CIMAC CONGRESS 25
ZURICH, MAY 19 – 23, 2025

2025 | 130

Dynamic performance of a high-pressure direct injection methanol-fueled engine

System Integration & Hybridization

Benny Mestemaker, Royal IHC

Martijn Kom, Boskalis
Martin de Geus, Van Oord
Juho Repo, Wärtsilä
Jens Lassila, Wärtsilä
Ari Haanperä, Wärtsilä

ABSTRACT

The maritime energy transition will have an immense impact on the vessels currently under construction. The upcoming legislation will force vessels to reduce their greenhouse gas emissions significantly. This will result in a switch to fuels of a non-fossil origin (e.g., biomass and/or carbon from air). The preference is for simple and short molecules as these are more efficient/less energy intensive to make. Methane, methanol, hydrogen, and ammonia are the main candidates for the world fleet depending on the application.

Methanol is a strong contender for use in work vessels such as dredgers. Dredging vessels often have a large crew and operate close to populated areas (in ports and coastal areas), therefore toxicity, environmental impact and safety of the fuel are important. The energy density of methanol and the ability to store it at atmospheric conditions ensure the lowest impact on the vessel's autonomy. The highly transient load profile of dredging vessels places strict requirements on the integration of the engine in the drive system.

The objective of the research in this paper is to investigate the integration of the Wärtsilä 32 methanol dual-fuel engine in the drive system of existing and future dredging vessels. To determine the integration potential and limitations, both static and dynamic engine tests have been performed on the Wärtsilä 32 methanol engine. The transient capabilities of the engine are compared with the operational profiles of dredging vessels. The research includes the potential requirements for energy storage systems to ensure the same or better performance of the vessels operation.

1 INTRODUCTION

The International Maritime Organization (IMO) and other governing bodies have since many years implemented legislation to reduce harmful emissions such as nitrogen oxides (NO_X), sulphur oxides (SO_X) and particulate matter (PM) [1]. More recently, legislation requiring reductions of greenhouse gas (GHG) emissions have been introduced. These regulations are meant to achieve net-zero GHG emissions in the maritime sector close to 2050 to be consistent with the Paris agreement long-term temperature goal [2].

This legislation is forcing a maritime energy transition away from the traditional heavy fuel oil (HFO) towards more clean alternatives such as methane, methanol, ammonia and hydrogen. Each of these energy carriers has environmental benefits such a less harmful emissions [3] and is cheaper to renewably produce than diesel-like fuels [4]. The most suitable fuel depends on the application and the required performance. Figure 1 shows the gravimetric and volumetric energy density of some of the considered fuel options including the effect of the fuel storage system.

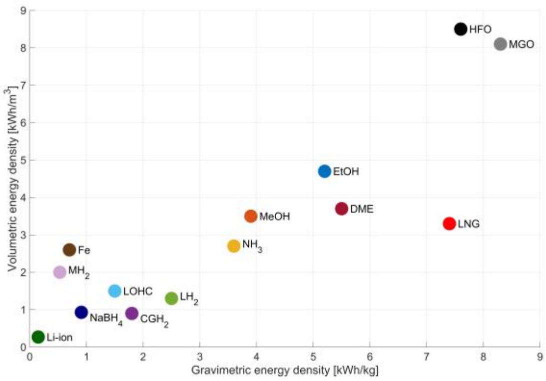

Figure 1. Gravimetric and volumetric energy density of marine fuels (including storage system). HFO: heavy fuel oil, MGO: marine gas oil, LNG: liquefied natural gas, DME, dimethyl ether, EtOH: ethanol, MeOH: methanol, NH_3: ammonia, LH_2: liquefied hydrogen, CGH_2: compressed gaseous hydrogen, LOHC: liquid organic hydrogen carrier, $NaBH_4$: sodium borohydride (fuel 30, spent fuel, reactor not included), Fe: iron powder (spent fuel, reactor not included), MH_2: metal hydrides (low temperature AB2 Ovonic) (adapted from: [5])

Methanol is especially interesting for work vessels such as dredging vessels. Dredging vessels are power dense, require a high autonomy and have a relatively large crew [5]. This rules out low energy density fuels for some applications (such as batteries) and highly toxic fuels (such as ammonia). LNG is already applied in dredging vessels, but the methane slip of current engines is an issue which has to be solved [6]. An additional issue of the port-fuelled LNG dual fuel (DF) engines is the limited transient capability of the engines [7]. The advantages of methanol for work vessels are the option to store the fuel as a liquid below deck without significantly impacting the operation of the vessel and the higher safety of methanol compared to the other alternative fuel options.

Dredging vessels have large power fluctuations and thus the transient capabilities of methanol-fuelled engines is important for the proper integration of methanol-fuelled drive system. Dredging operations have a characteristic behaviour of soil (inhomogeneous, particle size and fracturing dependency), during known extraction or discharging processes, in dredging. This results in highly dynamic operations of cutter, drag head and pumps and subsequently in dynamic power requirements as shown in Figure 2 [7-9].

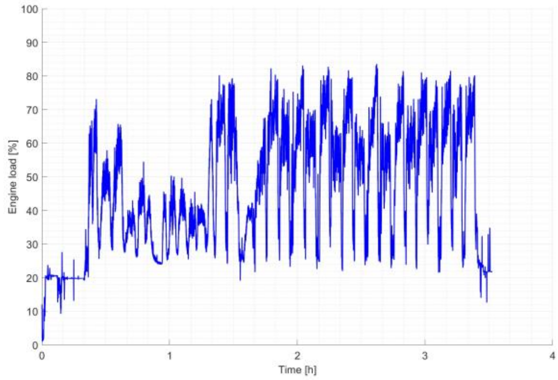

Figure 2. Example load profile dredging vessel [7]

The objective of this paper is to investigate the dynamic performance of methanol-fuelled engines. This is done to determine the applicability of these engines for vessels with a highly transient operational profile such as dredging vessels.

The following sections present the methanol fuel system, engine technology and engine test setup and test plan. The static and dynamic test results are presented, and the impact of the results on the integration of high-pressure direct injection methanol engines is discussed.

2 METHODS

This section discusses the technology required to integrate a methanol-fuelled drive system consisting of the methanol fuel system technology (subsection 2.1) and the methanol engine technology selected for the engine to ensure transient performance (subsection 2.2). To evaluate the transient performance of the methanol-fuelled engine, this section also discusses the engine test setup (subsection 2.3) and the engine test plan (subsection 2.4).

2.1 Methanol fuel system

The Wärtsilä 32M engine is engineered for continuous operation on Methanol (MeOH), Marine Diesel Fuel (MDF), and Heavy Fuel Oil (HFO). When utilizing the Wärtsilä 32M, HFO can be employed exclusively as the main fuel. Methanol operation requires an external feed system to ensure proper fuel delivery and engine performance.

Figure 3 shows a typical methanol fuel system layout. Methanol is being led from service tank to high pressure Methanol Fuel Pump Unit (MFPU) using low pressure transfer pump(s). Between low pressure pump(s) and MFPU there are fuel valve trains (FVT) which are acting as safety device in case of emergency shutdown.

MFPU builds the necessary pressure for the methanol system. Each engine is fed by one high-pressure methanol pump, which includes a closed loop lubricating oil system, oil cooler, pressure and temperature sensors, and a leakage collection system. The pump is driven by a variable speed electric motor, allowing full rail pressure even at low loads. MFPU is installed in a dedicated equipment room, separate from the engine room. Methanol in pressure up to 600 bar is led through double wall piping from methanol equipment room to engine room and corresponding engine's leak and connection block.

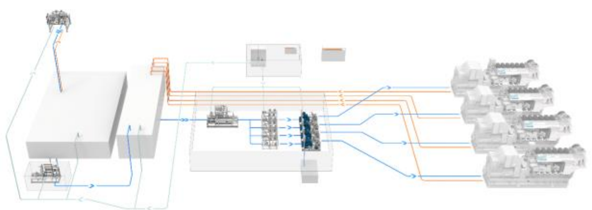

Figure 3. Typical methanol system layout (from the left MeOH bunkering station, low pressure pump, MeOH tanks, after low pressure transfer pump MFPU specific fuel valve train units and engine specific MFPU's. Above MFPU's nitrogen generator and below the MFPU's the drain tank for purging)

The sealing and control oil unit is providing sealing oil delivery pressure up to 700 bar and control oil delivery pressure up to 400 bar. This unit is typically installed in the engine room and includes a duplex filter before the pumps, ensuring the reliability and safety of the methanol injection system. Control oil is used to operate the methanol injection valves, while sealing oil prevents internal leak outs of methanol. The pressure of the sealing oil is always higher than the methanol pressure from MFPU. Both control and sealing oil are delivered through high-pressure double wall pipes and connected to corresponding engines leak and connection block.

From leak and connection block flexible high-pressure hoses are used for methanol, sealing oil, control oil, and methanol return lines, connecting the external system to the engine and allowing for flexible engine mounting.

Methanol is fed to the engine through a high-pressure double wall common rail pipe, which delivers methanol to each cylinder. The common rail pipe is connected to pressure accumulators, ensuring stable pressure during injection. Each cylinder is equipped with a twin fuel injection valve for diesel fuel and methanol injection. The methanol injection is controlled by an Electronic Rail Valve (ERV) located on top of the injection valve. When the ERV solenoid is activated, control oil activates the methanol needles. The Start-up and Safety Valve (SSV) is equipped with a solenoid-controlled valve to close the rail and release pressure. Control oil keeps the SSV closed during normal operation, and the rail pressure can be released through the valve. The SSV also features a mechanical safety valve that opens in case of overpressure [10].

2.2 Engine technology

While Liquid Natural Gas (LNG) can be utilized in pre-mixed Otto combustion, the lower octane rating of methanol may exacerbate challenges such as engine knock and pre-ignition [11]. These issues impose constraints on the required air-fuel ratio and may limit the maximum engine output, efficiency, and load-taking capability. To address these challenges and achieve full fuel flexibility, the pilot-ignited Direct Injection (DI) Diesel cycle was selected for the Wärtsilä 32M engine. This approach allows for optimized combustion control and improved engine performance, ensuring reliable operation across a range of fuel types.

With DI technology, the fuel injection pressure requirements are significantly higher compared to port fuel injection (PFI), as the fuel is injected during the compression stroke when cylinder pressure is already rising [12]. Wärtsilä 32M methanol fuel system uses a common-rail (CR) capable of producing 600 bar fuel pressure. For the diesel injection system, both the pilot fuel and the main fuel injection in back-up mode are performed using a conventional jerk-pump system. This configuration can offer certain benefits depending on the application for which the engine is used, as the availability and cost of different fuels vary significantly on a global scale. Additionally, this feature provides full redundancy in case the common-rail system requires maintenance.

While the complexity and cost of the fuel system increase with the pressure, there are several benefits to utilizing diffusive combustion with methanol. Firstly, it minimizes the possibility of methanol contaminating the lubrication oil, as the fuel is combusted while it is injected. Secondly, and more relevant for a dual-fuel engine, is that no compromise needs to be made with the engine's combustion geometry between methanol and backup diesel operation. The same valve timings, compression ratio, charge air temperature, and even turbocharger specifications are optimized for both fuel modes [13].

During dynamic load changes a PFI dual-fuel engine's loading performance on LNG is typically limited by the turbocharger's capability to produce the required air-excess ratio. It is often observed that the dynamic response of the engine at low loads is intuitively dismissed due to the operation on significantly leaner air-fuel mixtures. The slower response of the turbocharger at low loads can still cause knock if the load steps are fast enough, despite the engine operating on leaner mixtures [14].

Due to the compromise in turbocharger matching for both the Otto and Diesel combustion cycles, as well as variations in the methane number (MN), the typical loading process of a PFI dual-fuel engine from 0% to 100% load can be achieved in 5-6 incremental steps when using LNG. The stoichiometric air-fuel ratio of LNG and methanol are nearly identical when normalized for fuel energy content, presented in Table 1. However, due to the lower octane rating of methanol compared to LNG, a similar or slightly reduced loading performance can be anticipated when utilizing PFI technology. Additionally, the octane rating may require limiting the maximum output on methanol with Otto combustion cycle.

Table 1. Fuel properties [15]

Property	Unit	Methanol	LNG	Diesel
Molecular formula		CH_3OH	>90% CH_4	$C_nH_{1.8n}$ C_8-C_{20}
Carbon content	wt-%	37.5	75	87
Auto ignition	°C	464	540	240
Octane rating	RON/MON	109/89	120/120	-
Cetane number	-	5	-	45-55
LHV	MJ/kg	20	45	42
Maximum laminar burning velocity	cm/s	52	37	37
Stoichiometric air-fuel ratio	kg_{air}/MJ_{fuel}	0.33	0.34	0.34
Adiabatic flame temp.	°C	1910	1950	2100

In the context of the Wärtsilä 32M engine, which utilizes diffusive combustion and direct injection of methanol, the loading capability is primarily constrained by the capacity of the fuel injection system. Methanol, characterized by its high burning velocity, does not induce knocking in diffusive combustion. Due to these advantageous properties, the loading capability of the Wärtsilä 32M remains virtually identical when operating on either diesel or methanol. Specifically, loading from 0% to 100% can be achieved in three incremental steps when using diesel. The primary distinction when using methanol is the imposition of a minimum load limit of 10% [13].

Higher injection pressures can significantly enhance the load-taking characteristics when utilizing methanol by limiting the required injection duration. This is particularly critical because methanol necessitates higher delivery volumes compared to conventional hydrocarbon fuels due to its lower energy density. Furthermore, elevated injection pressures contribute to a favourable emissions profile during methanol combustion.

While the injection duration can also be modulated by employing a suitably large nozzle orifice, this method may adversely affect fuel atomization, especially towards the end of the injection process. The use of larger orifices can lead to increased emissions of unburnt and partially combusted hydrocarbons, which raises potential environmental and efficiency concerns.

2.3 Engine test setup

The test engine is equipped with various measurement sensors and devices to ensure accurate efficiency readings across the entire speed and load range. When operating on methanol, additional sensors are installed in the methanol, oil, exhaust, and crankcase ventilation systems to ensure all parameters remain within the limits specified by the manufacturer and regulatory standards.

Emission measurements are conducted using an FTIR (Fourier Transform Infrared) measurement device, which allows for the detection of a broader range of components compared to traditional devices typically used for diesel engines. Combustion analysis is continuously monitored using a combustion analysis tool called Dewesoft, which can integrate additional measurements as needed. The collected data can be further processed using other software programs if required.

The crankcase ventilation system is equipped with a new type of oil mist separator and filter to prevent oil mist from being released into the atmosphere. Additionally, the ventilation system includes an LEL (Lower Explosive Limit) measurement device to ensure that the LEL remains below a specified threshold. Depending on the engine setup, these crankcase gases can also be directed to the exhaust pipe.

2.4 Engine test plan

The stationary and transient performance may be evaluated by performing tests with the Wartsila 32M engine. The stationary tests are used to determine the basic characteristics within the engine envelope (different loads and speeds) such as the consumption/efficiency, emissions, and engine operational parameters in both diesel and methanol mode. The tests in diesel mode have been performed for 7 load points and in methanol mode for 10 load points as shown in Table 2. The test in both modes were performed for 5 speed setpoints as shown in Table 3.

The recovery time for the stationary engine tests between each load point was chosen at 120 seconds. While this is not sufficient time for the entire engine block to reach a temperature equilibrium, it was sufficient to determine the required information within an acceptable time frame.

Table 2. Stationary load points

Load points	Diesel mode	Methanol mode
5%	X	
10%		X
20%	X	X
30%		X
40%	X	X
50%		X
60%	X	X
70%		X
80%	X	X
90%		X
100% (nominal)	X	X
110% (overload)	X	

Table 3. Stationary speed setpoints

Speed setpoints	Diesel mode	Methanol mode
60%	X	X
70%	X	X
80%	X	X
90%	X	X
100% (nominal)	X	X

The transient tests are used to determine if the engine in methanol mode is able to cope with the dynamic power requirements required by the dredging process. These tests include determining the limitations which force a switch from methanol to diesel mode and the process of changing back from diesel to methanol mode. The behaviour of dredging equipment results both in cyclic load transients and in load ramps, therefore the transient tests have been performed with these 2 approaches in mind. All transient tests have been performed for the nominal engine speed.

The cyclic load tests consist of tests in which the load fluctuates between two load points at a certain rate for a number of repetitions, both including and excluding a recovery time. Figure 4 gives an example of a cyclic loading profile with and without recovery time. These tests are meant to show the effect of repeated load fluctuations on the response of the engine. The cyclic tests have been performed for both diesel and methanol mode. The diesel tests are used as a reference for comparing with the methanol transient performance and are used for the modelling of the engine behaviour at a later stage.

The ramp load tests consist of tests in which the engine is operating (stable) at a base load and is loaded as fast as the engine can handle to another load point to determine the load step/taking capability from each base load. Figure 5 shows examples of ramp load tests, both with an instant step and a gradual increase in load. The ramp load tests have been performed on methanol and a single diesel test to compare the results. These tests will be used for the modelling of the engine behaviour at a later stage and will be used to determine a static loading limit which will be used in a static analysis method as discussed in [16] (not in this paper). This method compares the load steps/ramps from measurement data to the

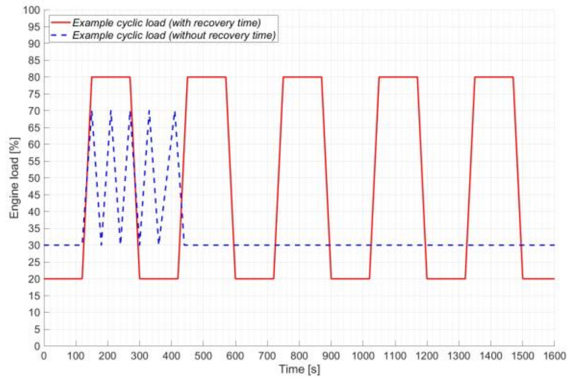

Figure 4. Example cyclic load tests with and without recovery time

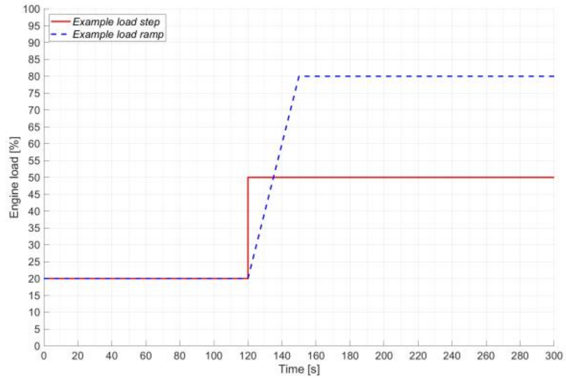

Figure 5. Example ramp load tests

limitations tested with the ramp load tests to determine if the engine is capable of performing the load changes in methanol mode.

3 RESULTS

This section discussed the results of the static (Section 3.1) and dynamic engine tests (Section 3.2). The impact of these results on the integration of the technology on the vessel will be discussed in Section 4.

3.1 Static engine performance

The static engine performance was tested according to the test as provided in Section 2.4. The engine on methanol has an operational envelope similar to the operation on diesel. The only exemptions are the overloading of the engine (>100%) and low load operation (<10%) which are only possible on diesel. Both result in an automatic switch to diesel mode. Switching back to methanol mode requires human intervention as well as the right circumstances (load criteria, methanol pressure, etc.). Ensuring the maximum utilisation of the methanol operation requires constant human control to prevent operation outside of the methanol operational limits and to switch the system back to methanol mode.

The operation of the engine in methanol mode increases the engine efficiency compared to the diesel mode of the engine over the entire load range at the nominal engine speed with 2-3 percent points (Figure 6). At its best efficiency point (at nominal speed), the engine has an efficiency of more than 45%. This increase in engine efficiency is a large benefit for vessel owners/operators. It reduces the fuel costs and increases the vessel autonomy which are both large concerns for renewably powered vessels. The engine is also capable of operating at a very high methanol energy ratio (MER) as shown in Figure 7, resulting in a higher renewable methanol percentage which may be used by the vessel owner, and which decreases the total greenhouse gas emissions of the vessel. The remaining energy percentage is provided by the diesel pilot.

The exhaust gas mass flow of the engine in methanol mode is similar to that of the diesel mode (Figure 8). This means that the methanol mode of the engine does not require a larger exhaust gas system to ensure a low enough back pressure for the engine to operate efficiently. Figure 9 shows that the exhaust gas temperature of the engine operating in methanol mode is about 20-30°C lower than in the diesel mode. The exhaust gas temperature has a decrease at 40% engine load for both diesel and methanol mode. This decrease is

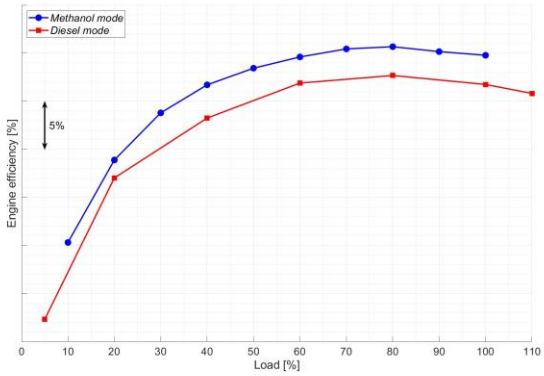

Figure 6. Engine efficiency at nominal speed

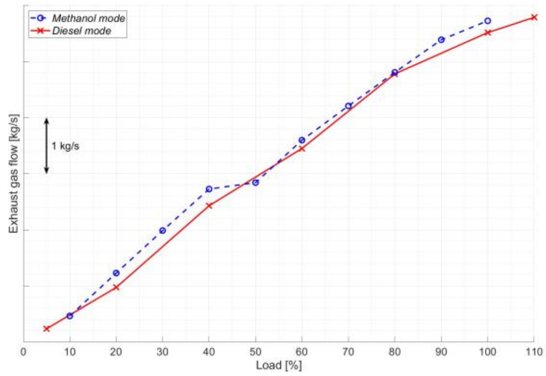

Figure 8. Exhaust gas mass flow at nominal speed

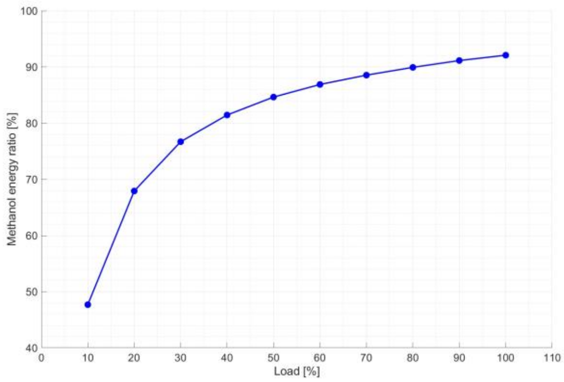

Figure 7. Methanol energy ratio at nominal speed

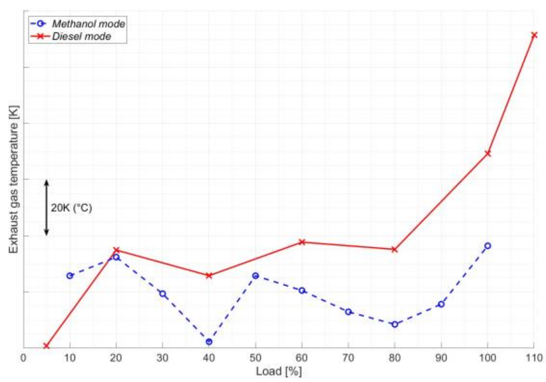

Figure 9. Exhaust gas temperature at nominal speed

especially clear for methanol and is in both cases the result of the variable inlet valve closure which changes at part load to ensure a higher fresh air intake in the cylinder.

Figure 10 shows that the NO_X emissions of the engine in methanol mode are substantially smaller than in diesel mode with about 40-60%. The exact methanol emissions for diesel and methanol mode are unavailable, however the total hydrocarbon (THC) emissions of the methanol mode are substantially higher than in the diesel mode (Figure 11). The formaldehyde (CH_2O) emissions of the engine in diesel mode are in the range from 3-7 ppm for diesel mode and 20-28 ppm for methanol mode.

3.2 Dynamic engine performance

The dynamic engine performance was tested according to the engine test plan presented in Section 2.4. This subsection presents the results of three engine tests, namely a cyclic load fluctuation with (subsubsection 3.2.1) and without recovery time (subsubsection 3.2.2) for both diesel and methanol and load step from 10% to 100% in 2 and 3 steps in methanol mode (subsubsection 3.2.3).

3.2.1 Cyclic load with recovery time

The cyclic load fluctuation with recovery time consists of a test in which the engine load is decreased in 11 seconds from 80% to 20% load, followed by a recovery time of 120 seconds, a load increase (20% to 80%) in 11 seconds and a recovery time of 120 seconds. This sequence is repeated 4 times as shown in Figure 12. Figure 13 shows the engine speed fluctuations caused by the load fluctuations. The engine in methanol mode is operating quite stable, the speed range varies from 729 to 768 rpm. This is a slightly wider range than for diesel (732-762 rpm), but within an acceptable range for the vessel integration.

3.2.2 Cyclic load without recovery time

The cyclic load fluctuation without recovery time consists of a test in which the engine load is decreased from 80% to 20% load and then increased from 20% to 80% load for 7 complete cycles as shown in Figure 14. The time in which each cycle (down and up) is completed in 23.5 seconds for the diesel operation and in 33 seconds for the methanol operation. Figure 15 shows that the diesel operation has a speed range from 732 to 762 rpm and the methanol operation from 729 to

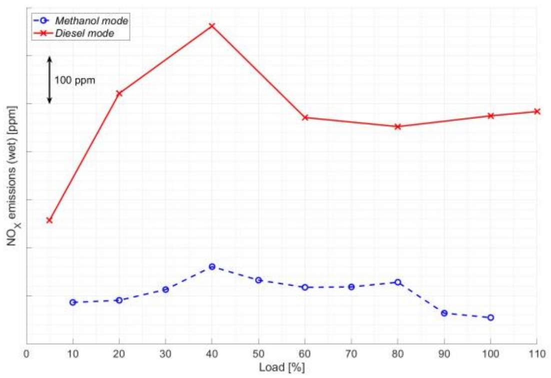

Figure 10. NO_X emissions in the (wet) exhaust gas

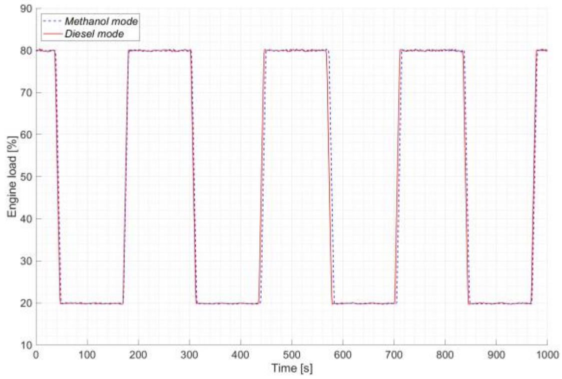

Figure 12. Cyclic engine loading 20%-80% in 11s with recovery time

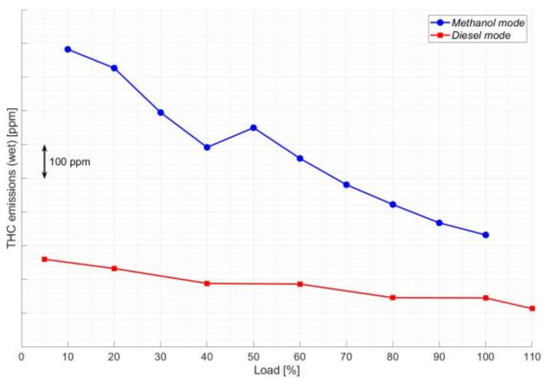

Figure 11. THC emissions in the (wet) exhaust gas

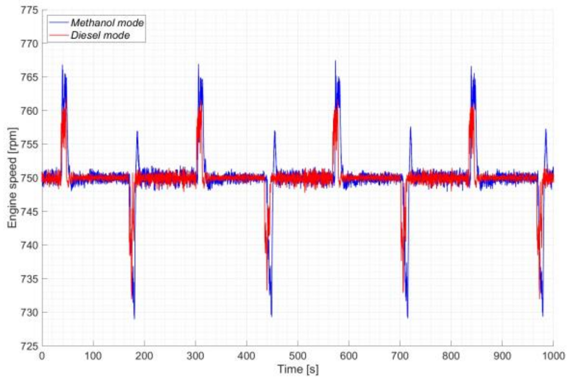

Figure 13. Engine speed for cyclic engine loading with recovery time

768 rpm. This is the exact same range as the cycling with the recovery time.

3.2.3 Load steps

The (instant) load step test from 10% to 100% load in 2 steps (10%-55%-100%) and 3 steps (10%-40%-70%-100%) are shown in Figure 16. The engine takes the step from 10 to 40% load without any issues in methanol mode and also the following steps from 40 to 70% and 70 to 100% load. Figure 17 shows the engine speed for both steps. The step from 10 to 40% load results in a speed drop to 717 rpm, a 4.4% drop in speed. This is well within the acceptable range. The subsequent steps from 40 to 70% and 70 to 100% load result in a drop to 728 rpm (a 3.0% drop) and 703 rpm (a 6.3% drop) respectively.

The load step from 10 to 55% load is not possible in methanol mode and requires the engine to switch to diesel mode to ensure the engine does not stall. The step from 55 to 100% load is no longer performed as the engine has switched to diesel mode (Figure 16). The 10 to 55% load step results in a total speed drop to 652 rpm which is a 13% drop in engine speed (Figure 17). This speed drop is not a problem for direct driven equipment, however if the engine is connected to a fixed speed shaft generator, the electric frequency would drop beyond the allowed range and result in a power outage. The engine switches from methanol to diesel mode at a speed drop of around 7.5% by strongly increasing the fuel rack position (Figure 18) and closing down the methanol flow (Figure 19). The moment the engine switches to diesel mode, the entire methanol injection system is turned off, despite Figure 19 showing some methanol injection after this point, but this is caused by setup of the measurement equipment.

The large speed drop from 10 to 55% load in methanol mode is a result of the methanol fuel supply system which is not able to increase the pressure fast enough to ensure sufficient methanol injection (Figure 20). This step may be possible if the methanol fuel pressure is increased before the load step occurs, but this requires further development of the control system and testing of the engine. The methanol pressure also drops during the steps from 10 to 40%, 40 to 70% and 70 to 100%, but for these steps the methanol pressure remains high enough to ensure sufficient methanol injection.

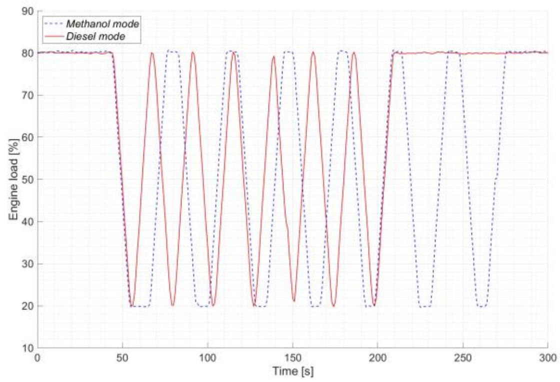

Figure 14. Cyclic engine loading 20%-80% without recovery time

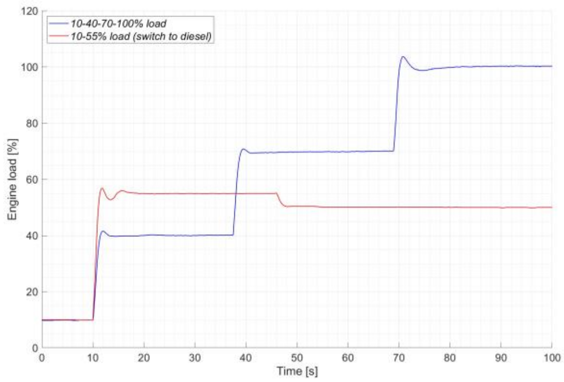

Figure 16. Engine load during the load steps

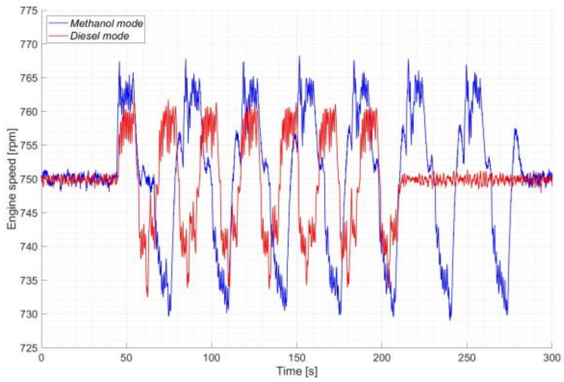

Figure 15. Engine speed for cyclic engine loading without recovery time

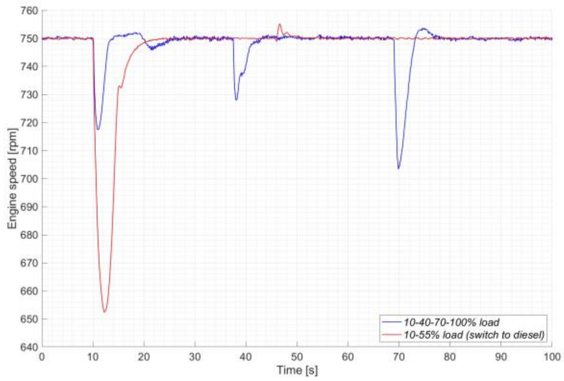

Figure 17. Engine speed during the load steps

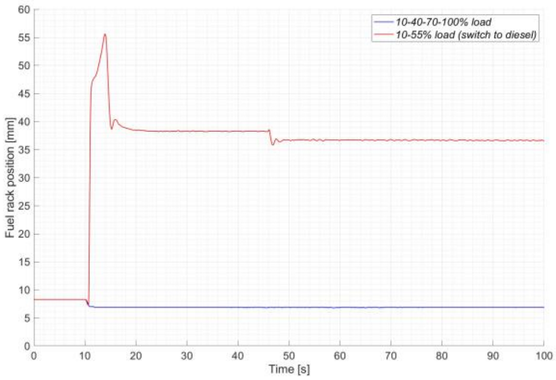

Figure 18. Fuel rack during the load steps

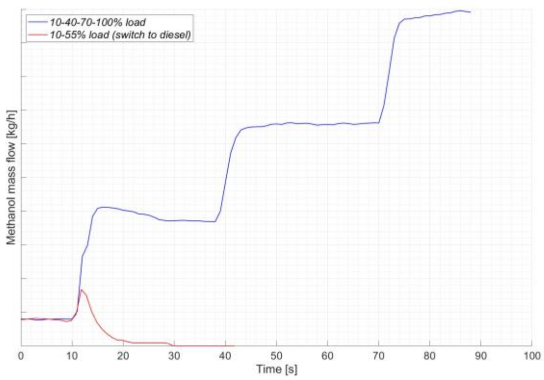

Figure 19. Methanol mass flow during the load steps

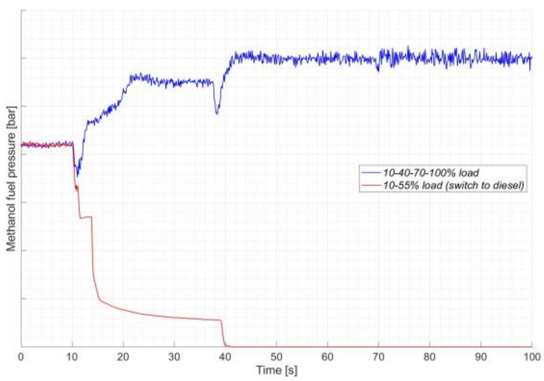

Figure 20. Methanol pressure during the load steps

4 DISCUSSION

The static engine performance results of the Wartsila 32M showed a fuel efficiency improvement in methanol mode of 2 to 3 percent points. This result is in line with data found in literature [17]. Karvounis et al. [17] looked at both port injection and direct injection of methanol with both computational fluid dynamics (CFD) and experimental data to analyse the impact of methanol on the engine performance. The study also showed that direct injection of methanol allows for high methanol energy ratio (>90%) compared to port injection with around 50%. This is a result of the more stable combustion process of the direct methanol injection.

The methanol engine has a reduced exhaust gas temperature (20-30°C) lower than in the diesel mode. This reduced temperature in combination with the methanol and (higher) formaldehyde emissions of the engine may have a significant impact on the exhaust gas aftertreatment of the engine. The lower exhaust gas temperature reduces the effectiveness of the selective catalytic reduction (SCR) system as the effectiveness of catalyst are temperature dependent and tend to be less effective at lower temperatures [18]. The formaldehyde and methanol emissions may produce hydrogen cyanide (HCN) in the SCR due to a reaction with the ammonia [19]. Additionally, the presence/formation of HCN reduces the SCR effectiveness with up to 20%. The HCN formation mechanism in SCR's is not entirely clear yet, but it may have a large impact on the SCR design and operation of methanol-fuelled engines. The integration of the engine with the exhaust gas (aftertreatment) system requires more investigation to ensure a save and effective implementation of methanol-fuelled engines and will require the SCR to be designed for both the diesel and the methanol operation. Potentially the exhaust gas system should include an oxidation catalyst before the SCR to reduce methanol and formaldehyde emissions to prevent the HCN problem.

The dynamic engine tests of the Wartsila 32M showed a promising transient performance of the engine in methanol mode. The engine speed deviation during the transient dynamic testing was limited to a narrow range of 729 to 768 rpm, only slightly wider range than for diesel operation (732-762 rpm). The cause of this difference in speed fluctuation range between diesel and methanol and the faster cycling of the diesel engine without the recovery time are part of further research, but likely a result of the methanol fuel supply system in combination with the control system. The direct injection of methanol greatly contributes to this transient performance as the combustion process is not limited by knocking and misfiring (as is the case for methanol-fuelled port injection engines). The methanol engine therefore has a much better transient response than port-fuelled LNG-DF engines which are limited by knocking and misfiring and the motor octane number of the fuel [7].

The methanol fuel supply and control system of the engine requires some further development. This system appears to be the cause of the switch from methanol to diesel mode during the large instant load step from 10-55% load. The engine did not manage to achieve this step in methanol mode and changed to diesel mode to prevent the engine from stalling. The tested load step is not expected to occur on a regular basis but may occur during for example the clutching in of large direct-driven mission equipment such as a dredge pump. Methanol engines are a new development and have not yet been applied in vessels with highly transient load profiles, but only in vessels with a more stable load profile. Thus, the engines, methanol supply system and the control system have not yet been optimized for this type of application and future operational experience and developments will likely improve the transient performance to equal that of the diesel engine.

5 CONCLUSIONS

The application of direct injection methanol-fuelled engines is a suitable option for dredging applications. These engines have the same operational range as diesel engines, an even better efficiency, less harmful emissions and a transient response similar to the diesel engines currently used in dredging vessels. The combination of these properties would allow for operating dredging vessels most of the time on methanol ensuring a strong reduction of fossil diesel fuel consumption.

The selection of direct injection of methanol is a strong contributor to achieve high methanol energy ratio, efficiency and power density. This also contributes to the transient performance of methanol-fuelled engines as the combustion process is not limited to the same narrow knock and misfiring limits common in port fuel injection methanol-fuelled engines. High pressure methanol injection does require some additional components on the fuel supply side such as the high-pressure methanol fuel pump and the sealing and control oil pumps. But on the integration side these additional components far outweigh the additional components required for port fuel injection engine such as energy storage systems and additional engines/cylinders to compensate for the reduced power density.

The occurrence of load steps from low to high load by clutching in a large piece of mission equipment should be prevented as this may trigger a switch from methanol to the diesel mode of the engine. This issue may be prevented by clutching in in a slower pace, but this will wear out the clutch faster. Additionally, an update to the engine control system may resolve this issue by instructing the engine to increase the methanol fuel rail pressure a head of the clutching in process.

6 RECOMMENDATIONS

The application of new fuels and the required prime movers in vessels does not only come with a large number of challenges on the vessel design side (fuel storage, safety, etc.), but also requires research and development to properly integrate the engine and fuel supply system. Engine testing is vital to ensure the system is able to deal with the operating conditions of the vessel and to determine the most suitable drive system design. This will prevent power outage issues during the operational lifetime of the vessel and ensure the largest possible reduction of fuel consumption and emissions.

7 ABBREVIATIONS

CFD	Computational fluid dynamics
CGH_2	Compressed gaseous hydrogen
CH_2O	Formaldehyde
CR	Common rail
DF	Dual fuel
DI	Direct injection
DME	Dimethyl ether
ERV	Electronic rail valve
EtOH	Ethanol
Fe	Iron powder
FTIR	Fourier transform infrared spectroscopy
FVT	Fuel valve train
GHG	Greenhouse gas
HC	Hydrocarbon
HCN	Hydrogen cyanide
HFO	Heavy fuel oil
IMO	International Maritime Organization
LEL	Lower explosion limit
LH_2	Liquefied hydrogen
LHV	Lower heating value
LNG	Liquefied natural gas
LOHC	Liquid organic hydrogen carrier
MDF	Marine diesel fuel
MeOH	Methanol
MER	Methanol energy ratio
MFPU	Methanol fuel pump unit
MGO	Marine gas oil

MH_2	Metal hydrides
MN	Methane number
MON	Motor octane number
$NaBH_4$	Sodium borohydride
NH_3	Ammonia
NO_X	Nitrogen oxides
PFI	Port fuel injection
PM	Particulate matter
RON	Research octane number
SCR	Selective catalytic reduction
SO_X	Sulphur oxides
SSV	Start-up and safety valve
THC	Total hydrocarbon

8 ACKNOWLEDGMENTS

Supported by the MENENS project (Methanol als Energiestap Naar Emissieloze Nederlandse Scheepvaart). Funded by the Netherlands Enterprise Agency (RVO: Rijksdienst voor Ondernemend Nederland) under the grant number MOB21012.

9 REFERENCES

[1] International Maritime Organisation. 2008. Amendments to the Annex of the Protocol of 1997 to Amend the International Convention for the Prevention of Pollution from Ships, 1973, as Modified by the Protocol of 1978 Relating Thereto. Revised MARPOL Annex VI. Resolution MEPC.176(58).

[2] International Maritime Organisation. 2023. 2023 IMO Strategy on Reduction of GHG Emissions from Ships. Resolution MEPC.377(80).

[3] Gilbert, P., Walsh, C., Traut, M., Kesieme, U., Pazouki, K. and Murphy, A. 2018. Assessment of full life-cycle air emissions of alternative shipping fuels, *Journal of Cleaner Production*, 172: 855-866. https://doi.org/10.1016/j.jclepro.2017.10.165.

[4] Brynolf, S., Taljegard, M., Grahn, M. and Hansson, J. 2018. Electrofuels for the transport sector: A review of production costs, *Renewable and Sustainable Energy Reviews*, 81: 1887–1905. https://doi.org/10.1016/j.rser.2017.05.288.

[5] Mestemaker, B.T.W., van den Heuvel, H.N. and Gonçalves Castro, M.B. 2020. Designing the zero emission vessels of the future: Technologic, economic and environmental aspects, *International Shipbuilding Progress*, 67(1): 5-31. https://doi.org/10.3233/ISP-190276.

[6] Georgescu, I., Stapersma, D., Nerheim, L.M. and Mestemaker, B.T.W. 2016. Characterisation of Large Gas and Dual-fuel Engines, *MTZ Industrial*, 6: 64–71. https://doi.org/10.1007/s40353-016-0029-z.

[7] Mestemaker, B.T.W., Goncalves Castro, M.B., van den Heuvel, H.N. and Visser, K. 2020. Dynamic simulation of a vessel drive system with dual fuel engines and energy storage, *Energy*, 194: 116792. https://doi.org/10.1016/j.energy.2019.116792.

[8] Shi, W. 2013. *Dynamics of energy system behaviour and emissions of trailing suction hopper dredgers [dissertation]*. VSSD, Delft, The Netherlands.

[9] Shi, W., Mestemaker, B.T.W., den Boer, L.J.A., Contessi, C. and Zotti, A. 2015. Application of LNG DF engines in dredging vessels, Proceedings of the *CEDA Dredging Days 2015*, Rotterdam, The Netherlands.

[10] Stojcevski, T., Jay, D. and Vicenzi, L. 2016. Operation experience of world's first methanol engine in a ferry installation, Proceedings of the *28th CIMAC World Congress 2016*, Helsinki, Finland.

[11] Daugas, C. and Chen, S.K. 2000. The empirical approach in the gas engine development, Proceedings of the 2000 ICE Fall technical conference, Peoria, Illinois, United States of America

[12] Heywood, J.B. 1988. *Internal Combustion Engine Fundamentals*, 2nd ed., McGraw-Hill Education, USA.

[13] Repo, J., Axelsson, M. and Heir, V. 2023. Methanol combustion concept alternatives for new build and retrofit of 4-stroke medium speed engines, Proceedings of the *30th CIMAC World Congress 2023*, Busan, South Korea.

[14] Georgescu, I., Stapersma, D. and Mestemaker, B.T.W. 2016. Dynamic Behaviour of Gas and Dual-Fuel Engines: Using Models and Simulations to Aid System Integration, Proceedings of the *28th CIMAC World Congress 2016*, Helsinki, Finland.

[15] Karvounis, P., Theotokatos, G., Vlaskos, I. and Hatziapostolou, A. 2023. Methanol Combustion Characteristics in Compression Ignition Engines: A

Critical Review, *Energies*, 16(24): 8069. https://doi.org/10.3390/en16248069.

[16] Mestemaker, B.T.W., van den Heuvel, H.N. and Gonçalves Castro, M.B. 2019. Energy storage system size estimation for hybrid dual fuel powered dredging vessels, Proceedings of the *CEDA Dredging Days 2019*, Rotterdam, The Netherlands.

[17] Karvounis, P., Theotokatos, G., Patil, C., Xiang, L. and Ding, Y. 2025. Parametric investigation of diesel–methanol dual-fuel marine engines with port and direct injection, *Fuel*, 381(B): 133441.
https://doi.org/10.1016/j.fuel.2024.133441.

[18] Koebel, M., Madia, G. and Elsener, M. 2002. Selective catalytic reduction of NO and NO_2 at low temperatures, *Catalysis Today*, 73(3-4): 239–247. https://doi.org/10.1016/S0920-5861(02)00006-8.

[19] Elsener, M., Nuguid, R.J.G., Kröcher, O. and Ferri, D. 2021. HCN production from formaldehyde during the selective catalytic reduction of NO_X with NH_3 over V_2O_5/WO_3-TiO_2, *Applied Catalysis B: Environmental*, 281: 119462. https://doi.org/10.1016/j.apcatb.2020.119462.