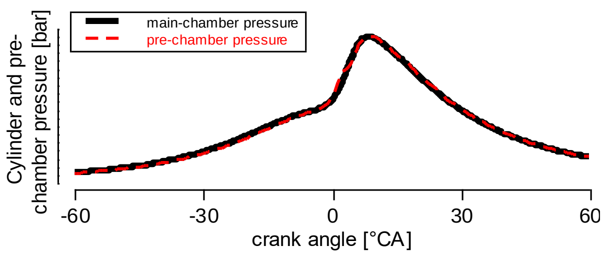


Figure 7. HCCI like combustion observed at hydrogen volumetric fraction of above 50%

At higher hydrogen ratio of above 80% indicated by the purple circle, the pre-ignition events started happening. The pre-ignition observed in this operating range is characterized by a start of combustion about 20 to 40 degrees crank angle earlier than the ignition timing. The onset of the pre-ignition was observed both in the pre-chamber and in the main chamber. When such a pre-ignition happens, the following combustion may result in knocking combustion. Near the shown limit line indicated by the purple circle in the Figure 6, it is rather a stochastic event, and normal combustion cycles can follow again for a while after a single pre-ignition event. However, the frequency and intensity of the pre-ignitions started growing when the engine is further operated beyond the limit.

In addition, very early pre-ignition events as shown by Figure 8 were observed in a wide area where the hydrogen volumetric fraction was above 70% as indicated by the red circle in the Figure 6. The ignition can happen even before the intake value closure (IVC), which is also known as backfire but also after the IVC in the very early phase of the compression stroke. As the mixture is already consumed at the event of the pre-ignition, there is no further combustion at the actual ignition timing. It was observed that such an early pre-ignition can happen both in the pre-chamber or in the main chamber.

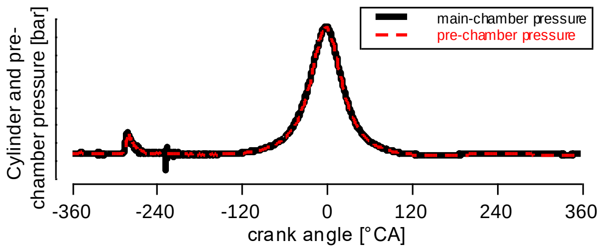


Figure 8. Early pre-ignition before intake valve closure (backfiring) observed in the range of hydrogen volumetric fraction above 70%

One possible cause for such an early pre-ignition is the contact of the new hydrogen mixture with the hot residual gas. An optimization of the valve timings to improve the scavenging of the combustion chamber with the fresh air and an optimization of the gas admission timing are ones of the key development topics to reduce the risk of the early pre-ignitions.

3.3 Hydrogen pre-mixed combustion with diesel ignition

The investigations were performed on an SCE260-based single cylinder engine in the section 3.1 equipped with a mechanical fuel injection system for diesel and a port gas admission valve for hydrogen.

Figure 9 and 10 summarizes the influence of diesel substitution rate on specific exhaust gas emissions and rate of heat release from one of the initial measurements. Diesel substitution rate of 0% means the engine is running only on diesel fuel and 100% would mean that the engine is running purely on hydrogen. The measurements were conducted at a constant diesel injection timing. The overall EAR calculated considering both the diesel and hydrogen mass flow rates was also kept constant at a level that is clearly higher than the normal diesel operation.

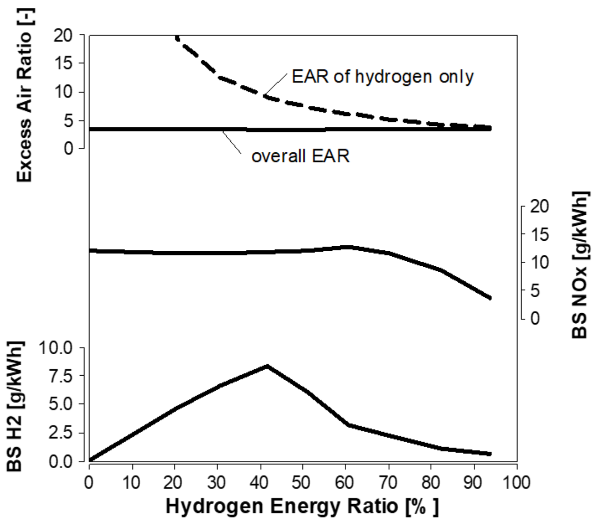


Figure 9. Influence of diesel substitution by hydrogen on engine performance at a part load

At lower diesel substitution rates, high unburned H2 emissions were measured. The reason is the EAR considering only hydrogen, which represents the actual EAR of air/hydrogen mixture between the diesel flames, is much higher than the above mentioned overall excess air ratio. The unburned H_2 emissions then decrease as the diesel substitution rate increases.

Similarly, the combustion duration becomes slightly longer at low substitution rates because a long and weak flame propagation continues after the main diesel combustion. As the substitution rate is further increased, the combustion duration is shortened at a hydrogen energy fraction in the range of about 50 to 70% (energy based) since the EAR considering only hydrogen is already low enough for fast pre-mixed combustion, while diesel combustion still accounts for a significant fraction, leading to simultaneous diffusive diesel combustion and pre-mixed hydrogen combustion. Pre-ignition events were also observed in this range of substitution rate. Finally, the combustion slows down again when the hydrogen energy fraction is further increased since the most part of the combustion then proceeds with the flame propagation.

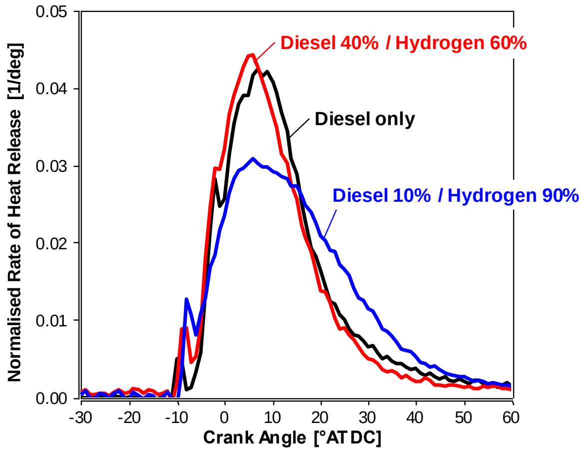


Figure 10. Influence of diesel substitution by hydrogen on rate of heat release at a part load

NOx emissions stay almost constant up to around 60% substitution rate and then decreases significantly as the major part of the heat release comes from the lean pre-mixed combustion.

The pre-mixed hydrogen combustion converted from conventional diesel engines or dual fuel engines suffers from the abnormal combustions at higher engine load points because the compression ratio of the baseline engine is too high. In addition, the oil consumption of diesel engines is tendentially somewhat higher compared to natural gas engines, which would additionally elevate the risk of pre-ignitions. Furthermore, the combustion deposits built from the diesel combustion form another possible source for the pre-ignitions. While a substitution rate of 30 to 40% (energy based) that correspond to 30 to 40% CO_2 reduction could still be reached without a big issue even at a rated load, a higher substitution rate is quite challenging without some additional modification to the baseline engines like a reduction of compression ratio or reduction of oil consumption.

3.4 Hydrogen diffusive combustion with pilot diesel injection

AVL published a paper that summarizes the initial test results on a truck engine measured with a high-pressure hydrogen direct injection and diesel pilot injection [13]. Figure 11 shows an example of the cylinder pressures and rate of heat releases measured with different hydrogen injection pressures at IMEP of 21 bar and 1200 rpm. A high injection pressure of hydrogen enables a compact and efficient combustion.

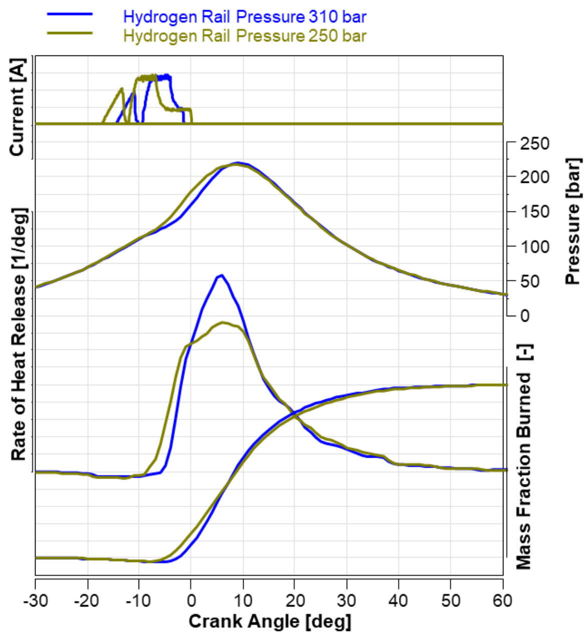


Figure 11. Examples of cylinder pressure and rate of heat release of the hydrogen diffusive combustion ignited by a diesel pilot injection measured at an IMEP of 21 bar and 1200 rpm

A substitution rate of 97.5% (energy based) at a BMEP of 26 bar was achieved. As long as the phasing of the hydrogen injection and pilot diesel injection is in a proper range, there was no abnormal combustion observed. The compression ratio can be increased without a concern of knocking combustions or pre-ignitions and a brake thermal efficiency of above 50% was successfully demonstrated.

The reliability and durability of the injector, however, was not fully proven at the time of the paper submission. One must also note that a compression of hydrogen to a pressure level required for reasonable diffusive combustions is energy intensive and deteriorates the efficiency of a whole system significantly.

3.5 Methanol pre-mixed combustion with spark ignition

Pre-mixed methanol combustion can be realized with both the OCSI and PCSI concepts. The fuel is to be injected in liquid form right upstream the intake valves. LPDI may also be an option but the installation in the cylinder head increases the engineering effort significantly. Injection, evaporation and homogenization of methanol with air are important development topics. Providing a fine spray is key for a good evaporation and homogenization of the fuel/air mixture leading to a reduction of unburned methanol emissions. Currently, however, there is not yet fully proven FIE solution available for methanol application for field engines. One of the notable characteristics of methanol combustion is the formation of formaldehyde emission, which may require an application of exhaust aftertreatment depending on the relevant emission regulations.

If the baseline gas engine uses the active pre-chamber, it would increase the engine complexity to accommodate methanol combustion as it would require either an additional methanol injection system for the pre-chamber or an additional fuel supply system to enrich the pre-chamber by natural gas or hydrogen.

Specific measurements on the SCE are not yet performed by authors for this configuration based on the assumption that the land-based spark-ignited engines would prefer hydrogen or synthetic methane, and methanol is more demanded by marine applications where the baseline engines are diesel or dual fuel engines.

3.6 Methanol pre-mixed combustion with diesel ignition

Premixed methanol-air mixture can be burned also on both diesel and DF engines by injecting the methanol into the intake port upstream of the intake valves. Depending on the compression ratio of the baseline engine and on the performance of the diesel injector at low injection quantities, a substitution rate of 90 to 95% is feasible. As described in the subsection 3.2, spray evaporation and mixture formation are ones of the key development items. The reliability and durability of such methanol injector, however, is still under development.

Authors performed some initial tests using an SCE260-based single cylinder test engine to characterize the pre-mixed methanol combustion. The baseline power unit was a conventional diesel engine with one main common rail diesel injector. A prototype methanol injector was installed in the intake port and liquid methanol was injected at a pressure in the range of 10 to 20 bars.

The engine performances over the entire engine load and the methanol substitution rate that could be measured without sever knocking combustions are shown in the Figure 12, 13 and 14.

As the substitution rate was increased at high load points above 50% engine load, the combustion became instable and knocking combustions started happening stochastically, which prevented a further increase of the methanol energy fraction. Such a limit is depicted in the Figure 12. The baseline engine had a typical compression ratio of a conventional diesel engine, which must have been one of the reasons for the observed knock limit. Another suspected cause of the stochastic knock events is the cycle-to-cycle fluctuation of the methanol fuel quantity in the combustion chamber due to the increased wall film in the intake port. When the methanol injection quantity is increased at high engine load and high substitution rate, it is suspected that the wall film on the surface of the intake port could also increase resulting in a non-timely delivery of the fuel to the combustion chamber and some cycle could receive too much methanol at once causing knocking combustion. Detailed CFD analysis of the methanol spray targeting and atomization as well as the optimization of evaporation and mixing processes would help to minimize the wall film and to improve the stability of the engine operation.

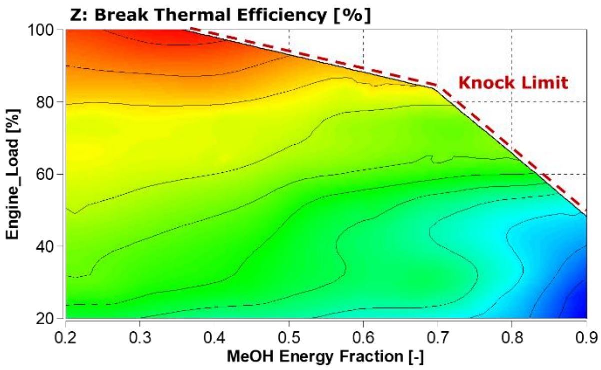


Figure 12. Brake thermal efficiency of the diesel-ignited pre-mixed methanol combustion measured on the SCE

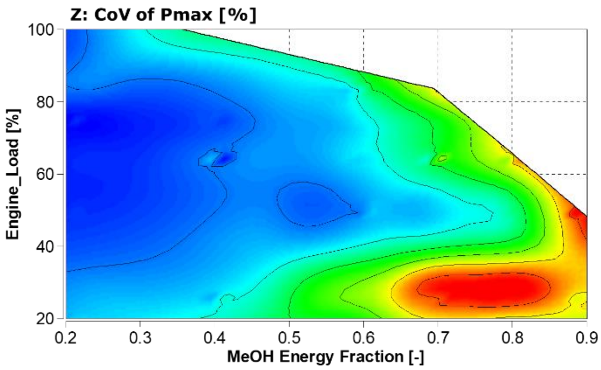


Figure 13. Variation coefficient of the peak firing pressure of the diesel-ignited pre-mixed methanol combustion measured on the SCE

At low load points below 50% engine output, 90% (energy base) of the diesel fuel could be substituted by the pre-mixed methanol without severe knocking combustions. Even at these low load points, however, the cylinder pressure oscillation with a high frequency grew significantly as the methanol energy fraction was increased to 0.7 to 0.8 as shown in the Figure 13. This high frequency cylinder pressure oscillation initiates right after the ignition of diesel fuel and therefore, considered to be an acoustic oscillation caused by a rapid rise of the cylinder pressure due to simultaneous combustion of the pre-mixed methanol and the pilot diesel fuel. As the substitution rate approached 90%, the cylinder pressure oscillation calmed down a little because the pressure rise at the combustion start became somewhat more moderate due to the decreased ignition energy from the pilot diesel. This kind of phenomenon is also known in the natural gas/diesel dual fuel engines.

Another challenge observed was the high levels of unburned methanol emissions and formaldehyde emissions. Figure 14 indicates that the unburned methanol emission increases almost proportionally to the methanol energy fraction. The value in the figure reaches some thousands of ppm in the highest (red) area. Increasing the load to above 50% engine output shows a positive influence to reduce the unburned emissions. The characteristics of the formaldehyde emission is quite similar to that of the unburned methanol emission and increases almost proportionally to the methanol energy fraction and decreases to some extent with the increase of the engine load reaching some hundreds of ppm in the highest area.

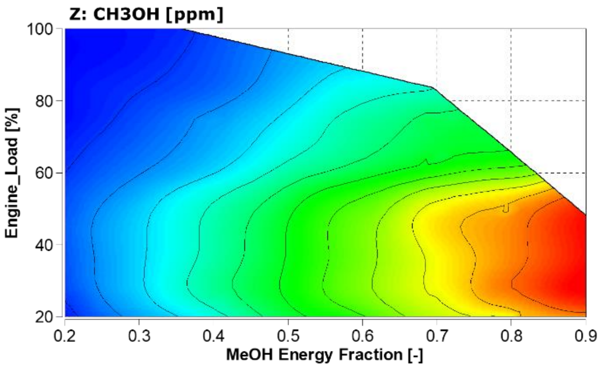


Figure 14. Unburned methanol emission of the diesel-ignited pre-mixed methanol combustion measured on the SCE

An optimization of the mixture formation including targeting, atomization and evaporation of the methanol spray as well as charge motions, intake air and intake manifold temperatures may have a chance to improve the unburned emissions and formaldehyde emissions. Nevertheless, depending on the relevant emission regulation, exhaust aftertreatment system would be still required.

3.7 Methanol diffusive combustion with pilot diesel injection

High pressure direct injection of methanol was tested by the authors on an SCE260-based single cylinder test engine to characterize the diffusive combustion of methanol and some of the main takeaways will be introduced in this subsection.

The baseline power unit was a dual fuel engine with one main common rail diesel injector and one pilot diesel injector. The main diesel injector was replaced by a prototype methanol injector. The liquid methanol was injected directly into the combustion chamber at a pressure in the range of 600 to 800 bars. A small quantity of diesel was injected to provide an ignition energy to the methanol spray at a pressure around 800 bar. The energy fraction of the pilot diesel was in the range of 3 to 5%. Despite the quite low diesel energy fraction, the combustion was quite stable in the wide range of the engine load and air fuel ratio. The engine performances over the entire engine load and the air fuel ratio are shown in the Figure 15, 16 and 17.

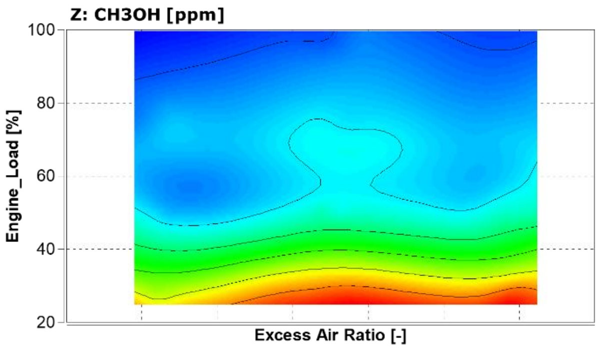


Figure 15. Unburned methanol emission of the methanol diffusive combustion with pilot diesel injection measured on the SCE

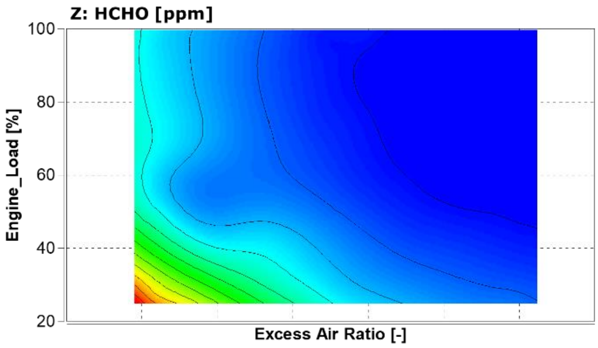


Figure 16. Formaldehyde emission of the methanol diffusive combustion with pilot diesel injection measured on the SCE

As reported also in the literatures [14] and [15], one of the most significant advantages of the high-pressure direct injection measured on the single cylinder engine is the drastic reduction of the unburned methanol and the formaldehyde emissions compared to the pre-mixed methanol combustion as depicted by the Figures 15 and 16, respectively. Unburned methanol emission is quite low in a wide range of the engine load and almost independent of the air fuel ratio. It increases only slightly at low load points but reaches only a few hundreds of ppm. Formaldehyde emissions show a dependency on the engine load and the air fuel ratio and increases slightly when the engine load and the air fuel ratio are decreased. Nevertheless, the value reaches only some tens of ppm in the highest area and otherwise, only a single digit of ppm.

Due to the nature of the diffusive combustion, however, the NOx emissions are high and comparable to that of conventional diesel combustion, which means that SCR or EGR must be considered to achieve stringent emission legislation limits e.g. EPA Tier 4 or IMO Tier 3.

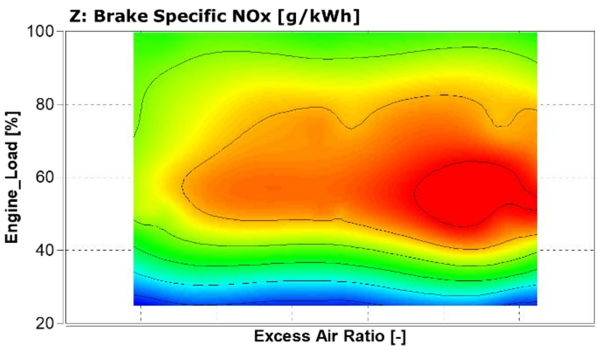


Figure 17. NOx emission of the methanol diffusive combustion with pilot diesel injection measured on the SCE

The handling of the methanol and characteristics of the diffusive combustion is among the other alternatives the closest to the conventional diesel fuel and it may be the most realistic candidate for the diffusive combustion concept. The test results shown here was based on the dual fuel engine configuration and the main injector was capable only of injecting methanol meaning that the fuel redundancy is not fully given. When the reliability of the high-pressure methanol fuel system is established and the availability of the fuel itself is assured, the tested configuration may be a promising candidate for marine applications enabling a utilization of the high methanol energy fraction. Otherwise, when the fuel redundancy is a must, an injector that is multi-fuel capable would be required. In this case, the additional pilot injector would not be necessarily required, and the baseline engine can be a conventional diesel engine.

4 CONCLUSIONS

Given the growing interest in switching from fossil fuels to a wide range of carbon-neutral alternatives, this paper has discussed different conceivable combustion concepts and expected applications of hydrogen and methanol. The combustion properties and specific challenges of each alternative fuel were outlined for the selected combustion concepts.

By adding or adapting the fuel supply system in the intake port, it is feasible, in a relatively simple manner, to burn alternative fuels with pre-mixed combustion concepts. However, the premixed combustion typically suffers from unburned emissions and abnormal combustions such as knocking combustion or pre-ignition. The diffusive combustion concept generally offers an opportunity to solve such issues but poses the challenge of increased engine complexity with two high-pressure fuel injection systems and faces also the challenges associated with the availability and reliability of the fuel systems.

Hydrogen can be seen as a promising energy carrier that could fill the gap between the power demand and power generation timely and locally. Especially, it could compensate the fluctuating nature of the power generation by renewable power sources. Any further fuel conversion for instance from hydrogen to methanol would consume additional energy, which reduces the total efficiency of the whole process from the storage of excessive power to the regeneration. From this perspective, it seems to be a reasonable choice to burn hydrogen directly on the engines.

Based on the measurement results on single cylinder test engines, it is feasible to burn pre-mixed hydrogen both on spark-ignited engines and on diesel-ignited engines. A substitution rate of 90% or higher could be achieved at least at part load in both cases. The suppression of combustion anomalies is essential for the use of hydrogen in pre-mixed combustions. The diffusive combustion of hydrogen demonstrated on a truck engine indicated that the combustion anomaly could be solved and a high substitution rate at high engine load is possible but a further development to prove the reliability and durability of the hydrogen injector is mandatory.

Methanol on the other hand can be seen as a reasonable candidate as an alternative fuel for marine applications due to its higher volumetric energy density compared to the gaseous hydrogen. Furthermore, the fuel redundancy could be highly appreciated in case of marine applications. From these perspectives, dual-fuel engines either with a diesel-ignited pre-mixed methanol combustion concept or a diesel-ignited diffusive methanol combustion concept may be preferred.

The single cylinder engine test results revealed that the substitution rate of 90% is possible at part load with the pre-mixed methanol combustion. However, a further optimization, especially, of the methanol injection and mixture formation is necessary to solve the knocking issues and realize a high substitution rate at high engine load points. The other challenges are high levels of the unburned methanol and the formaldehyde emissions.

A good performance of the diffusive combustion of methanol was demonstrated by the single cylinder test engine. A substitution rate of 95% and higher is feasible when the injection stability at the low injection quantity is given by the diesel injector. A significant advantage of the diffusive combustion is the quite low levels of unburned methanol and formaldehyde emissions. The two high pressure fuel systems would increase the engine complexity, but the diesel-ignited diffusive methanol combustion concept seems to be one of the promising candidates for the prime movers in the marine application. The maturity of the methanol injector in terms of reliability and durability are yet to be proven.

Finally, it must be added that even though the present paper described the feasibility of the utilization of alternative fuels on the engine side in detail such as combustion characteristics and performances, it was not considered as a focus of this paper to discuss the availability and economical feasibility of the alternative fuel itself as well as their infrastructures. When the hydrogen, for example, is not produced in a CO_2 neutral manner, neither the hydrogen combustion engines nor the fuel cells can be CO_2 neutral prime movers. Likewise, the methanol must be produced with the green hydrogen and CO_2 captured from the atmosphere using the renewable-based electricity. In order to face such a grand challenge, all industries in each country must align and work together toward a common goal.

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