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Hydrogen-based ship propulsion; first ever large two-stroke engine tests with hydrogen

Fuels - Alternative & New Fuels

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ABSTRACT

In order to meet global greenhouse gas (GHG) reduction ambitions, it is critical to focus on decarbonizing the shipping sector since shipping accounts for 3% of the total global GHG emissions. In the marine sector, decarbonization can be achieved by carbon-free fuels (such as hydrogen or ammonia), hydrocarbon fuels produced from renewable biological sources or fuels synthesized from H2 and CO2 (like e-methanol). The work presented in this paper focuses on the simplest of these fuels, hydrogen (H2) and its applicability to an MAN B&W large two-stroke marine diesel engine. Large two-stroke diesel engines are currently the de-facto standard for propulsion of large merchant ships due to their high efficiency and proven robustness. Hydrogen, as a carbon-free fuel for these engines is thus a significant and valuable target to strive for.

In the work presented here, the Mitsui E&S test engine, the four-cylinder and 50cm bore two-stroke 4S50ME-T diesel engine, was rebuilt to operate one cylinder out of the four on hydrogen. This allowed a high degree of testing flexibility, since the remaining three cylinders were operated on diesel, while also keeping the hydrogen consumption at an acceptable level.

The work was done in close collaboration between MAN-ES and Mitsui E&S. Mitsui's main focus was on the supply system side while MAN-ES focused on engine alterations and data evaluation.

The results for these Hydrogen tests indicated that hydrogen can be used as fuel as long as a pilot fuel injection is utilized for stabilizing the ignition around TDC. This is very similar to methane, and the same basic hardware, the ME-GI system, can also be utilized for both fuel types, as was shown here.

Combustion stability, ignition timing and combustion efficiency were evaluated during the tests and the results thereof will be presented in the paper. In general, the results showed a good performance for hydrogen combustion in a two-stroke marine diesel engine.

Experiments with premixed hydrogen combustion were also made. In these, the gas was injected into the cylinder well before the pilot fuel injection and was given time to premix with the charge air in the cylinder. Note that the engine configuration of a large two-stroke engine does not allow for gas mixing outside the cylinder, in the scavenge box, due to safety concerns.. The tests with premixed hydrogen showed that it is difficult to prevent ignition of the mixture prior to the pilot injection. Given the low revolution speed of the two-stroke engine, this results in an uncontrollable and unsafe combustion process.

Work was also made in attempts of optimizing and further understanding the hydrogen combustion process. Results indicate a similar SFOC and NOx response to the well-known diesel process.

1 INTRODUCTION

Climate change, driven by the emission of greenhouse gases (GHGs), predominantly CO_2, is one of the largest challenges humanity has ever faced. The expected continuous increase in global energy demand urgently necessitates the development of climate-neutral energy production technologies. The responsibility for this largely falls on the industry.

The shipping industry, one of the largest individual contributors to GHG emissions, currently accounts for around 3% of global GHG emissions [1][2]. The International Maritime Organization (IMO) has recently set a target to achieve net-zero GHG emissions by 2050 [3]. This has further intensified the push for a climate-neutral shipping industry.

Large vessels, such as container ships, bulk carriers, and oil and gas carriers, currently employ large two-stroke diesel engines for propulsion due to their robustness, high efficiency, fuel flexibility, and capability to couple directly to the propeller. These engines predominantly operate on various grades of petroleum oil, for example, 0.5%S VLSFO or 0.1%S ULSFO. However, over the last decade, engines running on new types of marine fuels like liquified natural gas (LNG) and methanol have been developed [4][5][6]. While these fuels can have a lower greenhouse footprint, achieving CO_2 neutrality for large-scale oceanic transportation will only be possible with the transition to completely carbon-free fuels like ammonia and hydrogen, or by using yet-to-be-fully-developed carbon capture and storage technologies [7].

The purpose of the tests presented in this paper was to determine whether hydrogen could be used as fuel in a large two-stroke marine diesel engine. Additionally, trends in operating conditions and emissions were to be determined and evaluated. Hydrogen has previously been used in smaller engines [8][9] and its global use as an energy carrier is expected to increase significantly in the future [10]. However, to the best of our knowledge, the work presented in this paper represents the first attempt to apply hydrogen to a two-stroke marine diesel engine [11].

1.1 Hydrogen compared to methane

The fact that hydrogen does not contain carbon, and therefore does not generate CO_2 when combusted, is the main reason it is so attractive as a fuel. However, several challenges make this concept more difficult than other fuels. Hydrogen is difficult to store, has a low flashpoint, generates high compression losses, can be hard to ignite using compression ignition, causes knocking easily (especially when premixed), can generate high NO_X emission levels, and can cause brittleness in many alloys [12].

However, in the context of high-pressure directly injected fuel, hydrogen performs quite similarly to methane [12]. Comparisons of relevant properties are shown in Table 1. The most important aspects of direct injection combustion are the injected energy amount per unit of time and the fuel jets' momentum when exiting the injector, which largely governs the mixing rate with the surrounding air in the combustion chamber.

Table 1. Properties of hydrogen and methane.

Characteristics	Methane	Hydrogen
Chemical formula	CH_4	H_2
Carbon (w%)	75%	0%
Critical temperature	-82.6°C	-240°C
Critical pressure	46 bar	13 bar
Lower calorific value (LCV)	50 MJ/kg	120 MJ/kg
Boiling temperature @5 atm	-137°C	-246°C
Density @15°C, 1 atm	0.68 kg/m^3	0.085 kg/m^3
Density @20°C, 700 atm	313 kg/m^3	40 kg/m^3
Air fuel ratio (AFR), stoichiometric	17.2	33.3
Laminar flame speed	46 cm/s	300 cm/s
Autoignition temperature	537°C	536°C

Comparing the numbers for hydrogen and methane (see Table 1), they might at first glance appear quite different. Hydrogen's lower calorific value (LCV) is 120 MJ/kg compared to 50 MJ/kg for methane, while the density at injection-relevant pressures is a factor of eight smaller for hydrogen. Additionally, the speed of sound is significantly higher, allowing the maximum jet speed, which is limited by the choked flow through in the injector holes, to reach higher velocities during the injection of hydrogen.

An analysis of these differences between methane and hydrogen shows that the injection system for hydrogen can be quite similar to that for methane. The modelled injector hole size for hydrogen turns out to be just 20% larger, while the modelled injection pressure is roughly 43% lower. Therefore, it is feasible to use an injection system designed for methane for hydrogen injection. It should be noted that the targeted higher injection pressure in the methane system compensates, to a certain degree, for the larger injection hole requirement for hydrogen, as it increases both the density and the flow speed during injection.

Additionally, it is possible to adjust the injector hole size to acquire the optimal injection duration and jet momentum. However, there are limitations to this as the upstream fuel system is scaled according to the required mass and volume flow through the injection holes. This means that significant deviations in the size of the holes from their original dimensions will require a resizing of the entire fuel supply and injection system.

2 METHODOLOGY

The reasoning in Section 1.1 led to the conclusion that a ME-GI MAN B&W two-stroke dual-fuel engine, originally built for operation on compressed natural gas (CNG), could potentially be modified for hydrogen operation by fairly moderate adjustments. These modifications were undertaken to discern their feasibility and to identify potential issues. The project was a collaboration between MAN Energy Solutions and Mitsui ES. This report presents the engine test results of operating the engine on hydrogen with a small diesel pilot to ensure stable ignition. Additional results regarding the hydrogen supply system and other aspects are presented elsewhere.

Figure 1. Picture of the hydrogen supply system showing the liquid hydrogen storage tank, vaporisers, and compressors.

2.1 Supply system

Hydrogen is commonly stored cryogenically, i.e., as a liquid at very low temperatures (see Table 1), to maintain a fairly adequate energy density. Hydrogen's extremely low storage temperature, and its tendency to creep into metals such as steel, which causes embrittlement [13], makes storage more difficult and expensive compared to methane. This necessitated a dedicated supply system for these tests.

The hydrogen supply system used in these tests is shown in Figure 1. The picture shows the large hydrogen storage tank, the vaporisers, and the several stages of compressors utilised in these tests. More details about the hydrogen supply system can be found in the paper from Ishibashi et al. at MES, which was published simultaneously with this paper.

The output pressure and temperature of the supply system were 300 bar and 50°C at high load to match the common target conditions for the ME-GI engine methane supply and injection system.

2.2 Hydrogen engine platform

The 4S50ME-T test engine at Mitsui Engineering and Shipbuilding (MES) in Tamano, Japan, was selected as a platform for these tests. The engine specifications are shown in Table 2. This test engine has previously been modified for dual-fuel operation using methanol [6] but has not previously been used for gas operation using methane (ME-GI or ME-GA).

Table 2. Engine specifications for the 4S50ME-T9 test engine at MES.

4S50ME-T	A
Engine type	Two-stroke, uniflow scavenging, crosshead direct injection
Fuel type	Diesel and hydrogen
No. cylinders	4
Bore	0.5 m
Stroke	2.214 m
Connection rod	2.214 m
Compression volume	20.3–19.8 liter
Power rating	7,600 kW
Speed rating	125 rpm
Max. cylinder pressure	200 bar
Hydrogen injection pressure setpoint	250 – 300 bar

In order to safely test hydrogen operation on the 4S50ME-T test engine, it was decided to utilise only one of the four cylinders (cylinder No. four) for hydrogen testing. The testing cylinder was equipped with an ME-GI Mk. II system, which includes a gas control block, two high-pressure gas injection valves (GIV), and gas supply and return piping, amongst other components.

Operating on only one cylinder has several benefits and implications when testing a new fuel. Firstly, the engine operating stability is greatly improved since the other three cylinders continue to operate on diesel. If ignition issues occur, the engine will continue to run without major concerns. Secondly, the number of components that can cause leakages is minimised, thus reducing the risk and making leak detection easier and safer. It also reduces the volume/mass of hydrogen present in the system, significantly lowering the risk associated with a fire on the engine or surrounding systems.

Several additional safety features, not commonly present on an ME-GI engine, were added to these tests to minimise the risks associated with hydrogen operation. Three major changes were:

1. Continuous crankcase supervision and ventilation were added, with the possibility of nitrogen purging.

2. Airflow in the double-walled piping was replaced with pure nitrogen, further reducing the risk of fire during a potential leakage.

3. Hydrogen sensors were added at several points in the surrounding building.

Test procedures were also revised and strictly adhered to, minimising the risk for personnel and machinery.

2.3 Test procedure

Hydrogen was combusted in only one of the four cylinders. This approach improved safety and reduced hydrogen consumption, resulting in significant cost savings. However, it also had implications for the testing procedure and measurement setup.

To measure the exhaust gas composition from the hydrogen-powered cylinder, it was necessary to perform the measurement before the exhaust gas from the individual cylinders were mixed in the exhaust receiver. This was achieved by sampling the exhaust gases right after the exhaust valve. A probe was inserted into the exhaust duct and connected to a fast sampling valve, which was timed to open only when the exhaust valve was open. This opening timing coincided with a positive exhaust flow through the exhaust duct caused by the positive pressure differential between the scavenge receiver and the exhaust receiver. This ensured that only gas from the hydrogen combustion was sampled. Great care was taken to avoid cross-talk from the other cylinders and to acquire an accurate sampling of the exhaust gases from the hydrogen combustion, as the gas composition varies significantly while the exhaust valve is open.

The sampled exhaust gases were then collected in a mini receiver (see Figure 2), which was kept at a fixed low pressure using a dedicated pump to quench any chemical reactions still ongoing in the exhaust. The receiver was heavily insulated and temperature-regulated to avoid condensation of, e.g., hydrocarbons. Sampling from the mini receiver to the gas composition analysers was made via a separate outlet.

Figure 2. Picture of the mini receiver that was used to sample exhaust gas from the cylinder running on hydrogen. The sample was extracted right after the exhaust valve as shown in the top left of the picture.

Exhaust samples were also collected from the total exhaust flow out of the turbine, i.e. the mixture from all four cylinders corresponding to common exhaust sampling practises. Diesel references, where all four cylinders were operated on diesel, were used to verify that all sampling techniques provided similar and representative results.

The exhaust samples were analysed using both an IMO standard exhaust analysis system, measuring CO_2, O_2, CO, THC, and NO_X, and an FTIR for additional measurements of, e.g., N_2O and H_2O. Hydrogen was measured using a thermal conductivity sensor [14].

Most of the tests were performed with a normal engine layout profile with the same MIP, P_{comp}, and P_{max} for all four cylinders. However, to significantly vary the airflow through cylinder four at a given engine power, it was also possible to operate the three diesel cylinders with a different MEP or PcPs compared to the hydrogen cylinder. This allowed for more extensive testing of various effects. However, the trapped charge in cylinder four had to be modelled using in-house code to be estimated accurately, since the airflow through the different cylinders is not the same when the engine is deliberately unbalanced.

3 RESULTS

3.1 First hydrogen test

The results from the very first hydrogen test are presented in Figure 3. The initial test was conducted with surprisingly little effort once the components were correctly mounted, and the communication between the supply system and the new engine software was properly synchronised. The ME-GI system was first set to activate a small hydrogen injection, which was then gradually increased while tuning of the injection timing (SOI), pilot offset timing, and diesel pilot amount was made to achieve the desired hydrogen combustion properties.

The test labelled T24_06, presented in this section, was recorded at 25% engine load (8.1 bar MEP) right after the pilot tuning was completed. The pilot flow was ~5% of the diesel consumption at full load.

The peak heat release, see Figure 3(a), is slightly higher for hydrogen (red curve) compared to diesel. This indicates a higher rate of combustion due to a relatively higher injection rate (in energy, not mass). Because of the engine tuning, where the pressure rise (P_{rise}) is kept fixed at 40 bar, the start of injection (SOI) is delayed, which gives a delayed start of combustion (SOC) as is seen in the heat release (HR) trace for hydrogen (red curve). Ultimately, the combustion duration of hydrogen is shorter with a significantly faster end of combustion, which is consistent with a less pronounced soot chemistry (a small amount of diesel pilot is still present.).

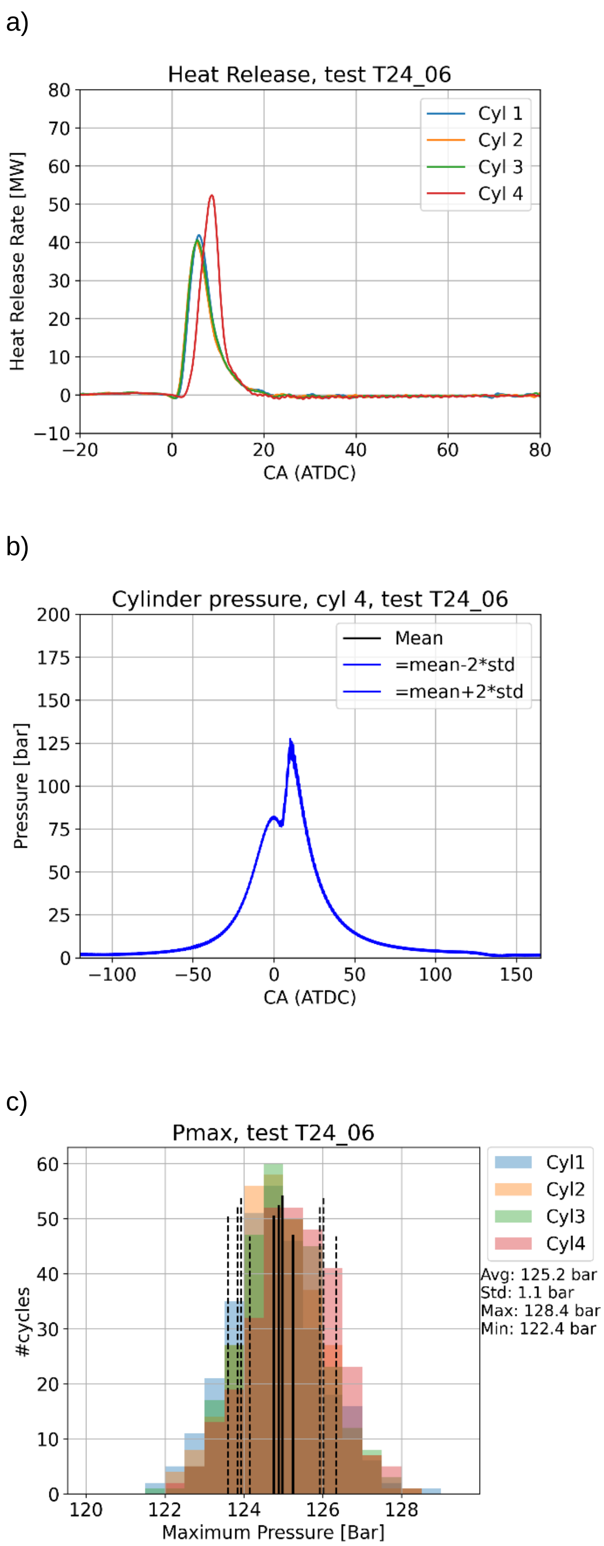


Figure 3. First hydrogen test results: (a) apparent heat release (HR) for all four cylinders, cylinder four (red) is operated on hydrogen, (b) accumulated cylinder pressures for cylinder four, (c) histogram of maximum firing pressure (P_{max}) for all four cylinders.

The accumulated cylinder pressure traces (300 revolutions) in Figure 3(b) shows good stability for hydrogen with low cycle-to-cycle variations. There are no indications of unstable ignition or unstable combustion, e.g. diesel knocking.

The histogram of the maximum cylinder pressure for each cycle in (c) also highlights the combustion stability. There is no significant difference between diesel and hydrogen histograms, indicating a similar process stability for the two fuels. Since the diesel combustion stability of a two-stroke engine is known to be excellent and with a very robust process, it is rather fair to assume that the same applies to the hydrogen combustion process.

3.2 Hydrogen nozzle variations

To further test and verify the hydrogen combustion properties, the geometrical injection pattern of the hydrogen nozzle was varied. This is commonly done for diesel engines to optimise the combustion process, aiming for the lowest possible fuel consumption while keeping NO_X emissions below the legislative limits.

The HR at 75% engine load for three hydrogen nozzles is shown in Figure 4. The nozzle details are listed in Table 3.

Table 3. Hydrogen nozzle specifications. The total hole area is for two injection valves per cylinder.

Nozzle	No. holes	Hole diameter	Total hole area
H-10	5	2.0 mm	31.4 mm^2
H-12	4	2.2 mm	30.4 mm^2
H-15	5	2.2 mm	38.0 mm^2

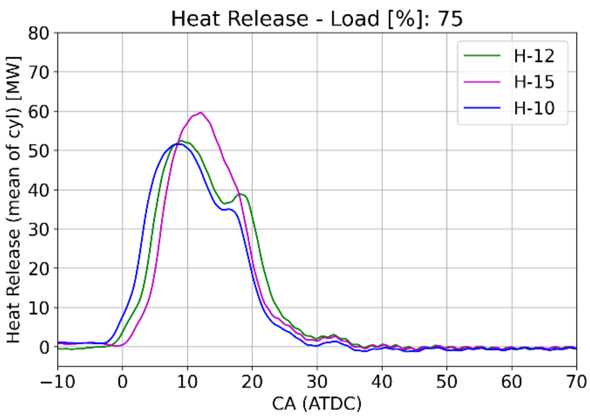


Figure 4. HR for various hydrogen nozzles at 75% engine load.

Nozzle H-15, which has a significantly larger hole area, produces a higher peak HR compared with the other two nozzles. Additionally, H-15 does not show the second peak characteristic for jet combustion in large two-stroke engines at higher loads where jet-to-jet interactions are more prominent. This is consistent with experiences from the combustion of other fuels, including diesel, in two-stroke engines. Hydrogen thus shows the same trends when varying the nozzle hole size, and the same general considerations therefore apply.

Comparing H-10 with H12, it is clear that roughly the same HR can be achieved with both four-hole and five-hole nozzles. The difference in SOI is related to slightly different P_{max} values for these tests and is therefore not an effect of the nozzle itself. This conclusion is somewhat surprising, as changing the number of holes in a diesel nozzle usually has a substantial impact on the HR. A hypothesis is that the higher laminar flame speed, coupled with the higher AFR for hydrogen, helps to reduce localised effects on the jet combustion process, thus reducing differences in the apparent HR.

3.3 Variations in compression volume

To illustrate the effects on emissions from engine parameter variations when burning hydrogen, two different cases with different compression volumes are compared in this section.

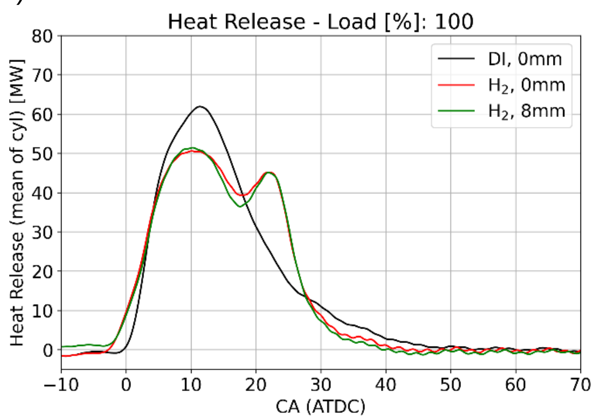
The first case has the largest possible combustion chamber for this engine configuration, achieved by removing all shims under the piston, i.e. 0 mm shims. This results in a combustion chamber volume at TDC of 21.35 liter (at cold conditions). The second case is with 8 mm shims, resulting in a smaller combustion chamber at TDC of 19.78 liter. The hydrogen tests were performed using the H-10 nozzle (see Table 3). The tests are labelled according to shim height, i.e. “0 mm” and “8 mm”, in Figure 5 to Figure 9. The figures show values for both hydrogen and diesel to illustrate the relative differences in each case.

Note that each measurement at a given load point used the same target in P_{rise} and PcPs. Differences in scavenge air pressure were within 0.1 bar, resulting in differences in compression pressure (P_{comp}) and maximum firing pressure (P_{max}) within 1 bar at each load point when comparing all the test series.

The HR curves in Figure 5 show that shim height does not noticeably affect both the hydrogen and the diesel combustion rates. This is expected given that hydrogen is combusted using the diesel principle with a small pilot for stable ignition. The hydrogen HR is slower and longer compared to diesel. This can be mitigated by increasing the nozzle hole size, as shown in Figure 4. However, the optimal balance in nozzle hole size is commonly achieved when SOI is slightly before top dead centre (TDC) at 100% load, which is the case here for hydrogen. This is also clearly seen in the indicated efficiency shown in Figure 6, where the mean indicated efficiency for hydrogen is higher than that for diesel. The indicated efficiency (defined as η = W/Qin = IMEP/QMEP) is based on the heat release calculations as shown in Figure 5. It should be noted that the HR calculations do not take variations in gas composition into consideration.

The higher shims result in a higher indicated efficiency, because of the smaller compression volume, leading to a longer effective Miller expansion. The specific fuel oil consumption is not shown here due to technical uncertainties in measuring the hydrogen consumption accurately enough when using hydrogen on only one cylinder. Very minor changes in efficiency on the other three cylinders can easily overpower or hide any effects from the hydrogen combustion.

a)



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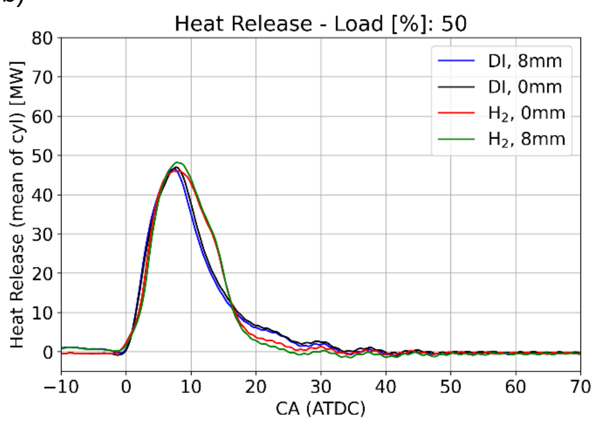


Figure 5. HR from cylinder four at (a) 100%, and (b) 50% engine load for diesel (DI) and hydrogen (H_2) for both 0 mm and 8 mm shims.

As mentioned concerning the HR calculations in Figure 5, there is a significant difference in exhaust gas composition between the two fuels. This is illustrated in Figure 7, where water (H_2O) emissions are plotted on the y-axis against the remaining oxygen (O_2) in the exhaust on the x-axis. The gases are sampled directly from cylinder four using the mini receiver as described in Section 2.3.

In these tests, the water content in the exhaust for hydrogen operation ranges between 6–10% while the oxygen is between 15.4–17.3%. In contrast, diesel, with a hydrogen fraction of 13.2%, produces only 3–4% water and consumes more oxygen due to the additional production of CO_2. The differences in air amount in cylinder four (both trapped and total) are small, especially when compared to the differences in chemical composition.

These differences in exhaust compositions need to be considered when designing a hydrogen engine as they can affect both the selection of the turbocharger and the engine performance parameters that will generate the lowest fuel consumption.

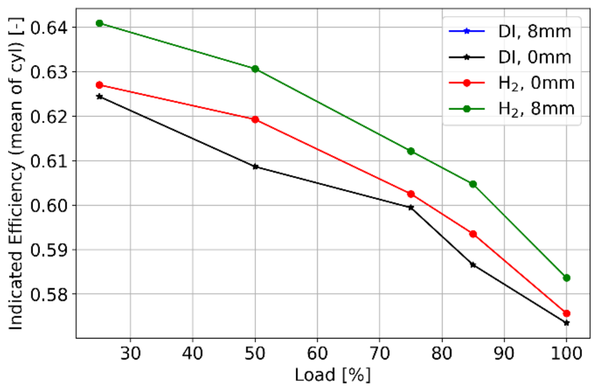


Figure 6. Indicated efficiency for hydrogen. Averages are calculated from 300 engine cycles.

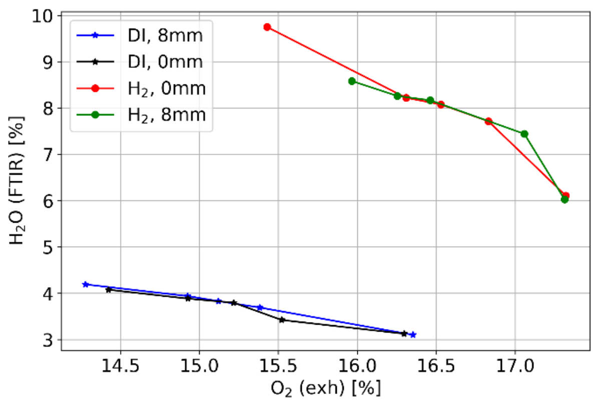


Figure 7. Emissions from cylinder four with water (H_2O) on the y-axis and remaining oxygen (O_2) on the x-axis.

3.3.1 Emissions – minor species

Hydrogen has a higher flame temperature compared to diesel, resulting in increased thermal NO_X production. This is clearly shown in Figure 8, where NO_X emissions are 300–600 ppm higher for hydrogen compared to the corresponding diesel case. Some load dependencies contribute to this range, and some measurement uncertainties are included.

The increase in NO_X emissions when removing the shims is similar when comparing the hydrogen curves with the diesel curves. This suggests that the NO_X formation mechanisms are similar, despite the vastly different flame chemistry for the two fuels. Therefore, it is likely that the same well-known engine NO_X tuning methods can be used for hydrogen.

However, the very high NO_X emissions make it unlikely that an two-stroke engine operating fully on hydrogen will do so without active NO_X reduction technologies, such as selective catalytic reduction (SCR) or exhaust gas recirculation (EGR). This applies to both Tier II and Tier III zones.

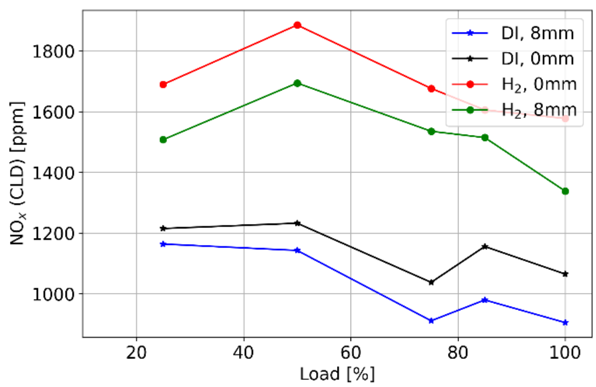


Figure 8. NO_X emissions from cylinder four (in ppm) as a function of engine load for hydrogen and diesel using both 0 mm and 8 mm shims.

Three additional minor emission species are shown in Figure 9: (a) THC (total hydrocarbon emissions), (b) laughing gas (N_2O) emissions, and (c) hydrogen (H_2) emissions/slip. These figures show several interesting aspects of the tests that are not yet fully explained.

Operating one cylinder on hydrogen appears to increase the total hydrocarbon emissions. There is also a larger effect from shims on the THC from hydrogen operation compared to diesel operation, which does not seem to be significantly affected by changing shims. There are three possible sources for the increased THC: 1) the diesel pilot, 2) the lubricating oil, and 3) the sealing oil in the ME-GI system.

Increased diesel injector dribble is likely since the small diesel pilot injection may allow for a higher temperature in the diesel nozzle sac volume during hydrogen operation. The sealing oil consumption probably also contributes significantly to the THC. However, an increase in the THC from the lubricating oil is deemed unlikely based on a more holistic engine analysis.

a)

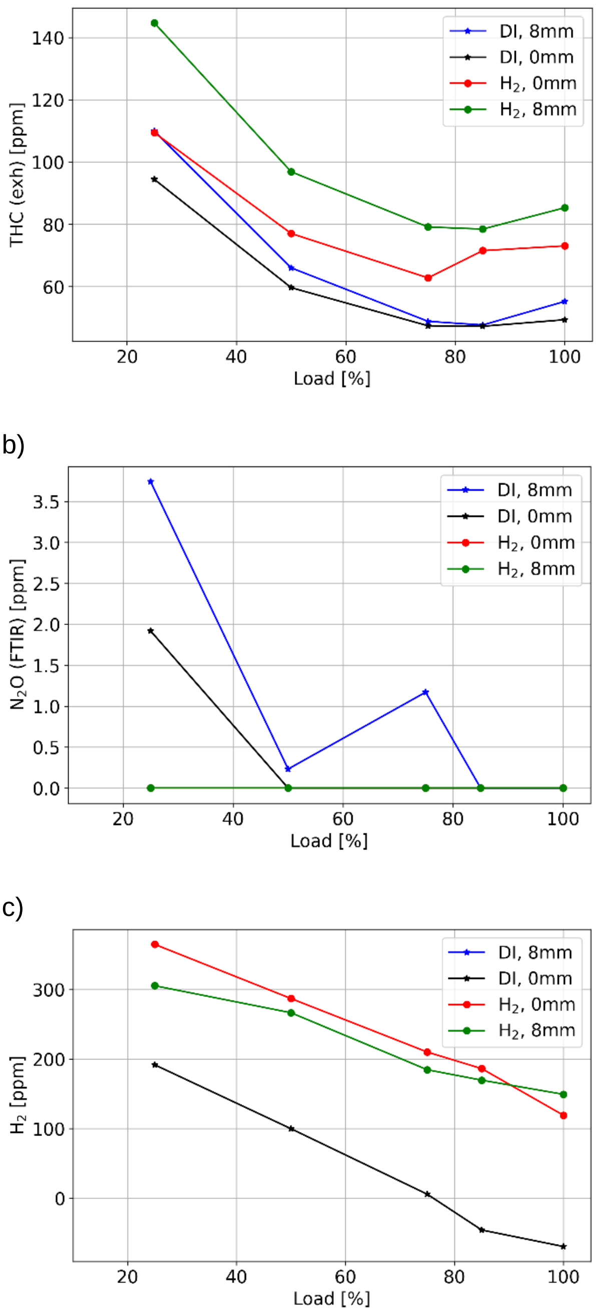


Figure 9. Minor emission species: (a) THC (total hydrocarbon emissions as CH_4), (b) laughing gas (N_2O), and (c) hydrogen (H_2) emissions. All are shown as a function of engine load for hydrogen and diesel, using both 0 mm and 8 mm shims.

These THC results are perhaps not the most interesting regarding the tests presented here, since the measured THC cannot originate from hydrogen. However, for ME-GI engines operated on methane, the results indicate that the common increase in THC when operating on gas is not just due to methane slip, but is perhaps mostly due to sealing and diesel oil.

N_2O emissions are non-existing in Figure 9(b) for the hydrogen combustion. This shows that hydrogen's high adiabatic flame temperature, in combination with the diesel pilot, is not causing increased greenhouse gas effects via N_2O generation in a two-stroke engine. While this result is not particularly surprising, it is reassuring to confirm.

The measured hydrogen slip (c) is around 100-300ppm, with a decreasing trend as a function of load. However, the hydrogen slip is measured using a thermal conductivity sensor that is sensitive to practically every species present in the exhaust gas [14]. Although the greatest care was taken to calibrate the measurements as diligently as possible, it is unavoidable that these results are confounded with other changes in exhaust gas composition. This is clearly seen in the diesel case, which shows up to 200 ppm H_2 emissions, an unrealistic and inaccurate measurement value. Unfortunately, there were no alternative measurement techniques available at the time of testing that could measure hydrogen with greater accuracy in the challenging conditions of the two-stroke engine exhausts.

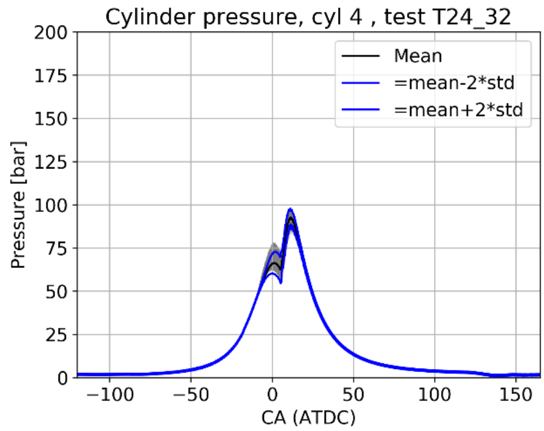
3.4 Premixed hydrogen operation

Previous tests with premixed hydrogen combustion on smaller engines have shown that premixed lean burn of hydrogen is an attractive way to reduce NO_X emissions. Therefore, there was an interest in verifying if this was also the case for a large marine two-stroke engine.

Due to its design with a large common scavenge air box, an MAN B&W two-stroke diesel engine cannot utilise port fuel injection, as this would pose a significant fire hazard. A true Otto mode was therefore not possible. Instead, a substitute was performed by significantly advancing the hydrogen injection from the regular high-pressure gas injectors to well before TDC, allowing for premixing of the hydrogen in the combustion chamber before combustion.

The hydrogen injection was first advanced to before TDC while the diesel pilot was delayed to well after TDC, resulting in a negative overlap of 15–30 CAD. The amount of hydrogen was then increased in the hope of finding a stable operating point where a significant part of the power generated was from hydrogen, while the diesel pilot maintained a stable combustion phasing.

a)



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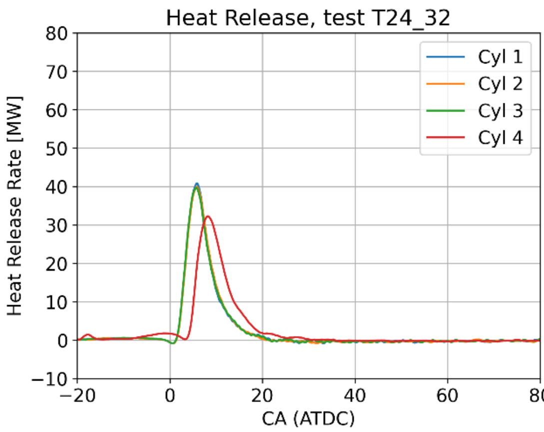


Figure 10. (a) cylinder pressure with an advanced hydrogen injection before TDC, (b) average HR release for the same test on cylinder four.

However, it was quickly observed that the hydrogen ignited well before the diesel pilot injection, resulting in preignition. This is a known issue for smaller premixed hydrogen engines as well [12]. Given the high tendency for preignition, every possible measure was tried to mitigate the preignition tendency in order to stabilise the hydrogen combustion process at timing of the diesel pilot injection.

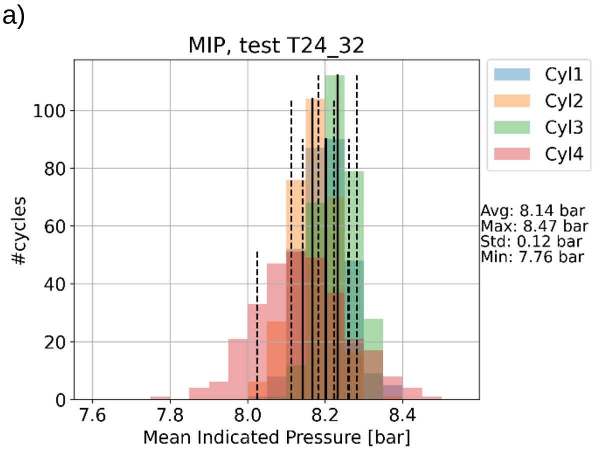
- The SOI of hydrogen was advanced to -27 CAD ATDC, which was as early as possible due to safety concerns.
- The compression ratio was reduced in order to lower the charge air temperature.
- The mean indicated pressure (MIP) was reduced on cylinder four (less diesel

- injected) to reduce the surface temperatures of the combustion chamber.
- Hydrogen supply pressure and temperature were reduced to avoid wall impingement and to further lower the charge gas temperatures.

Finally, the amount of hydrogen was increased as much as was deemed safe. Using this methodology, it was only possible to reach a maximum hydrogen injection of 8% index, which is slightly below the common idling level for this engine.

The resulting cylinder pressures for 300 consecutive engine cycles are shown in Figure 10(a). The greyed zone represents the highest and the lowest cylinder pressure among all the measured cycles. Although there are some cycles without combustion prior to the diesel injection, the majority show a pressure rise, and thus combustion, before TDC which is well before the start of injection (SOI) for the diesel pilot.

a)



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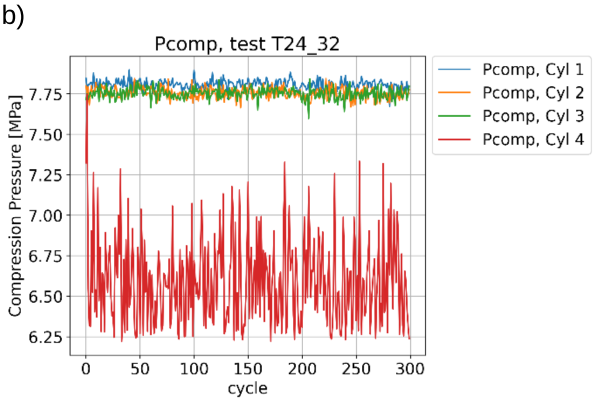

Figure 11. Combustion stability for the premixed hydrogen tests showing a) the MIP histogram and b) the cylinder pressure at TDC for each of the 300 consecutive engine cycles for all four cylinders.

The corresponding apparent HR (average curve for all 300 engine cycles) is shown in Figure 10(b). There are clearly some cycles where the hydrogen combustion starts right after SOI, as indicated by the small HR spike between -20 and -18 CAD ATDC. However, the majority of the cycles show combustion closer to TDC (-9 to +3 CAD ATDC), which is still significantly ahead of the pilot injection.

Preignition is not necessarily an issue as long as the pressure rise is controllable and stable. The combustion stability for this test is shown in Figure 11. (a) shows a histogram of MIP for each of the four cylinders. Cylinder four (with premixed hydrogen) clearly has a wider distribution, indicating that the combustion is more unstable, which will negatively affect the engine control system (ECS). More troubling, however, is the variability in cylinder pressure around TDC (P_{comp}) shown in (b). P_{comp} varies roughly 10 bar between the cycles for this case with only an 8% injection index. This variability is larger than can be accepted by the ECS due to safety concerns and component lifetime considerations.

Based on these results, it is concluded that it is not possible to operate a two-stroke engine with premixed hydrogen using these systems. A full injection of hydrogen, needed to sustain a relevant engine load, would cause excessive variations in cylinder pressure and power output, posing a high risk of component damage. The preignition observed here is significantly stronger than what has been seen for a similar premixed methane combustion in the so-called ME-GA engines. Therefore, it is very unlikely that high levels of EGR can mitigate this situation, although these tests do not specifically rule out this possibility.

4 CONCLUSIONS

The results presented here show that it is possible to replace methane with hydrogen as fuel in an ME-GI dual-fuel marine two-stroke combustion engine.

The parameter variations, such as the nozzle and shim variations, showed the expected trends in emissions and indicated efficiency. Therefore, the hydrogen combustion largely behaves like the methane combustion in a normal ME-GI engine.

The NO_X emissions were, on average, 49% higher for hydrogen compared to diesel. The performance variations observed show the usual trends, indicating that the NO_X emissions are mainly due to thermal NO_X generation, as is the case for diesel combustion. Therefore, it is expected that there is not enough potential for NO_X reduction using engine tuning methods to reach the required Tier II NO_X emission levels, at least not without significantly increasing fuel consumption. Therefore, it is expected that a potential future two-stroke ME-GI engine will require an SCR or EGR system to reduce NO_X emissions while operating on hydrogen.

The tests with pre-injection of hydrogen, mimicking premixed hydrogen operation showed that it is very difficult to run a marine two-stroke engine with a premixed Otto-like engine operating mode when using hydrogen. It was not possible to avoid severe preignition of the hydrogen charge in these tests. Therefore, it can be concluded that the diesel operating mode is the preferred option, even though the NO_X emissions are excessively high.

5 DEFINITIONS, ACRONYMS, ABBREVIATIONS

AFR: Air fuel ratio

CAD ATDC: Crank angle degrees after top dead centre

CNG: Compressed natural gas

DI: Diesel

ECS: Engine control system

EGR: Exhaust gas recirculation

GIV: Gas injection valve

GHG: Greenhouse gas

HR: Heat release (rate)

IMO: International Maritime Organization

LNG: Liquified natural gas

LCV: Lower calorific value

ME-GA: Dual-fuel gas engine with low-pressure injection after exhaust valve closing before TDC

ME-GI: Dual-fuel gas engine with high-pressure injection around TDC

MEP: Mean effective pressure

MES: Mitsui ES

MIP: Mean indicated pressure

NO_X: Nitrous oxide emissions (mainly NO and NO_2)

P_{comp}: Compression pressure (pressure at TDC)

PcPs: Compression pressure ratio

P_{max}: Maximum cylinder pressure

P_{rise}: Pressure increase from the combustion

SCR: Selective catalytic reduction

SOC: Start of combustion

SOI: Start of injection

TDC: Top dead centre

THC: Total hydrocarbon

ULSFO: Ultra-low-sulphur fuel oil (0.1%S)

VLSFO: Very-low sulphur fuel oil (0.5%S)

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