Experimental study on a retrofitted marine size spark-ignition engine running on port-injected 100% methanol

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Synopsis

The maritime sector is currently facing a serious challenge of reducing emissions to meet the strict international and national emission regulations. In response to this challenge, the industry is initiating several research initiatives focused towards employment of alternative fuels in marine internal combustion engines. One of the promising alternatives is methanol. Methanol has multiple advantages such as higher volumetric energy density (including storage) compared to alternative fuels such as hydrogen and ammonia. Methanol is available at a relatively larger scale than hydrogen. Furthermore, compared to liquefied fuel storage of alternative fuels, methanol can be easily stored as a liquid onboard ships because of its liquid aggregation state at atmospheric conditions. Other advantages such as low flash point compared to LNG, fast biodegradation capabilities, and high water miscibility make it a high potential marine fuel with good safety considerations.

Recognizing these benefits of methanol, the Green Maritime Methanol (GMM) project aims to investigate the uptake of methanol as an alternative shipping fuel for maritime vessels. GMM is a Dutch collaborative project among the entire sector. Research on application of methanol in marine engines is in its infancy stages and there is a knowledge gap pertaining to validated effects of methanol combustion in marine engines. This paper focusses on experimental investigations of methanol combustion in a marine spark ignited engine.

To study the impact of combusting methanol, experiments were performed on a CAT G3508A natural gas (NG) engine, which is a turbocharged, spark-ignited (SI) engine with 8 cylinders and a rated power of 500 kWe at 1500 rpm. The engine was retrofitted and controlled to allow port-fuel injection (PFI) of methanol. Tests with stable engine operation were achieved with 100% methanol at 25%, 50% and 75% engine loading and constant engine speed of 1500 rpm. Engine performance on 100% methanol was also studied for varying ignition timings.

In this paper, we discuss the retrofitting and safety considerations required for methanol combustion. We also present the experimental methodology employed to allow operation of the SI NG test engine on 100% methanol. The performance results of the engine operating on methanol are analyzed and compared against the results for natural gas. The engine performance for the two fuels is compared with respect to cyclic variations, in-cylinder combustion performance, engine efficiency. The results showed that engine efficiency improved by 2.2% and 0.9% at 50% and 75% load with methanol compared to natural gas for the same test conditions of ignition timing and NOx emissions. These preliminary methanol engine results show that improvements in engine efficiency can be obtained from a retrofitted SI NG engine converted to operate on 100% methanol without making any modifications to the geometrical engine specifications.

Keywords: Marine engine; Spark-ignition; Methanol; Port-injection; Ignition timing; Experiments

1. Introduction: Moving away from fossil fuels

Currently, the maritime sector is facing the major challenge of reducing CO_2 emissions by 50 % until 2050, while the global transport demand for goods increases by a factor of 2.5 to 4 [1],[2]. To reduce the CO_2 emissions from ships, one of the promising solutions is to employ fuels with less carbon emissions per kilojoule of fuel compared to marine diesel. When using hydrogen or ammonia as fuel, CO_2 emissions from tank-to-propeller are eliminated. Additionally, these alternative fuels promise ultra-low well-to-tank greenhouse gas (GHG) emissions if produced renewably [3]. Unfortunately, at present, application of hydrogen for maritime applications can be challenging due to its lowest volumetric energy densities (including storage) compared to other alternative fuels in addition to very high (current) production costs and very low (current) production capacity [4]. Ammonia (with liquid storage) has a better volumetric energy density than liquid hydrogen. However, it is toxic to humans, which may restrict its wide scale implementation as a marine fuel.

Compared to alternative fuels such as hydrogen and ammonia, methanol has benefits such as higher volumetric energy density (including storage) [4] , and can be easily stored as a liquid. Methanol can be produced from renewable sources, thus, showing promise of extremely low GHG emissions. It also has a much lower toxicity

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than ammonia, which is evidential from the lethal 30 minute exposure levels for methanol of 14000 ppm compared to 1600 ppm for ammonia [5], [6]. Furthermore, the short biodegradation time period of methanol, in combination with its infinite water miscibility, make it an attractive marine fuel from a safety perspective in case of any spills in the marine environment, which strongly contrasts the disasters of a conventional hydrocarbon fuel spillage [7]. Recognizing these benefits of methanol, the Green Maritime Methanol (GMM) project aims to investigate the uptake of methanol as an alternative shipping fuel for maritime vessels.

In the maritime world, few demonstration projects have been focused towards implementation of methanol. Stena Germanica, a passenger ferry, employs medium-speed four stroke marine engines by Wartsila running on (separate) directly injected methanol and marine gas oil as pilot fuel [8]. Besides this, tankers by Waterfront Shipping operate with MAN low-speed two-stroke engines, again using separate direct injection of methanol and pilot fuel (MGO or HFO) [9]. Measurement and engine performance data on these engines is limited, with only information showcasing emission regulations compliance and diesel-like efficiencies being available by manufacturers. In recent years, there are additional demonstration projects that have been initiated for maritime applications such as LeanShips [10], MethaShip, SUMMETH [11], etc. that focus on different aspects of methanol application such as marine engines, safety, life-cycle costs, environmental impact, bunkering, etc. However, research and publications on combustion and engine performance by application of methanol in marine engines is scarce.

In existing literature, researchers have investigated the application of methanol in automotive engines through port-fuel injection (PFI) and direct injection. Verhelst et al. provided a comprehensive review of methanol application in an ICE [12]. The authors presented various studies showcasing improvements in efficiency compared to gasoline for automotive applications [13], [14]. Furthermore, studies have also shown that methanol combustion can lead to lower NOx emissions compared to gasoline due to reduced in-cylinder temperatures [12]. Thus, researchers have shown promising results for implementation of methanol in automobile engines. However, the qualitative and quantitative impacts of methanol combustion in a marine size engine are expected to be different in comparison to automotive engines due to various factors such as lower surface-to-volume ratios, higher power output (brake mean effective pressures), different emission regulation requirements, possibility of different operating window like marine natural gas engines, etc. All these factors can impact engine efficiency, emissions, engine operating limits and load-pick up capabilities, which need to be studied for marine engines. Consequently, there is a knowledge gap pertaining to validated effects of methanol combustion in marine engines. Therefore, this paper focusses on experimental investigations of methanol combustion in a marine spark-ignited (SI) engine. In this paper, we discuss the preliminary results of tests conducted on a CAT G3508A SI test engine. The methanol is injected using a port fuel injection system, and ignited with a spark.

2. Methodology

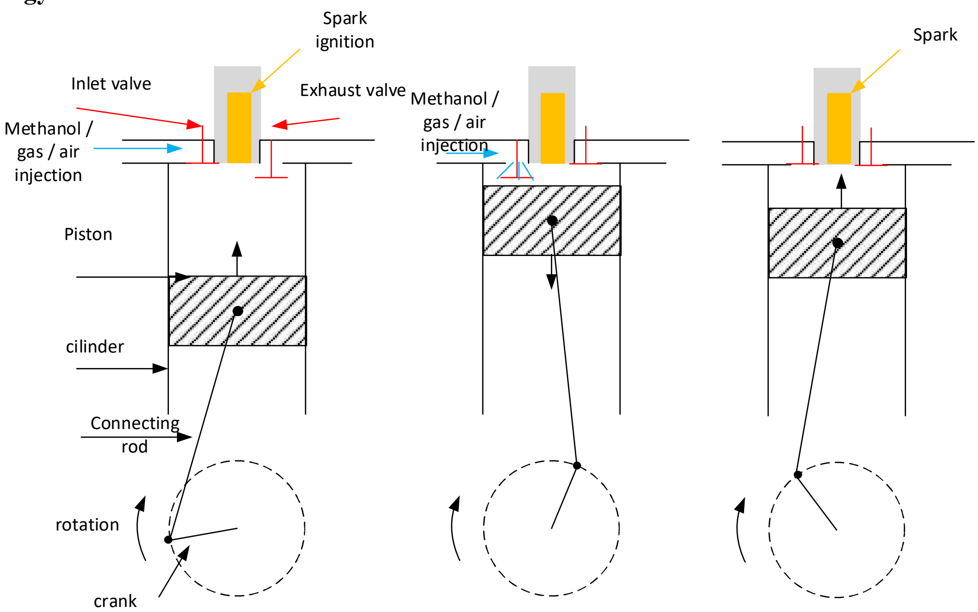

Figure 1: Schematic overview of the port-fuel injection and spark ignition process in the engine.

The low cetane number and high auto ignition temperature requirements compared to diesel, do not allow ignition of 100% methanol (M100) in standard compression ignition engines with low to normal compression ratios [15], [16]. For this study, tests were performed on a spark-ignition engine, and methanol was injected along

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with air through the inlet valve as depicted in figure 1. The incoming mixture of methanol and air is compressed during the compression stroke to allow for homogenous mixture formation, which is ignited with a spark leading to methanol combustion. The engine used for this research is a CAT G3508A, which is a turbocharged, lean-burn, spark-ignited, natural gas engine with 8 cylinders, zero-valve overlap and a rated power of 500 kWe at 1500 rpm [17]. Figure 2 shows a picture of the test engine and the control room setup for monitoring engine operation; Table 1 provides the engine specifications. The engine drives a generator. The following subsections describe the engine adaptations, sensors for engine performance measurement and the test plan.

(a) (b)

Figure 2: The converted CAT G3508A SI engine (a) and control room (b).

Table 1: Engine specifications

Parameter	**Value**	**Units**
Number of cylinders	8	-
Bore	170	mm
Stroke	190	mm
Displacement	34.5	L
Rated speed	1500	rpm
Rated Power	500	kW
Compression ratio	12:1	-
Boost pressure	2.2	bar
Cycles	4	-
Firing order	1-2-7-3-4-5-6-8	-
Inlet valve open (IVO)	8.7	^{0}CA ATDC
Inlet valve close (IVC)	21.5	^{0}CA ABDC
Exhaust valve open (EVO)	20.1	^{0}CA BBDC
Exhaust valve close (EVC)	11.8	^{0}CA BTDC
Geometry of piston bowl	Centric hemispherical combustion chamber (HCC)	-
Volume of piston bowl	250	cm^3

2.1. Engine modifications for safe operation with methanol

Several systems on and around the engine were modified for safe operation and measurements with 100% methanol. The following systems are retrofitted on the CAT G3508A SI engine to operate a 100% methanol engine with port fuel injection (PFI):

For operation:

- Modified fuel system
  The modified fuel system includes modifications to fuel storage, fuel pump, filters, piping, special designed fuel rail, adjusted PFI system with high flow methanol injectors and valves. Methanol is corrosive to aluminum alloys, therefore, chemical resistant materials such as stainless steel and rubber are used. The stainless steel fuel rail and the port fuel injection system are shown in figure 3.

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- Modified charged air cooling system
  New three-way valve and a modified charger air cooling control system were installed to control the temperature after the cooler. The standard system on the engine allows for constant temperature regulation after the cooler. However, the air temperature after the air cooler could not be kept constant with methanol injection. Due to the high evaporation energy requirements of methanol, the temperature after fuel injection at inlet port is load dependent. The modified charge air cooling system was installed because of concerns of reaching operating limits with the previous constant temperature regulator after the cooler at higher loads.
- New designed control system to control the fuel flow, air flow, engine speed, etc.

More details about charge air cooling system and the new control systems are given in Appendix A.

(a) (b) (c)

Figure 3: The modified fuel system of methanol including a double walled stainless steel fuel rail (a), injector housing for PFI (b) and the used 2200 cc/min injectors (c).

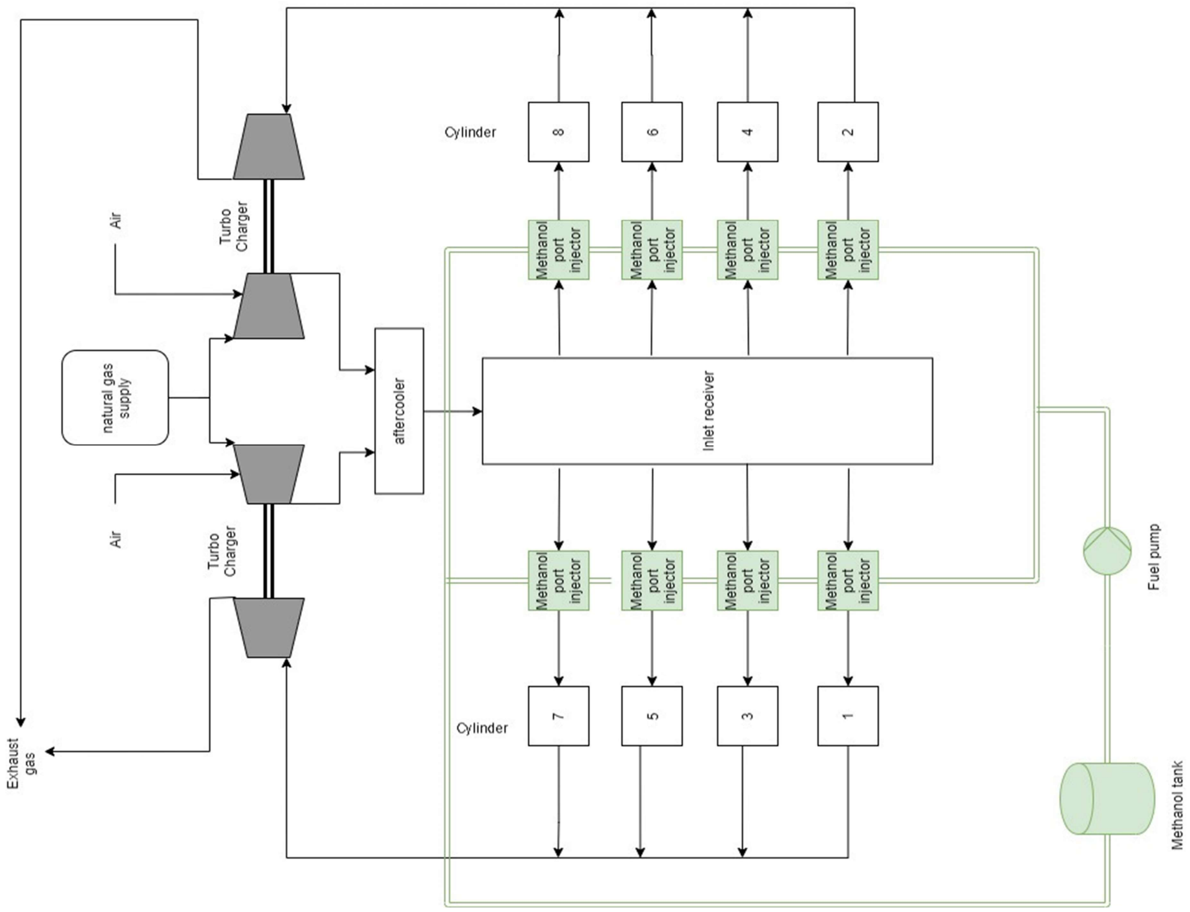

Figure 4: Schematic overview of the test set-up.

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For safety:

- Two continuous methanol vapor detectors capable of measuring methanol content from 0-1000 ppm were installed. Such a detection system was necessary because methanol is toxic to humans beyond 200 ppm. Furthermore, human senses can detect methanol only when the concentration exceeds 10 times the 200 ppm safe limit [18].
- A fire detection system capable of detecting alcohol fires inside the test bed and outside at the fuel tank was installed.
- A nitrogen inert gas system was installed to reduce the amount of oxygen present in the fuel system and to clean the fuel system.
- Emergency stop buttons and the vapor detection systems were connected to the control system to instantly stop the engine and the fuel pump.

For measurements:

- A Testo-350 is used for exhaust gas measurements
- A Kibox 2893A is used for crank angle and in-cylinder pressure measurements.
- A Dewetron Dewe-2600 is used to collect and store all sensor data excluding the Kibox data.

Figure 4 shows the engine test setup schematic with natural gas (NG) injection (for 100% NG operation), and methanol injection through port injectors (shown in green). The fuel filters, pressure sensors and valves are left out of the overview for simplicity. Both fuel supply systems can operate simultaneously.

2.2. Sensors

The most important input sensors are given in table 2, including the range and sensitivity of these sensors. The NOx emissions are measured with the Testo-350 together with other engine emissions for further research. All sensors were calibrated prior to the testing.

Table 2: Summary of most important sensors

Sensor	Use	Range	Accuracy
In cylinder pressure sensor (Kistler 7061B)	In-cylinder pressure on cylinder 3,4,5 and 6	0 - 250 bar	± 1.25 bar
	Crank angle (CA) determination	0 - 360°	± 0.23°
TC Model TE1260	Air and exhaust gas temperatures	-40 ÷ +1000 °C	± 1.5 °C ± 0.004*(T)
PT100	Lubrication oil and cooling water temperatures	-220 ÷ +600 °C	± 0.3 °C ± 0.005*(T)
Pressure sensor 10 bar	Air and exhaust gas pressures	0 - 10 bar	± 0.02 bar
Flow sensor KRAL BEM 500	Methanol flow	0 - 300 kg/s	± 0.3 kg/s (± 0.1%)

2.3. Test plan

In this study, NG performance tests were performed for comparison with methanol performance. For NG tests are conducted at 250 kWe (50% load) and 375 kWe (75% load) with an ignition timing of 20 ^{0}CA BTDC. The engine tests on 100% methanol were performed according to a step-by-step action plan to ensure the tests were conducted safely and efficiently during the 2 weeks of testing. To get the engine running on 100% methanol, it was necessary to find the amount of methanol and air needed for stable engine operation at 0% load because this would allow to switch from the NG to methanol at start-up in 5 seconds, For this reason, the following steps were performed to achieve the 0% stable engine load run with methanol, which has been also mentioned in table 3: Step 1: 100% NG operation at 125 kWe, Step 2: Slowly increasing methanol flow and decreasing NG flow at 125 kWe, Step 3: Achieve 100% methanol at 125 kWe, Step 4: Decreasing to 0% load and determine amount of methanol and air needed at 1500 rpm. After running the engine with 100% methanol at 0% load, stable load conditions were found for 50% and 75% load. The general operating conditions for these stable load runs are:

- Fuel injection pressure: 5 bar
- Engine speed:1500 RPM.
- Ignition moment: 20 ^{0}CA BTDC.
- End of injection was fixed between inlet valve open (IVO) and inlet valve closed (IVC). Injection ended exactly at the middle of this duration to allow all injected methanol to flow to the cylinder.

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- Injection duration was load dependent varying between 5 to 25 ms.
- Constant NOx emission of 500 mg/Nm^3 at 5% reference oxygen.
- $T_{manifold} = 40\ ^0C$
- Air-excess ratio was varied and load dependent. Although the air-excess ratio varied, it was close to 1.60, thus, depicting lean burn engine operation.

After the stable load runs a series of additional performance tests were executed as shown in table 3. The spark-timing was advanced by steps of 2 degrees, to study the impact of ignition timing sweep while keeping the NOx and load constant. This 2nd set of methanol tests was conducted at a constant NOx emission of 500 mg/Nm^3 at 5% reference oxygen (equal to ± 340 ppm), with the main reason being the tightening emission legislations. For the NG test engine, the 500 mg/Nm^3 of NOx value is close to and lower than the NOx IMO TIER-III limit for this engine, which is 2.08 g/kWh [19]. The air-excess ratio was varied to keep the NOx emissions constant. The fuel pressure and engine speed were kept constant as during the stable load run conditions.

Table 3: Performance tests conducted with 100% methanol and 100% NG.

	125 kW (25 %)	**250 kW (50 %)**	**375 kw (75 %)**
Natural Gas			
20 ^{0}CA BTDC and constant NOx of 500 mg/Nm^3		X	X
Methanol			
Stable 0 % load run			
Stable load runs (at ignition timing 20 ^{0}CA BTDC) and constant NOx	X	X	X
2nd set of methanol tests (varying ignition timings)			
Ignition timing 16 ^{0}CA BTDC and constant NOx		X	X
Ignition timing 18 ^{0}CA BTDC and constant NOx		X	X
Ignition timing 20 ^{0}CA BTDC and constant NOx		X	X
Ignition timing 22 ^{0}CA BTDC and constant NOx		X	X
Ignition timing 24 ^{0}CA BTDC and constant NOx		X	X

3. Preliminary experimental results

In this section, the experimental results are discussed and the impact of methanol combustion on various performance parameters such as average cylinder pressure, cycle-to-cycle variations and engine efficiency are presented. In this analysis, the results for 100% methanol are compared against those found for 100% natural gas. Additionally, the section discusses the impact of variations in ignition timing on engine performance with 100% methanol. In this results section, all the measurements are compared at 500 mg/Nm^3 of NOx.

3.1. Average cylinder pressure

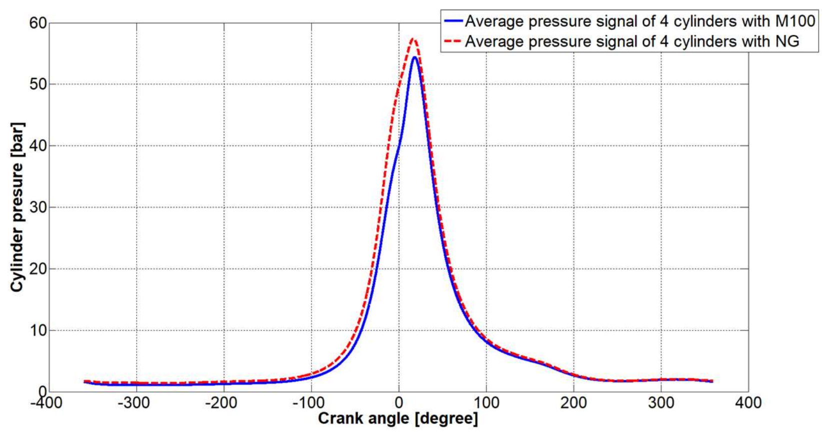

Figure 5: Mean cylinder pressure of NG compared to methanol at a power set at 375 kWe.

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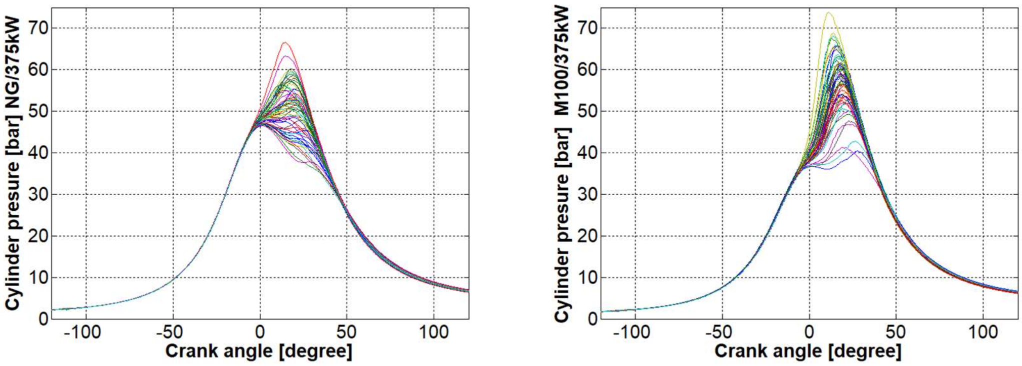
To compare and study the in-cylinder pressures measured during 100% NG operation and 100% methanol, we calculated the average pressure of the 60 cylinder cycles for all the four measured cylinders as function of crank angle. The engine settings corresponding to the measurements shown in figure 5 were 375 kWe power output, 500 mg/Nm^3 of NOx, ignition at 20 ^{0}CA BTDC and 1500 rpm. As seen in figure 5, and based on average of all four cylinders, methanol showed lower in-cylinder pressures compared to NG at the same engine operating conditions. However, it is vital to note that this was not the case for every single cylinder. For instance, cylinder 4 showed higher average in-cylinder pressures with methanol compared to cylinder 4 with NG. To understand the exact impact of methanol on the in-cylinder pressures further modelling and analysis will be performed in the near future.

3.2. Cycle-to-cycle pressure variations

Figure 6 shows the cyclic variations of in-cylinder pressures measured for natural gas (a) and methanol (b) at 375 kWe load, 500 mg/Nm^3 NO_x, ignition-timing of 20 ^{0}CA BTDC and 1500 rpm.



(a) (b)

Figure 6: cycle-to-cycle pressure variation plots using natural gas (a) and methanol (b).

To quantify the cycle-to-cycle variations and compare them for methanol and natural gas, first, the coefficient of variation (COV) for the maximum pressure is estimated [20]:

$$COV_{pmax} = \frac{std_{P_{max}}}{P_{max,avg}} \cdot 100\% \tag{1}$$

where $std_{P_{max}}$ is the standard deviation of the maximum in-cylinder pressure:

$$std_{P_{max}} = \sqrt{\frac{\sum_{i=1}^{m}\left(P_{i,max} - P_{max,avg}\right)^2}{m-1}} \tag{2}$$

where, $P_{max,avg}$ is the average of the maximum in-cylinder pressure over *m* cycles, $P_{i,max}$ is the maximum in-cylinder pressure of the *i*-th cycle, and *m* is the number of cycles = 60.

With methanol, a COV_{pmax} of 11.41% was found for cylinder 4, and the COV_{pmax} varied between 9.53% and 14.82% for the other cylinders. For NG a COV_{pmax} of 9.34% was found for cylinder 4, and for the other cylinders the value of COV_{pmax} varied between 7.63% and 9.34%. Higher COV_{pmax} values were found for all cylinders with methanol compared to NG. Xie [20] found comparable COV_{pmax} between 3.4% and 14% values for a SI engine running on methanol. Ma et al. [21] found values between 9% and 11 % for NG at similar ignition timings.

In this study, we also compared the cycle-to-cycle variations between NG and methanol by computing the coefficient of variation (COV) of the indicated mean effective pressure (imep) [22]:

$$COV_{imep} = \frac{std_{imep}}{imep} \cdot 100\% \tag{3}$$

Where std_{imep} is the standard deviation of the indicated mean effective pressure (imep)

$$std_{imep} = \sqrt{\frac{\sum_{i=1}^{m}\left(imep_i - imep_{avg}\right)^2}{m-1}} \tag{4}$$

Where imep is the work done during one cycle divided by the stroke volume:

$$imep = \frac{W_i}{V_s} = \frac{\int_{cycle} p \cdot dV}{V_s} \tag{5}$$

Where W_i is the indicated work, V_s is the stroke volume, p is the pressure and V is the volume.

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For the four cylinders with in-cylinder pressure measurements, the value of COV_{imep} varied between 3.72% and 6.11% at 375 kWe power and ignition timing of 20 ^{0}CA BTDC when running on 100% methanol. For NG, the cylinders varied between 1.76% and 2.93% at the same conditions. In Heywood [18], a COV_{imep} exceeding 10% in an automotive engine is considered as poor driving conditions. Furthermore, Sapra et. al [19], found that the current test engine operation became rough and unstable when COV_{imep} crossed 10 % for 100% natural gas. Therefore, the 10% value is considered as the misfire or stability limit. For test conditions listed in table 3, the values of COV_{imep} for methanol and NG were not at all close to the 10% value, thus, indicating stable operation. However, by comparing the COV_{imep} values of methanol and natural gas in the current test engine, it is clear that engine operation with methanol was found to be less stable than that with natural gas. The authors hypothesize that the lower stability from methanol could be due to one or a combination of the following reasons:

- Prior to the reported measurements, the fuel injector of cylinder 4 was replaced due to dirt in the injector. After replacing and fitting cylinder 4 with a clean injector, the COV_{imep} for that cylinder reduced from 10.59% to 3.72%, which is within the range of the above reported values found for methanol. It is vital to note that even when cylinder 4 indicated COV_{imep} value of 10.59%, other cylinders had COV_{imep} in the acceptable range of 3.15 to 5.21%. Furthermore, after replacing the injector of cylinder 4, the COV_{imep} value of 3.72% was slightly lower than the 4.02 to 6.11% values found for cylinder 3, 5 and 6. Thus, cleaning and replacing the cylinder clearly showed an improvement, however, the variations were still higher than those found for NG at the same operating condition. Given that dirt can cause injector blockage, which can lead to significantly high COV_{imep} values, the authors would like to consider the possibility that the higher variations in COV_{imep} with methanol could be due to still remaining dirt particles in the injectors, originating from the production process of the new fuel system. This phenomenon will be further investigated in future experiments.
- Another reason could be the 4 times higher evaporation energy requirement of methanol compared to diesel in combination with port injection instead of injection more upstream. For NG, there is no evaporation energy needed in the engine because it is in a gaseous state. The high evaporation energy requirement of methanol inside the cylinder could lead to non-uniform fuel evaporation before ignition, thus, leading to a non-homogenous mixture formation with local air-excess ratios that are higher than 1.6, and, therefore, prone to cause combustion instability and higher in-cylinder pressure variations.
- Lastly, Winterbone and Turan [23] have reported that at lean air-excess ratios (approximately higher than 1.17) methanol can have a lower (laminar) flame speeds than that of methane. The current test engine operates at much higher air-excess ratios of 1.6 when operating on 100% methanol and 100% natural gas. Reduced premixed flame speed of methanol at these leaner air-excess ratios could lead to slower combustion rates, which may contribute to higher cycle-to-cycle variations [24]. This hypothesis of lean methanol combustion causing lower combustion rates will be investigated in near future through in-cylinder process and heat release modelling.

3.3. Impact of ignition timing variation

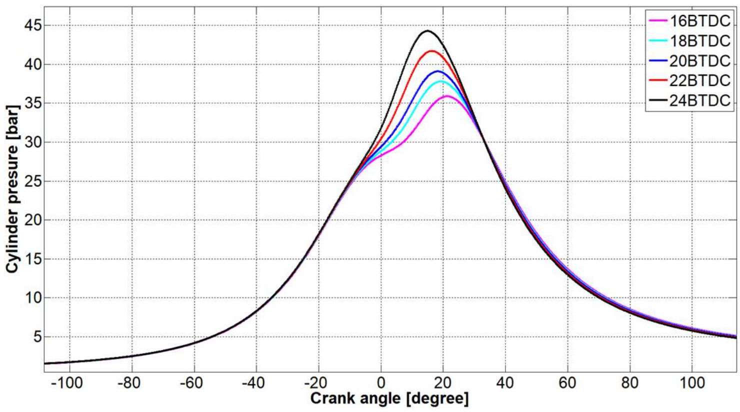

Figure 7: Mean pressure of all cylinders with changing ignition timing at 250 kWe on 100% methanol.

Figure 7 shows the average pressure versus crank angle for various ignition timings at operating conditions of 250 kWe, 500 mg/Nm^3 NOx and 1500 rpm when operating on 100% methanol. As seen in figure 7 the value of maximum pressure increased and the position shifted closer to TDC, as ignition timing was advanced from 16 to

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24 ^{0}CA BTDC. Furthermore, the figure indicates a higher rate of pressure rise with advanced ignition timing. With respect to combustion stability, no clear improvements were found at different ignition timings as seen from table 4. Compared to 100% NG operation at 250 kWe load, 500 mg/Nm^3 NOx and 20 ^{0}CA BTDC ignition timing, once again, methanol combustion showed higher values of COV at the same operating conditions, thus, indicating relatively lower stability. However, the COV_{imep} values with 100% methanol did not exceed 10%, thus, showcasing reasonable combustion stability over the range of tested ignition timings, as seen in table 4.

Table 4: Average COV_{pmax} and COV_{imep} of 4 cylinders at 250 kW and 1500 rpm for methanol at different ignition timings

Ignition timing	**COV_{pmax}**	**COV_{imep}**
Ignition timing 16 ^{0}CA BTDC	12.48 %	5.02 %
Ignition timing 18 ^{0}CA BTDC	10.59 %	5.76 %
Ignition timing 20 ^{0}CA BTDC	13.09 %	4.62 %
Ignition timing 22 ^{0}CA BTDC	13.46 %	4.10 %
Ignition timing 24 ^{0}CA BTDC	12.89 %	4.55 %

3.4. Efficiency

For the efficiency on NG operation, an engine specific performance datasheet from the engine manufacturer has been used at 18 ^{0}CA BTDC (ignition timing) instead of our own performance tests at 20 ^{0}CA BTDC, because of challenges in accurate measurements of NG fuel flow during our experiments [25]. The effective (brake) engine efficiency at 75 % load was 33.9 % when running on 100% natural gas (LHV = 34.5 MJ/m^3), available from Dutch NG grid. LHVs for the two fuels are given with respect to their corresponding fuel flow measurement units, i.e., m^3/hr for NG and kg/hr for (liquid) methanol. We found that at 75 % engine load, NOx emissions of 500 mg/Nm^3 and ignition timing of 18 ^{0}CA BTDC engine efficiency improved to 34.8 % when operating on 100% methanol (LHV = 19.9 MJ/kg). The efficiency improved by 0.9 % and 2.2 % with methanol compared to natural gas at power outputs of 375 kWe and 250 kWe, respectively, with same ignition timing and NOx emissions. Furthermore, on 100% methanol, the engine efficiency increases up to 35.9 % with an advanced ignition timing of 24 ^{0}CA BTDC as shown in table 5. Thus, the efficiency of the engine improved by 2% at 75% load with advanced ignition timing (of 24 ^{0}CA BTDC) while operating on 100% methanol compared to natural gas (at 18 ^{0}CA BTDC). This improvement in engine efficiency at the advanced ignition of 24 ^{0}CA BTDC with 100% methanol increases to 3% at 50% load as seen in table 5.

These preliminary methanol engine results show that improvements in engine efficiency could be obtained from a retrofitted SI NG engine converted to operate on 100% methanol without making any modifications to the geometrical engine specifications such as cylinder or piston geometry. Additional efficiency improvements could be gained by optimising the engine for methanol combustion, which is recommended for future research. One direction of future research to achieve improvements in efficiency could be increments in compression ratios for methanol combustion [12].

Overall, the efficiency is higher compared to NG. According to Verhelst, a higher efficiency with methanol is expected to be gained possibly due to lower wall heat losses and from the increased charge density, which leads to a higher volumetric efficiency [12][26][26][25][24][24]. The impact of methanol on combustion durations and heat losses and on engine efficiency will be further studied in near future by developing in-cylinder heat-release models.

Table 5: Engine efficiency at 50 and 75 % load at 500 mg/Nm^3 NOx emission.

Testing conditions	**250 kWe (50 % load)**	**375 kWe (75 % load)**
Methanol (LHV = 19.9 MJ/kg)		
Ignition timing 16 ^{0}CA BTDC	32.5%	34.4%
Ignition timing 18 ^{0}CA BTDC	33.2%	34.8%
Ignition timing 20 ^{0}CA BTDC	33.6%	35.5%
Ignition timing 22 ^{0}CA BTDC	33.8%	35.7%
Ignition timing 24 ^{0}CA BTDC	34.0%	35.9%
Natural gas (LHV = 34.5 MJ/m3)		
Ignition timing 18 ^{0}CA BTDC	31.0%	33.9%

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4. Looking forward

The experiments discussed in this paper reveal that a marine size spark-ignited NG engine can be retrofitted to run stable on 100% port-injected methanol, without making any modifications to the geometrical engine specifications such as cylinder or piston geometry. We collected sufficient data to build and verify adapted diesel models [27], [17], [28] to further study methanol combustion. Due to the PFI of methanol, an evaporation sub-model is being developed, which will be used to further study the 100% methanol heat release and, then, compare it with that of 100% natural gas.

The engine will be installed in the lab facilities of the Netherlands Defence Academy (NLDA) in Den Helder, where we will further vary combustion parameters to validate our models and find optimum settings for emissions and efficiency. Next, we intend to study the impact of load steps and load variations on the internal combustion process and resulting efficiency and emissions. These emissions will not be limited to NOx. It is our intention to study the impact of the combustion parameters on other than NOx emissions as well.

5. Conclusions

A spark ignited natural gas driven engine was successfully adapted to run on methanol, injected via the gas inlet port. The following can be concluded from the performance tests:

- The engine runs stable on 100% methanol at 250 kWe (50% load) and 375 kWe (75% load) between 16 ^{0}CA BTDC and 24 ^{0}CA BTDC ignition timing at 500 mg/Nm^3 NOx emission.
- The experiments showed higher COV_{imep} for methanol compared to natural gas at all tested conditions, thus, indicating lower stability. The authors hypothesize that the higher cyclic variations are due to one or a combination of following reasons: a.) Blocked injectors, b.) High evaporation heat of methanol and c.) Lower flame speed of methanol at the tested leaner air-excess ratios.
- Efficiency improved by 0.9% and 2.2% with methanol compared to natural gas at same test conditions of power output, ignition timing and NOx emissions. These preliminary methanol engine results show that improvements in engine efficiency could be obtained from a retrofitted SI NG engine converted to operate on 100% methanol without making any modifications to the geometrical engine specifications such as cylinder or piston geometry.
- The maximum pressure increased and shifted closer to TDC with advanced ignition timing when operating on 100% methanol. Thus, leading to engine efficiency improvements of about 1.5% at advanced ignition timing of 24 ^{0}CA BTDC compared to 18 ^{0}CA BTDC.

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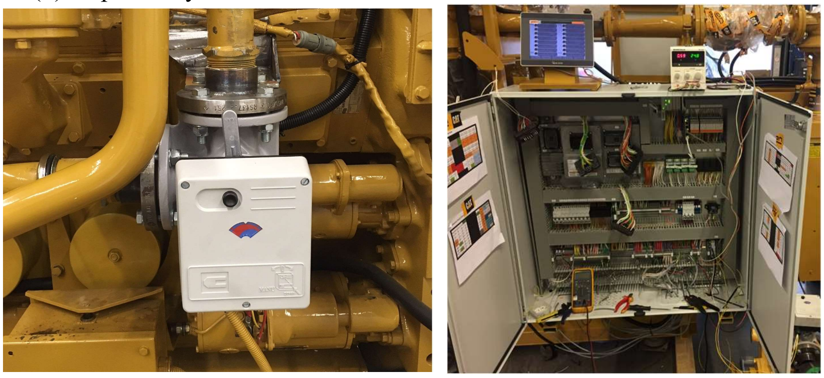
Appendix A: Engine modifications

Several systems are modified on the CAT G3508A SI engine to convert it to a port-injected 100% methanol engine. These systems have been detailed in sub-section 2.1. In this appendix, the engine cooling system and control system have been discussed in more detail.

A.1 Charged air cooling system

The charged air cooling system on the G3508A SI NG engine includes an aftercooler, which is placed after the compressors and before inlet receiver. The aftercooler cools the hot compressed air (and natural gas when operating on natural gas). The original regulator will designed to keep the combined air and natural gas flow in the receiver at a temperature of 35°C. The air cooling system is separate from the internal engine cooling system.

But for running the engine on methanol modifications are made on the charged air cooling system. Due to the PFI the temperature of the air inlet will be changed just before the cylinder inlet valve. This temperature change is significant due to the high heat of vaporization of methanol. The air temperature could not be kept at a constant value after the air cooler because with methanol injection the temperature after the air cooler is load dependent. Therefore, a three-way valve was installed before the cooler instead of the previous constant temperature regulator. The three-way valve was opened and closed by an electric motor, which was regulated by a PLC. With the three-way valve the cooling water flow to the air cooler was controlled, thus controlling the manifold temperature. The three-way cooling valve, the electric motor and the control system cabinet are shown in figure A.1(a) and (b) respectively.



(a) (b)

Figure A.1: Cooling three-way valve with electric motor (a) and control system cabinet (b).

A.2 Control systems

There are two electric control systems on the engine to allow operation on methanol. The main control system is designed and delivered by Woodward. The secondary control systems is a PLC made by PON power. Both electric systems are described as follows.

A large engine control module (LECM) is delivered and designed by Woodward to PON specification. The LECM controlled the following engine's operations: Speed/load control, Air/fuel ratio control, Air flow control, Ignition timing, injection timing control and misfire and knock detection. The LECM controlled the air throttle, methanol injectors and the spark-ignitors to perform these operations. The LECM used the air throttle and methanol injection timing for the speed/load control. Initially, the LECM will have a mapping for the injection time and ignition of every cylinder which will be adjusted during the first operations to get stable load conditions. The LECM will be also able to use the pressure sensors in the cylinder heads and control the injection timing of the individual methanol injectors for optimizing load control over the different cylinders, but this has not been tested yet and will be for future experiments.

A PLC was designed by PON Power to control all other secondary engine systems, e.g. the charged air cooling system. The PLC system is placed in a Rittal control cabinet together with the LECM as shown in figure A.1(b). This cabinet will be placed on the engine and connected to a human machine interface (HMI). The following functionalities to control and monitoring of all secondary systems are taken into the PLC: engaging/disengaging methanol fuel system, controlling manifold temperature, nitrogen inert gas system, handling of system alarm and registration of system events.

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