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Tailor-made solutions for high-pressure direct injection of methanol and ammonia

Fuel Injection & Gas Admission and Engine Components

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ABSTRACT

Direct high–pressure injection of liquid fuels such as methanol and ammonia is posed to be the main technology for future-proof marine engines. In comparison with low pressure injection in the intake manifold, with direct injection higher compression ratios can be achieved, increasing efficiency, and diesel pilot injection quantity can be minimised, maximising usage of the main fuel. Furthermore, the introduction of fuel near the end of the compression stroke minimizes the presence of unburnt fuel near the combustion chamber walls, limiting emissions and contamination of engine oil.

On the other hand, this requires the installation of two injection stages – i.e., for main and pilot fuels – in the already crammed space previously occupied by the diesel injector, and the need to adapt the diesel system to operate both in full power and pilot mode using the same injector. Additionally, different diesel injection technologies (i.e., pump-line-nozzle vs common rail) might already exist on a given engine platform, and injector designs need to adapt to each existing concept to minimise the investment required to upgrade the engine to full dual-fuel operation.

Recognising the need for such flexibility, OMT has built on the experience with its single fluid injectors for methanol and ammonia and designed a family of injectors that integrate in a single body both the diesel and the alternative fuel nozzles. Depending on the layout of the existing diesel system, the related injection stage can be laid out to be operated mechanically or through an electrohydraulic actuator. Moreover, different designs of the alternative fuel side are possible, mainly depending on which fluid is used in the pilot stage. The paper presents the resulting injector architectures and designs and discusses the advantages and disadvantages of each solution, as well as presenting a comparison of simulation and rig data to detail the performance obtainable from each solution.

Finally, results of engine tests with both methanol and ammonia are reported and discussed.

1 INTRODUCTION

In July 2023, the International Maritime Organization (IMO) significantly raised the standards for the marine industry by updating its strategy to reduce greenhouse gas (GHG) emissions from ships [1]. This revision introduced more ambitious targets, aiming for a 30% reduction in emissions by 2030, 80% by 2040, and finally net zero emissions by or around 2050. Given that ships typically have a lifespan of 30 years [2][3], this means that the global fleet responsible for meeting these targets will largely consist of vessels built within the next decade. This realization has prompted ship owners to carefully consider their future investments, and it has increased the pressure on vessel and engine designers to quickly develop commercially viable solutions that can operate on synthetic carbon-neutral fuels like methanol and ammonia [4].

Fuel injection has become a central figure in maritime decarbonization as a crucial enabler of future carbon-neutral fuel use. With a clear trend toward dual fuel engines in new ship orders, methanol claiming second place in new build fuel choices [5], and ammonia emerging as a promising long-term fuel [6], the adaptation of fuel injection systems has become critical to both immediate decarbonization measures and to the long-term net zero transformation of the industry.

Engines burning carbon-neutral fuels can operate either with lean-burn premixed combustion, where fuel is introduced at relatively low pressure (1-3 MPa) and mixed with air in the intake manifold, or with diffusive combustion, where the fuel is introduced at high pressure (typically 60-80 MPa) directly in the combustion chamber at the end of the compression stroke. In both cases, combustion needs to be triggered, typically via the injection of a small quantity of diesel fuel, which spontaneously combusts and ignites the main fuel mass. Engine operation with premixed combustion requires simpler and cheaper port fuel injection (PFI) systems [7] than those used for operating with diffusive combustion through high pressure direct injection (HPDI), as the latter requires more complicated injectors and an expensive high pressure pump. On the other hand, direct injection technology offers higher engine efficiency and a better fuel substitution ratio, because, compared to port fuel injection, it requires smaller pilot injections for stable methanol combustion as reported in detail in [8] and [9]. Additionally, the diffusive combustion approach limits the contact of unburnt fuel with cylinder liner walls, reducing issues related to engine oil contamination, which can lead to explosive conditions in the crankcase, as well as scuffing to the piston/liner due to oil film breakup.

To provide solutions able to fulfil all future IMO targets, as well as the related needs of shipping customers, engine makers will need to leverage both technologies. This will provide both: (i) quick upgrades to meet the short-term IMO targets, even if this means sacrificing some efficiency and GHG reduction, and (ii) longer-term solutions characterised by the highest efficiency and best utilization of carbon-neutral fuels.

While PFI technology and premixed combustion are ideally suited to address the first use case, HDPI technology is set to become the de-facto standard for new engine designs that need to deliver maximum efficiency when operating with fuels [10], while minimising the amount of pilot fuel used and the issues related with unburnt fuel emissions and engine oil dilution. On the other hand, in order to enable an engine to operate with diffusive combustion while switching back and forth between primary carbon neutral fuel (e.g. methanol or ammonia) and a diesel-like backup fuel, it is necessary to place one injector for each fuel into the same combustion chamber. This is a challenge, since a typical marine medium-speed engine only has one good place for locating injectors, i.e. in the centre of the combustion chamber, surrounded by the four engine valves. In such an engine design, two separate injectors would require too much space. For this reason, fuel system suppliers have started developing dual fuel injectors equipped with two nozzles, which can perform both primary and secondary fuel injection functions from within a single body.

OMT's integrated dual fuel injectors combine conventional and carbon-neutral fuel injection capabilities in a single unit, enabling engines to operate efficiently across different fuel types, while limiting the modifications required to adapt existing engine designs for dual fuel operation. This approach allows engine makers to provide shipping companies with both the flexibility to manage fluctuating and evolving fuel availability, and the confidence that their investments will remain viable throughout the energy transition, from now to 2050.

This paper presents OMT’s latest, integrated, high pressure dual fuel injectors, which were tailormade to fulfil specific customer needs, and differ for the diesel stage injection technology (mechanical vs. electronic) and the control fluid used to actuate the carbon-neutral fuel stage. The effect of such differences on injector performance and operation are presented and discussed. In addition, combustion tests performed on a single-cylinder engine with methanol and ammonia are presented and discussed.

2 DUAL FUEL INJECTOR DESIGN CHALLENGES

High-pressure direct injection of methanol and ammonia presents unique challenges [10]. These chemicals lack the natural lubricity of conventional marine fuels, requiring low friction coatings on all moving parts, and act as cleaning agents, removing protective oil films from steel surfaces, thus increasing corrosion risk. Additionally, their lower energy density compared to diesel requires larger injectors to be able to deliver more than twice the amount of fuel to yield the same power output, and their high ignition temperature requires a separate ignition source, such as a small diesel injection delivered by a dedicated nozzle, to initiate combustion [4]. Furthermore, methanol and ammonia are toxic, so class regulations require a gas-safe, sealed and alarmed environment [11] around all possible leakage points, including those inside the injector, thus requiring dedicated leakage collection lines to be integrated in its design. An additional challenge comes from the low boiling point of such fuels, which increases the risk of cavitation in the areas where the fluid reaches the highest velocities, e.g. spray holes, nozzle seat, control valve seat, and can lead to additional, localised thermal stresses due to the cooling effect of the evaporating fuel [10] that reduce the useful lifetime of the injector components.

Such challenges, and how they were addressed from a design point of view in a first, single-fluid, methanol and ammonia capable fuel injector were discussed extensively in [4]. In that case, a small, separate pilot injector was used to ignite the main fuel. However, any commercial engine must guarantee operability despite the current scarcity of carbon-neutral fuels, so a full-size diesel injector is also required to allow the engine to run on distillate fuels whenever the “greener option” is not available.

The main diesel injector is usually located on the cylinder axis, as shown in red in Figure 1 (left) as this makes use of the space among the intake and exhaust valves and it allows an optimal and fully axisymmetric fuel distribution in the combustion chamber. However, such space is not large enough to house also a dedicated injector for the carbon-neutral fuel next to the diesel one. One option would be to locate it as shown in green in Figure 1 (left), but this makes it difficult to achieve a uniform fuel distribution in the whole combustion chamber.

Furthermore, the interaction between main (green) and pilot (red) fuel jets would not be optimal with such configuration, possibly leading to the creation of several flame fronts and areas of unburnt fuel.

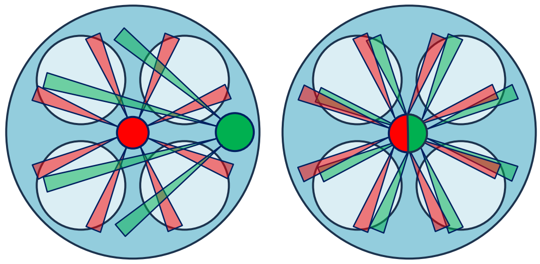

Figure 1. Possible location of diesel (red) and carbon-neutral (green) fuel injectors on a medium speed engine cylinder head.

It follows that, in order to retain a near-optimal fuel distribution and space utilization, a single dual fuel injector must be developed to incorporate, within a cylindrical envelope, two injection stages, one for diesel and one for the carbon-neutral fuel, as shown in Figure 1 (right), in red and green, respectively. In this case the center of the spray pattern is not perfectly lying on the cylinder axis anymore, but the deviation is small in relation to the combustion chamber radius. This approach simplifies the cylinder head design and improves combustion efficiency, but it complicates the design of the injector, which can no longer be made of axisymmetric parts, as it needs to host, side by side, two injection stages, as shown in Figure 2.

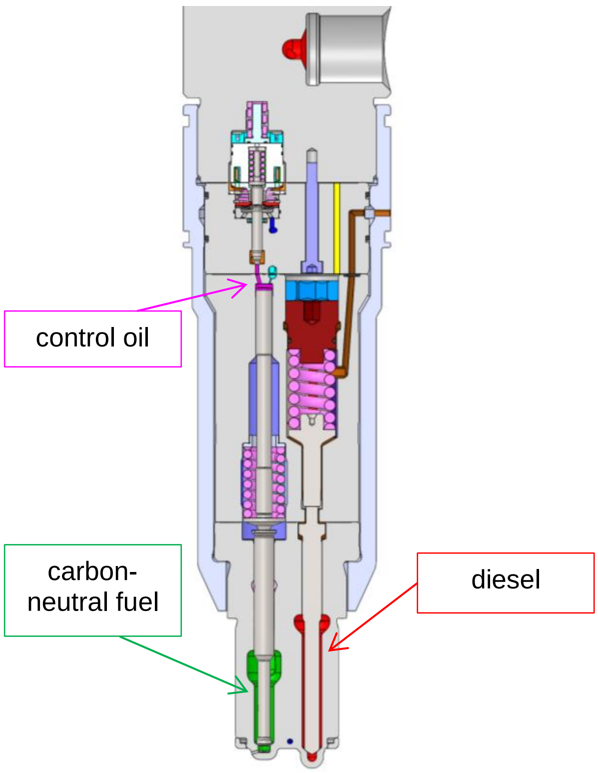

Figure 2. Cross section schematic of the nozzle area of a dual fuel injector, showing the presence of a nozzle for diesel (red) and one for the carbon-neutral fuel (green), controlled by a third fluid (magenta).

While most dual fuel injectors share this basic internal arrangement, the detailed architecture of each model is the result of the combination of the experience gathered during technology development [4][10] and the specific requirements of each customer. In particular, the choice of actuation fluid for the carbon-neutral fuel injection stage, and of the actuation technology to be used for the diesel stage, lead to different performance, combustion control possibilities and overall system complexity, as will be described in detail in the next sections.

2.1 Actuation fluid

In electronically controlled injectors for distillates, the fuel is used also as the pilot fluid, because its viscosity and boiling temperature are high enough to allow using it also as hydraulic fluid. This solution makes the injectors simpler, because they only have to handle one fluid under pressure and because the viscosity of the fuel is such that it sufficiently lubricates the guides and sliding parts without requiring any special arrangements.

Applying the same philosophy to an injector for alcohols or ammonia does not require particularly sophisticated arrangements, other than to treat the sliding surfaces with low-friction, wear-resistant coatings such as diamond-like carbon. The result is thus still a simple, sufficiently robust, and compact injector.

However, from a system integration point of view, a major complication comes from the fact that the low-pressure fuel return line must be kept at a high enough pressure to prevent vapor formation. In addition, since these fuels are toxic, such line must also be double-walled to safely detect and collect any potential leakage. On the other hand, however, to prevent the escape of fuel to the environment, it is sufficient to provide enclosures around the possible leakage points (sealing points) and to link them to the double-wall of the pipes connecting the injector to the rest of the circuit. This simple expedient allows, in addition to having a low-pressure sealed environment, to realise a circuit with an inlet and an outlet, enabling easy flushing and safer detection of possible leaks.

From a performance point of view, as shown by the simulation results reported in Figure 5, the injector controlled directly with fuel is characterised by good versatility of use, as the needle dynamics is little affected by the fuel supply pressure. As can be seen, a 50 bar pressure drop between one cylinder and its neighbour (dashed lines) would result in a negligible difference in opening and closing times, and the drop in injected mass (by about 5%) would be substantially attributable to the lower available injection pressure.

If, on the other hand, the pressure in the rail were to be lowered (e.g., to 600 bar, blue curves) to lengthen injection at part loads, there would be a slowdown in injector opening that would still be acceptable and would not affect its overall operation. The most delicate point in this injector architecture is the control valve which, operating with a much more volatile fluid, may be more susceptible to the risk of having erosion due to intense cavitation. Limiting this phenomenon requires careful study of the shape of the valve piston and its seat to ensure that any cavitation does not end up eroding key parts of the valve.

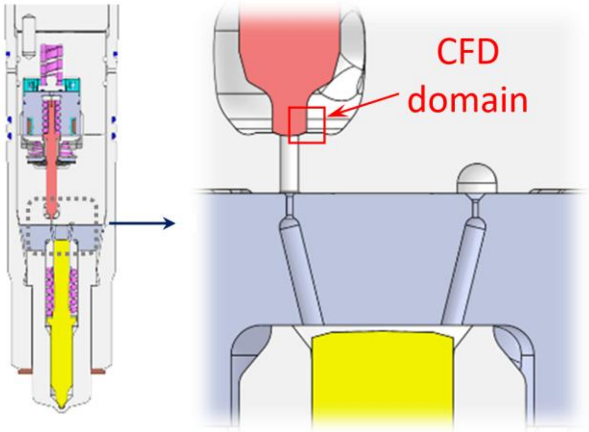

Figure 3. Cross section of piston and seat of a control valve for electronically controlled diesel injectors.

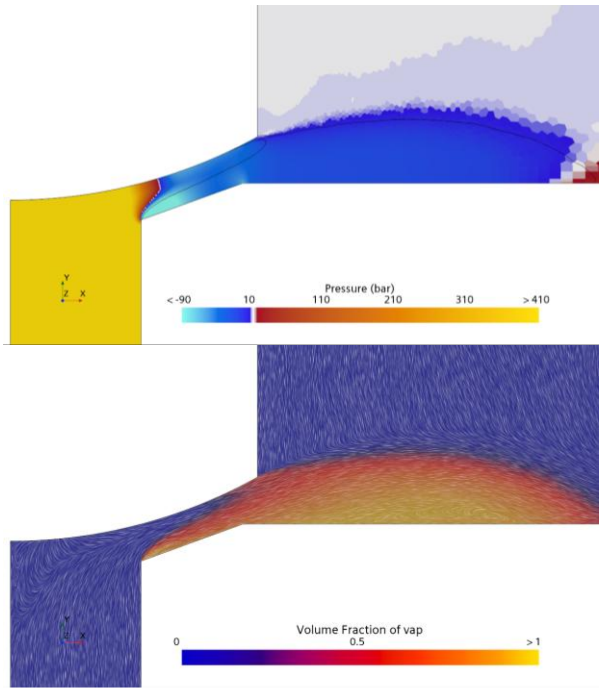

Figure 4. Absolute pressure and void fraction (cavitation) of methanol flow through a simple control valve with spherical piston tip and conical seat.

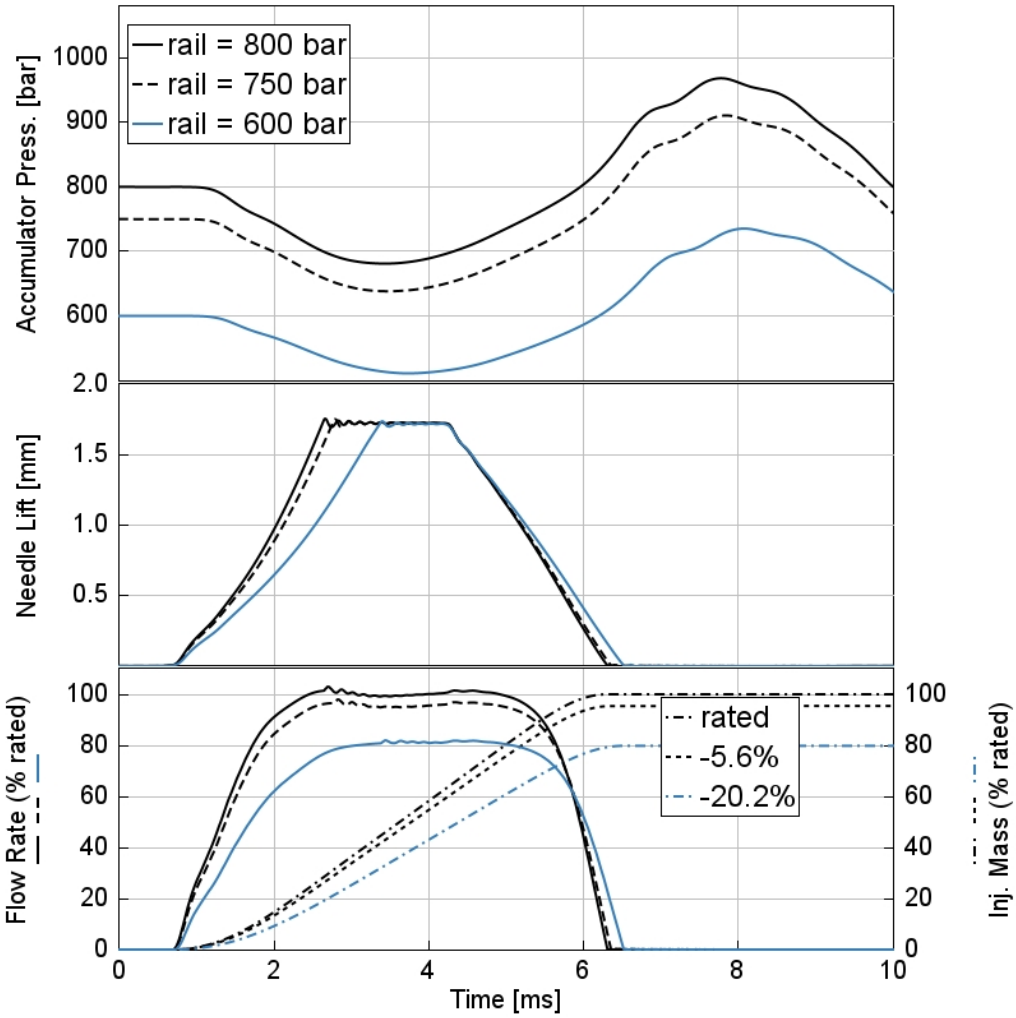

Figure 5. Simulated accumulator pressure, needle lift, injection flow rate and injected mass trends and their sensitivity to fuel pressure variations in an injector that uses fuel as control fluid.

As an example, Figure 3 reports the cross section of a control valve of an electronically controlled injector for distillate fuels. If this same valve were used to actuate an injector operating with methanol, even providing for a constant back-pressure (e.g. 10 bar) in the fuel return line to ensure that the fuel remained liquid, the dynamic effects that occur in the valve seat region would still produce an intense vapour formation as shown in Figure 4.

However, if, as a fuel sealing technique, a second (non-toxic) pressurized fluid is used as a barrier between the (toxic) fuel and the environment outside the injector, then it may be convenient to use the same barrier fluid also as a control medium to actuate the injector needle: its motion would then depend on the balance of the force generated by the pressure of the fuel (which drives the needle in the opening direction – in green in Figure 2) and the combined action of the spring and the force generated by the pressure of the control fluid (in magenta in Figure 2) that pushes the needle in the closing direction, i.e. towards its seat.

In this arrangement, the injector control valve operates with the control fluid, which is usually a lubrication oil with a high viscosity and boiling temperature, and this is advantageous for reliability and endurance because such physical properties reduce the fraction of vapor that can be generated near the valve seat and the energy associated with the subsequent implosion of bubbles (cavitation erosion). Thus, from this point of view, operating a control valve with hydraulic oil is even less challenging than using diesel fuel, and so existing and proven designs can be used without requiring extensive validation tests.

On the other hand, this architecture results in greater injector design complexity, because it requires the handling of two pressurized fluids, although it simplifies the handling of the return fluid from the control valve, which (being nontoxic) does not require a double-wall line.

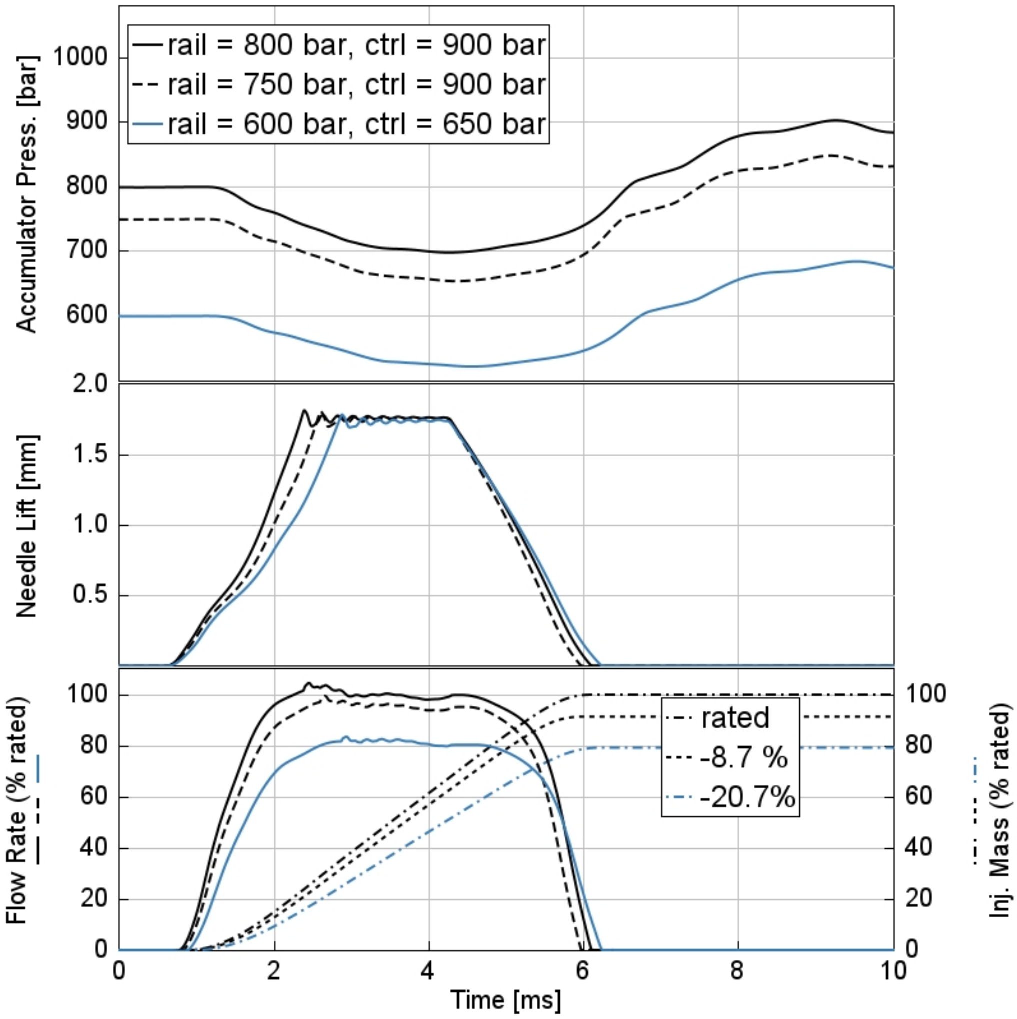

Figure 6. Simulated accumulator pressure, needle lift, injection flow rate and injected mass trends and their sensitivity to fuel pressure variations in an injector that uses a dedicated control fluid.

From an operational point of view, an injector that uses confinement fluid for control purposes requires a smaller control piston cross-section than the needle guide to keep the needle closed, because the sealing fluid is always kept at higher pressure than the fuel. It follows that, for the same needle lift, a smaller control volume is required. This feature, coupled with the larger flow rate that the control valve is able to discharge due to the higher upstream pressure, allows for faster actuation of the injector (as can be seen by comparing the needle lift pattern in Figure 5 and Figure 6). Conversely, any pressure fluctuation in the fuel supply line (generated by the opening of a nearby injector) tends to affect the behavior of the injector in a more significant way, because it would result in a fluctuation in the opening force but not in the closing force generated by the control fluid (which remains common to all injectors).

As can be seen from the trends shown in Figure 6, a decrease of 50 bar in fuel pressure would slow down the needle opening phase more markedly than found in Figure 5 above, and this would result in a greater reduction in injected mass (about 8%).

On the other hand, when operating at part load with low fuel pressures it is possible to compensate for some of the sluggishness of the needle by adapting the control fluid pressure and to obtain dynamics still substantially equivalent to those in high-pressure operation.

2.2 Diesel pilot injection technology

The diesel injection stage of a dual fuel injector is also subject to additional requirements, compared to an equivalent injector for an engine operating only with distillate fuels, because it needs to provide the injection quantity for achieving full engine power (actually up to 110% of the rated value) in diesel mode and also to be capable of repeatably delivering the small fuel quantities needed for piloting combustion ignition when carbon neutral fuels are used as the main source of energy. This requirement demands from the diesel injector exceptional flexibility of operation and very precise metering capabilities even when injection is very short (<5%).

In pilot operation, the main injector requirement is to create a spray that can reliably ignite the main fuel while using as little diesel fuel as possible. In other words, the distribution of diesel fuel produced by the pilot injector must be able to self-ignite and then heat the carbon-neutral fuel until it is evenly and repeatably ignited. However, the cross-sectional area of the diesel nozzle holes must be large enough to inject the amount of fuel needed to develop full engine power in an appropriate time (about 30°CA) and with limited ignition advance to avoid having too high combustion pressure (due to engine structural strength) and too high temperature (which could lead to excess NOx emissions)

To best meet these conflicting needs, it is convenient to take advantage of an electronically controlled injector because it allows the injector to operate at the most convenient fuel pressure, and to inject fuel at the most suitable time, independent of engine operating conditions. With such technology it is possible to dose and deliver a small amount of diesel at the right timing advance, so as to properly ignite the carbon-neutral fuel and, when switching to diesel operation from one cycle to the next, to achieve 100% diesel injection with a timing that optimizes the efficiency and pollutant emissions of the engine. However, this solution, which is very interesting from the point of view of combustion control flexibility, requires to install two common-rail systems on the engine (one for each fuel) and might present significant injector design challenges as it requires to package in it two control valves with associated solenoids, and might clash with cylinder head space constraints.

However, if the dual fuel injector is intended to retrofit an existing engine equipped with a PLN injection system, it might be convenient to use the existing fuel pumps and connect them to dual fuel injectors that integrate a mechanical diesel injection stage, i.e. using a spring-loaded nozzle needle. This requires less space in the injector than its electronically controlled equivalent, thus simplifying packaging issues. However, in this configuration it is more challenging to provide the needed flexibility in changing injection timing advance between pilot- and full- diesel injection and in controlling small injection quantities, as these features are limited to what can be achieved with a mechanical injection pump.

2.3 Nozzle cooling

In HFO injectors the nozzle is cooled to prevent overheating the seat and the consequent loss of hardness that would accelerate its wear. Cooling is normally implemented by flowing engine oil or water through an annular jacket realised in the front part of the nozzle, so that the heat coming from combustion can be removed.

In dual fuel injectors the cooling circuit is more complicated to implement because it needs to remove heat from both nozzles. In diesel mode, it is very important to cool the carbon-neutral fuel nozzle because, in absence of fuel flow, its seat would heat up and would experience thermal shock when operation with the carbon-neutral fuel commences. Additionally, when such nozzle is closed and heats up, it is important to avoid that the fuel trapped inside reaches critical conditions and changes phase from liquid to vapour. Conversely, when operating with the carbon-neutral fuel, it is important to cool down the diesel nozzle because the limited fuel flow through it would not be sufficient to remove the combustion heat flux through the nozzle.

3 OMT DUAL-FUEL HPDI INJECTORS

The need to rapidly develop dual fuel engines able to operate with high efficiency creates a high demand of dual fuel injectors designed to fit specific engine layouts, so that each engine modifications can be kept to a minimum. This fits very well with OMT’s philosophy of developing and offering products and solutions fully tailored to each engine maker’s needs. For example, while a new engine design might want to leverage the latest injection technologies to offer the best performance and flexibility, a retrofit of an existing engine will aim at minimising changes to the diesel injection system in order not to affect established performance with distillate fuels and to limit conversion costs. The choice of actuation fluid for the new fuel injection stage might be driven by reliability concerns and therefore fall on hydraulic oil, or cost and complexity considerations, and therefore lean towards fuel actuated injection stages.

Aware of the advantages and limitations of each solution, OMT is able to advise its customers on the impact of their engine layout choices and to design dual fuel injectors compliant with their technical specifications, leveraging its know how in technology and packaging it into tailor made solutions. In the next sections two design examples are reported, showing how different injection technologies are integrated into unique products capable of fulfilling customer requests.

3.1 Hybrid dual fuel injector

The injector described in this section was designed for engines delivering more than 1000 kW per cylinder and is laid out with a mechanical injection

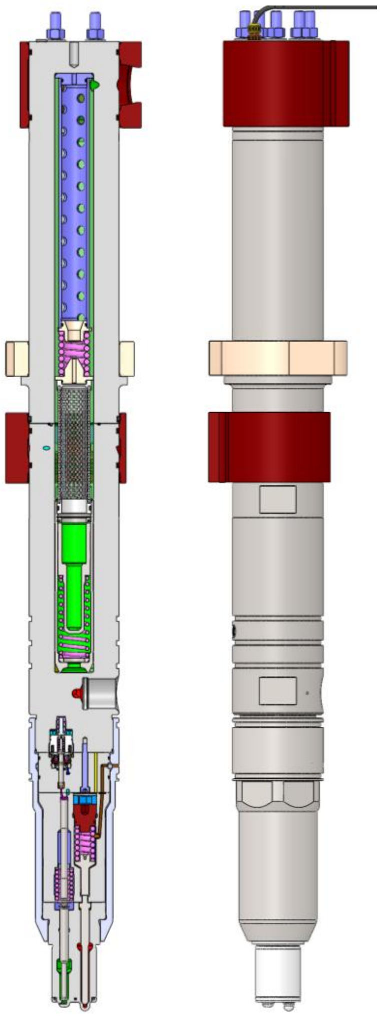

Figure 7. Hybrid dual fuel injector for retrofitting 1000+ kW/cyl. engines using PLN injection systems for the distillate fuel.

stage for the distillate fuel and an electronically controlled, hydraulic oil-actuated stage for the carbon-neutral fuel, as shown in Figure 7.

The design was driven by the customer goal of retrofitting its existing engine fleet to enable it to operate with methanol, as well as with MDO and HFO. The engines involved would already be operating in vessels, so high initial reliability, low conversion costs and no impact on performance in diesel mode were the main design drivers. For these reasons, the existing mechanical, spring-loaded diesel nozzle was packaged inside the nozzle body, taking care to minimise changes in fuel channel lengths and sizes, as well as needle and nozzle geometry. The same connection point to the pump to injector pipe was kept to avoid costly modifications to the cylinder head design.

The diesel injection stage was kept as compact as possible to maximise the space needed for the methanol injection stage, because, due to the lower volumetric heating value of carbon neutral fuels, larger quantities need to be injected per cycle and thus all the passages need to be sized for larger areas than a diesel injector for the same engine would require. On the methanol side, as common in modern electronic injectors for large medium speed engines, a fuel accumulator was integrated in the top part of the injector, where the methanol inlet pipe connection was also located. There was no need for a second methanol pipe connection on the injector to feed the adjacent ones, as the overall piping layout foresaw a methanol line running along the engine block with a T-junction outside each cylinder for distributing fuel to the related injector. The fuel accumulator placed at the injector inlet allows damping pressure oscillations in the supply lines to ensure that boundary conditions during injection are as similar as possible on all cylinders, to minimise fluctuations in injected mass from shot to shot as described in 2.1.

A last-chance filter was also integrated in the fuel accumulator to prevent stray metal particles, generated in the pump, or introduced when assembling the pipes, from reaching the injection stage and jamming moving parts, or preventing proper sealing if stuck between needle and seat. This is particularly important for low viscosity, volatile fluids like methanol and ammonia: a continuous leak due to improper sealing can rapidly damage the seat due to cavitation erosion. To prevent over fuelling of the combustion chamber in case of loss of tightness between injections, a flow limiter valve was also integrated in the bottom part of the fuel accumulator. This device was designed to shut off fuel supply to the nozzle in case the injected quantity exceeded by a specified margin the rated quantity injected during every combustion cycle. If this occurs, methanol can no longer be injected in the cylinder until fuel pressure is lowered below a reset level. In the meantime, that cylinder would only be providing the power associated with the diesel pilot injection, but the rest of the engine could continue to operate.

After passing the flow limiter valve, the fuel reaches the nozzle form which methanol is injected. In any dual fuel injector, care must be taken during the design phase to ensure that the jets delivered by one nozzle tip do not impinge the other one. Figure 8 shows how this was achieved: considering that diesel performance had to be preserved, it was decided to extend the diesel nozzle tip further in the combustion chamber than the methanol tip, so that all diesel spray holes could remain equally spaced as in the original diesel engine design. Conversely, the methanol nozzle spray pattern was designed with a larger angle between spray plumes in the direction of the diesel nozzle tip, so as to avoid impacting it.

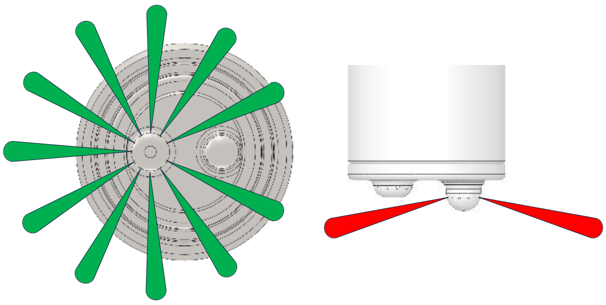

Figure 8. Detail of the dual fuel nozzle, showing the arrangement that prevents impingement of the other nozzle tip during injection: (left) wider gap between two jets in the methanol spray pattern, (right) further protrusion of the diesel nozzle tip.

To minimise potential reliability issues related to the operation of the control valve with methanol, as discussed in 2.1, it was decided to use SAE40 hydraulic oil for the actuation circuit. Figure 2 shows, on the left side of the injector cross section, how this was implemented. The control valve is located in the upper part and, through opening or closing a passage of hydraulic oil towards tank, modulates the pressure in the control volume located above the pushrod. The resulting force, plus the one generated by the spring, is transferred by the pushrod to the nozzle needle, acting in the direction of keeping it closed. Conversely, the fuel force acts on the nozzle needle to displace it upwards, i.e. in the opening direction.

The chamber where the spring is located is connected to the hydraulic oil tank, so that the leakages flowing through the gap between pushrod and its guide are collected and the spring chamber remains at atmospheric pressure. To prevent the fuel from leaking towards the spring chamber and polluting the hydraulic oil, a barrier was created by delivering pressurised hydraulic oil in a groove around the needle. As the oil pressure is always kept higher than the fuel pressure, a small oil leakage (due to the high oil viscosity) towards the nozzle and then the combustion chamber will occur, but no methanol would be able to reach the hydraulic oil return line. The drawback of this approach is represented by the introduction of an, albeit small, continuous consumption of hydraulic oil that is burnt with the fuel, and requires careful design of the sealing area to minimise such waste. In our case, the estimated oil consumption is less than 0.1 g/kWh and therefore lower than what is generally accepted as lubrication oil consumption in these types of engines.

3.2 Fully electronic dual fuel injector

The injector described in this section was designed for a rated cylinder output of more than 600 kW and integrates electronically controlled, fuel-actuated injection stages for both distillate and carbon-neutral fuels, as shown in Figure 9.

This design was the result of the customer desire to (i) maximise the possibility to efficiently control combustion in both diesel and carbon-neutral fuel operation, by taking advantage, for the diesel stage, of the existing common-rail injection system used on the single fuel engine, and (ii) to minimise engine cost by simplifying as much as possible the overall engine layout.

For these reasons it was decided to use the carbon-neutral fuel for controlling its own injection stage, rather than a third oil like in the injector design presented in 3.1, because this allowed to save the cost associated with the additional tank, pump and pipes needed for the control fluid supply circuit. On the other hand, such choice required, for safety reasons, the use of a double walled pipe for collecting the fuel spilled for control purposes and safely returning it to tank, as discussed in 2.1. In terms of injector architecture, accumulator, last-chance filter and flow limiter valve were integrated in the top part of the injector to provide pressure wave damping and operational safety on the carbon-neutral fuel line in a similar fashion as already described for the design shown in 3.1. On the other hand, these components were not included on the diesel fuel line because the existing injection system already provided them outside the cylinder head.

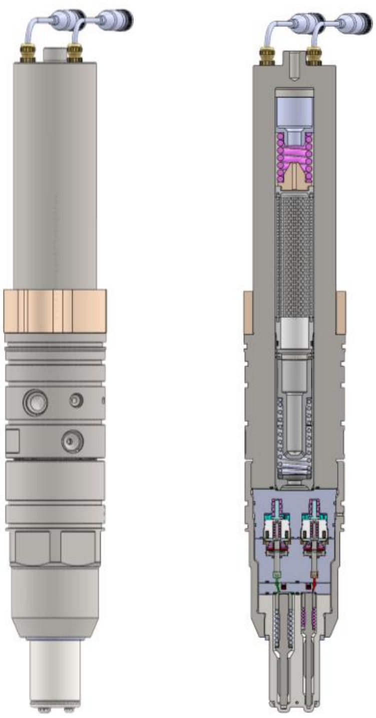

Figure 9. Fully electronic dual fuel injector that does not require an additional sealing/control fluid and is able to supply >600 kW/cyl. engines.

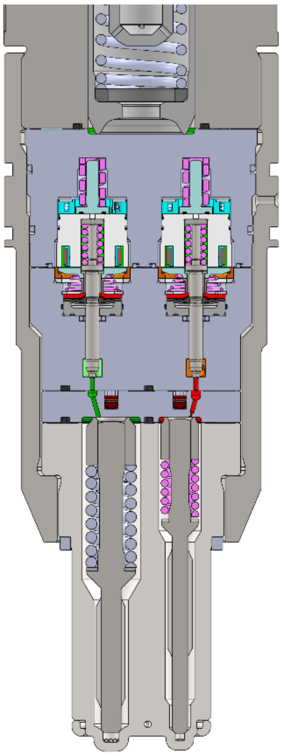

Figure 10. Detail of the fuel actuated injection stages of the full electronic dual fuel injector.

Using fuel accumulators far from the injector has been shown to negatively impact injection rate stability due to the relevant influence of the pressure wave propagations along the rail-to-injector pipe [12] but, considering that the engine performance in diesel mode was already proven and adequate, it was decided to keep the same diesel injection system layout also for the dual fuel version, which helped to reduce injector design complexity and cost, and to facilitate retrofit of existing engines. As per customer request, all the fluid interfaces were integrated in the cylinder head, leading to a compact design that favoured ease of maintenance of the engine, as no pipe needs to be disconnected from the injector to access the rocker arm and engine valve spring area.

As for the actuation, the design follows OMT's high performance, proven architecture that foresees to place the control valve very close to the nozzle, to minimise hydraulic delays and enhance controllability of very small injections such as the pilot shots that the diesel stage must deliver when operating with the carbon-neutral fuel. In this particular design, shown in Figure 10, two control valves are housed in the same block above an orifice plate that contains the calibrated holes that control the dynamics of each needle, and also integrates OMT's continuous monitoring and fault detection sensor [13] in both injection stages.

Hence, the customer could, at any time, request the monitoring system to be integrated with the engine, so as to keep under control any drift that could occur in injection performance and correct it via software changes or through condition-based maintenance. This is particularly important to avoid drifts in pilot injection quantity over time, as this would affect fuel consumption, emissions and engine performance.

The nozzle body integrates two fuel actuated injection stages, each designed according to best practices developed during an extensive research project [4], which include the best choice of materials, coatings and geometry to ensure to reliably deliver the expected performance, as discussed also in chapter 4. The spray arrangement was chosen in a similar way as shown in Figure 8 except that in this case the diesel nozzle could not be extended any further towards the combustion chamber, and so a slightly larger angle between spray holes was provided in the direction of the carbon-neutral fuel nozzle to prevent impingement.

4 ENGINE TEST RESULTS

This chapter presents the combustion performance results obtained with a single fluid injector technologically equivalent to the carbon-neutral fuel injection stage integrated in the injector presented in 3.2, but delivering a lower unit power. Due to cylinder head space constraints, it was not possible to use a dual fuel injector, and so a separate pilot injector was located close to the main injector and used to ignite the fuel. Even though these conditions do not fully reproduce the operation of a dual fuel injector, they nevertheless allow to study the combustion process of new fuels and derive important considerations that help in designing future-proof injection systems and engines.

In particular, the goal of the experimental investigations here presented was to assess the impact of diesel fuel fraction variations on engine performance when running with either methanol or ammonia, and either port or direct injection.

4.1 Single cylinder research engine

The medium-speed 4-stroke single cylinder research engine (SCE) used for this investigation had a displacement volume of approximately 15 litres, and had been modified for dual fuel operation. For the investigation of the diesel-methanol and the diesel-ammonia operation, a non-reentrant piston bowl and a compression ratio of 17:1 were chosen.

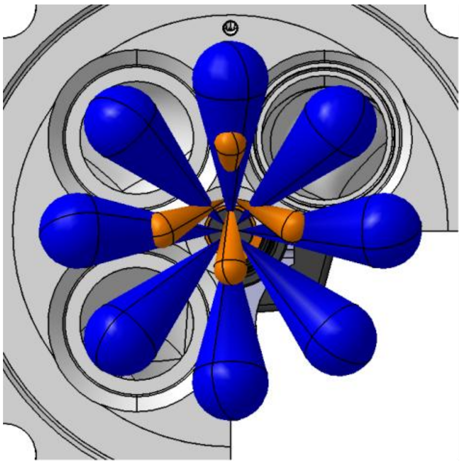

Figure 11. Illustration of diesel pilot (orange) and methanol or ammonia (blue) fuel jet interaction.

The low-swirl cylinder head was equipped with two intake and two exhaust valves. Exchanging the cam shaft lobes allowed a modification of the valve lift curves. Additionally, the valve timing could be adjusted individually for the intake and the exhaust valves. For this investigation, an intake valve lift profile with early closing before bottom dead centre was selected. The engine configuration is summarized in [4].

Instead of a turbocharger, an air compressor upstream of the engine and a flap in the engine exhaust system were used to adjust intake and exhaust manifold pressures. A flush mounted piezoelectric cylinder pressure transducer enabled real-time calculation of the indicated mean effective pressure of each cycle.

The out-of-centre position and the inclined orientation of the diesel injector nozzle in the combustion chamber required a special spray hole configuration. An illustration of the fuel jet interaction of the diesel spray and the methanol/ammonia spray is shown in Figure 11.

Additionally, the cylinder head was modified to provide two separate fuel return passages from the injector. While one of the fuel return streams was maintained at atmospheric pressure, the second stream was maintained at elevated pressure to avoid two-phase flow conditions in the injector control valve. In particular, this line was operated at 10 bar when using methanol operation and at 50 bar when using ammonia.

4.2 High pressure fuel supply and injection systems

The high-pressure fuel supply and injection systems used for operating the test engine are described in detail in [14]. One of the main challenges in designing the ammonia supply system was to ensure that it was maintained at a sufficiently high pressure to safely avoid ammonia evaporation and fulfil minimum inlet pressure requirements of the high-pressure fuel pump. During implementation, the highest standards were applied to the safety concept and material compatibility to ensure safe operation. Furthermore, a temperature controlled catalytic exhaust gas aftertreatment system ensured that no increased pollutant concentrations were emitted. Advanced sensorics for ammonia and nitrogen oxides were installed for pre- and post-catalyst monitoring and detailed exhaust gas specification was performed via FTIR spectrometer measurements.

Two independent high-pressure fuel systems were built for the diesel pilot and the renewable fuels injection. The diesel pilot injection system was capable of operating up to 1200 bar, and the pilot nozzle used had a nominal flow rate of 1.6 l/min. The pilot fuel flow rate was measured via an AVL Fuel Exact. The high-pressure fuel system for the renewable fuels included a pump designed for a maximum injection pressure of 1500 bar. The high-pressure fuel system also included the fuel conditioning, the fuel mass flow rate measurement and the actuators and controls to maintain the desired pressure in the injector leakage return line.

4.1 SCE measurement results

During the SCE investigations the key operating parameters, e.g. excess air ratio, diesel fraction, injection timing, were adjusted in order to have full load conditions (IMEP of about 24 bar) possibly with a centre of combustion (i.e. the point at which 50% of the combustion heat is released) around 10°CA, and an excess air ratio equal to 2.0

The start of the injections was also maintained with a dwell time between the diesel and the carbon-neutral fuel injection of 2.5 °CA. The longer carbon-neutral fuel injection for the lower diesel fraction resulted from the operation at fixed brake mean effective pressure. The excess air ratio was determined from the measured air and fuel mass flow rates and the stoichiometric air-to-fuel mass ratio for the selected share of diesel and carbon-neutral fuel. Adjustment of the excess air ratio was achieved via boost pressure adjustment. Exhaust gas pressure was adjusted to achieve a desired ratio of boost pressure to exhaust gas pressure. The impact of the diesel pilot fraction was investigated for two fuels (methanol and ammonia) and for the two different injection and combustion techniques: port fuel injection (PFI), with combustion by flame propagation in premixed charge, and direct injection (DI), with combustion by controlled diffusion by adjusting the duration of injection.

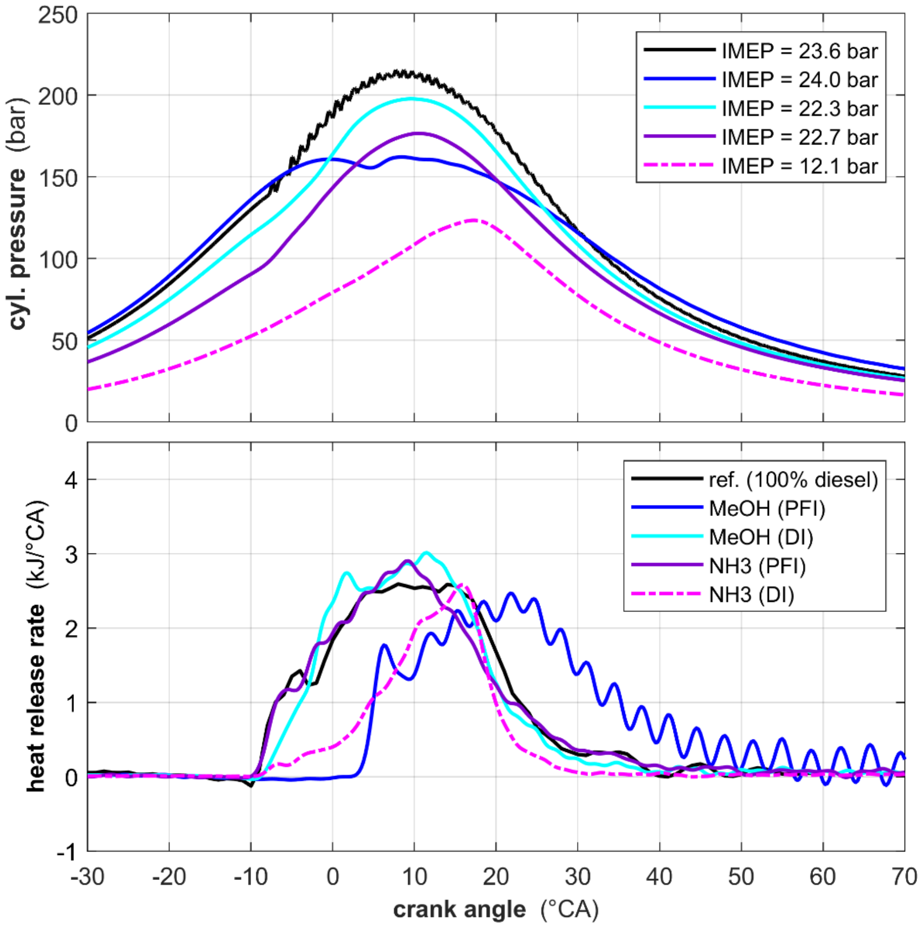

Figure 12. Cylinder pressure (top) and heat release rates (bottom) for:

- (black, reference) a pure diesel injection (EAR = 2.04, SoI = -14°CA);
- (blue) methanol port fuel injector operation with 30% diesel pilot (EAR = 2.3, SoI = +2°CA);
- (cyan) methanol direct injector operation with 4.6% diesel pilot (EAR = 2.03, SoI = -17°CA);
- (purple) ammonia port fuel injector + 5.2% diesel pilot (EAR = 1.4, SoI = -16°CA);
- (cyan) ammonia direct injector operation with 7.3% diesel pilot (EAR = 1.78, SoI = -17°CA);

Figure 12 shows the trends of the pressure measured in the combustion chamber (top) and of the heat release calculated from it (bottom) for the minimum quantity of diesel required to ignite the different fuels under different operating conditions.

Figure 13 and Figure 14 below show the effects of the fraction of diesel used as pilot on engine performance and pollutant emissions for different fuels and injection conditions.

Specifically, Figure 13 shows engine efficiency (ratio of useful work delivered over the sum of the energies provided by main + pilot fuels), combustion duration (calculated as the time required to release from 5% to 90% of the heat developed during combustion), cyclic work fluctuation (percentage CoV of IMEP), and effective dosage (shown as the ratio of the mass of air intake to the mass of stoichiometric air).

Figure 14 shows, on the top chart, the effect of the diesel pilot fraction on the amount of greenhouse gases emitted (evaluated as the amount of CO_2 equivalent), while the middle chart shows nitrogen oxides, in particular NO_x and N_2O, and any residual ammonia; and the bottom chart reports CO and unburnt hydrocarbon emissions (generated by the combustion of main and pilot fuel).

As a term of comparison, trends obtained with a 100% diesel injection operated at 1600 bar with an 8 x 0.33 mm spray hole nozzle are also shown in the same figures (black curves or symbols).

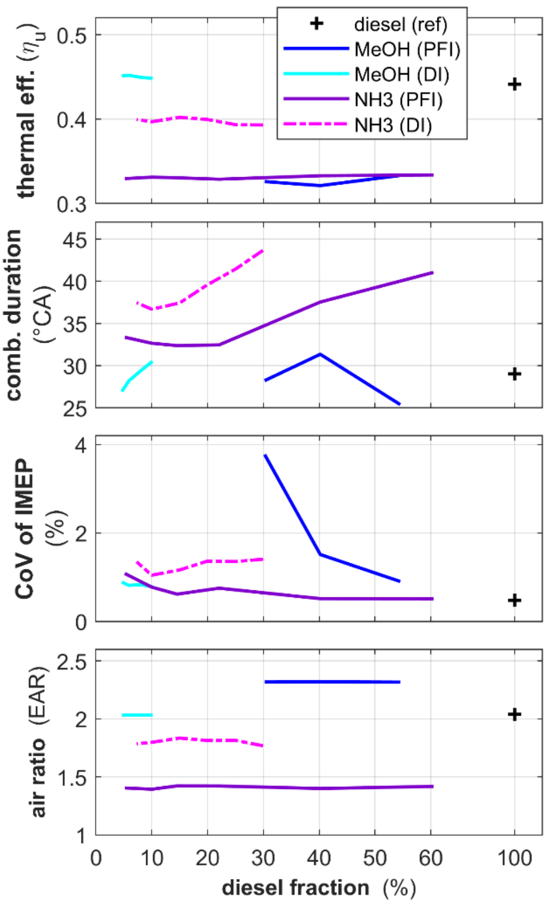

Figure 13. Engine efficiency as a function of the percentage of diesel used as a pilot (for ignition initiation) for different fuel conditions: methanol or ammonia injected directly into the cylinder or into the intake manifold.

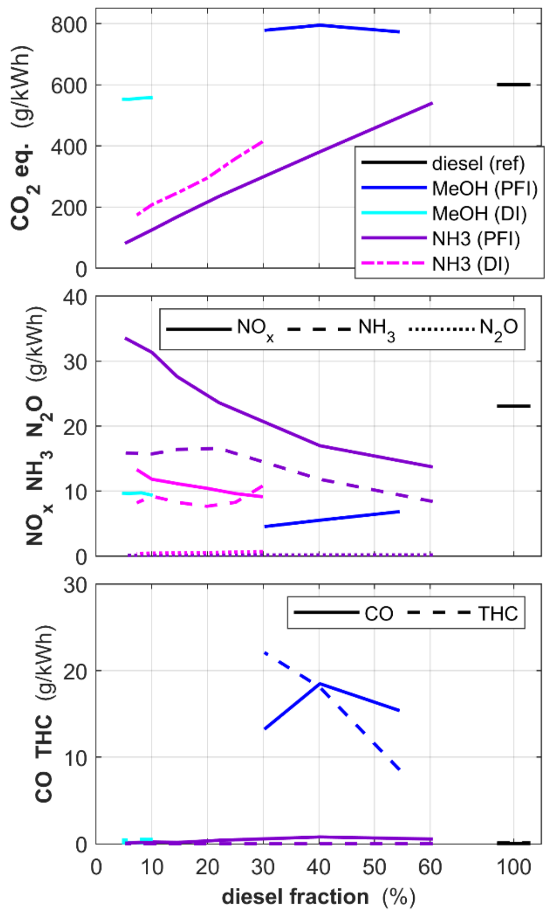

Figure 14. Pollutant emissions as a function of the percentage of diesel used as a pilot (for ignition initiation) for different fuel conditions: methanol or ammonia injected directly into the cylinder or into the intake manifold.

4.1.1 Methanol (DI)

As can be seen from the graphs in the previous figures, the combustion of methanol injected directly into the cylinder is the combustion method that comes closest to traditional diesel combustion, developing equivalent trends in terms of combustion evolution (Figure 12) and consequently showing substantial equivalence of performance and emissions as the diesel fraction varies (Figure 13 and Figure 14). Therefore, it can be said that, with this combustion technique, it is not advisable to use high diesel fractions (>10%); on the contrary, an increase in temperature produced by an increase in the diesel fraction seems not to accelerate the ignition of methanol, probably because the increase in the diesel fraction tends to reduce the concentration of the oxidant in the areas where there is overlapping of the two fuel sprays.

During the engine tests, the diesel fraction was reduced to less than 5% without encountering any problems and maintaining good cyclic stability (CoV < 1%). The efficiency measured under these conditions was equal to that of the reference diesel, with NO_x emissions more than halved, while the other substances measured were essentially negligible. Incidentally, it should be noted that CO_2 eq. emissions were actually slightly lower than those of the reference diesel, but since more than 90% were produced by the combustion of methanol, if methanol from renewable sources (bio-methanol, green-methanol, etc.) were used, the overall well-to-wake balance could in practice be very interesting (CO_2 equivalent < 10% ref).

4.1.2 Methanol (PFI)

Combustion of methanol by flame propagation achieved by mixing methanol with air in the intake manifold (port fuel injection) seems to be the easiest solution to implement, especially the least invasive on the cylinder head. Unfortunately, on the engine used in the tests, this technique showed severe limitations due to knocking. To achieve acceptable operation, the test conditions were modified by increasing the excess air by a further 30% (EAR = 2.3 instead of the reference EAR = 2.0) and delaying the ignition from the pilot by about 16°CA, with the result of moving the centre of combustion to about 19°CA (instead of 10°CA).

However, by using significant fractions of diesel (over 30%) and accepting a much greater cyclic fluctuation than usual (>3.7%), the engine still managed to develop 100% of the desired load, although the lower efficiency recorded ended up penalising CO_2 eq. emissions. In terms of emissions, this delayed combustion technique leads to lower maximum temperatures, resulting in significantly lower emissions of nitrogen oxides than the reference diesel (six times lower) and also almost half as much as the case with DI. Unfortunately, CO and unburnt hydrocarbon emissions were significantly higher.

4.1.3 Ammonia (PFI)

The combustion of ammonia by flame propagation, achieved by mixing it with air in the intake manifold (port fuel injection), presented opposite characteristics and requirements to those seen for methanol PFI. Probably, the fact that ammonia is completely gaseous in the intake manifold ensures good and uniform mixing with the air and this helps to yield a more regular cyclic behaviour (CoV ~ 1%).

On the other hand, however, the air/ammonia mixture tends to burn much more slowly, and in order to have a heat release trend close to that of ref. diesel (Figure 12), it is necessary to use a much richer dosage (EAR = 1.4, instead of the reference EAR = 2.0). In spite of this, flame propagation is slower than with other fuels (methanol), combustion tends to last longer, and efficiency is also severely penalised in this case.

Regarding the effect of the pilot diesel fraction, while using a very small fraction (<10%) still allowed good ignition with low CoV (~1%) and ensured low CO_2 emissions, high NO_x emissions (greater than the diesel reference) and also a significant amount of residual unburnt ammonia were found. Increasing the fraction of pilot diesel (up to 60 per cent), and consequently depleting the air/ammonia mixture, yielded benefits in terms of NOx emissions, and reduced residual NH3 in the exhaust, but inevitably resulted in progressively higher CO_2 emissions. Moreover, an important quantity of diesel injected into an air-poor chamber (because of low EAR and the presence of NH_3) led to a progressive increase in CO, a symptom of imperfect diesel combustion.

4.1.4 Ammonia (DI)

Due to problems with the test apparatus, it was not possible to supply the engine with the full load quantity of ammonia fuel, hence it was not possible to test it in comparable conditions to those presented for the other fuel and combustion strategies, because only about 50% of the rated power was reached in this configuration. Nevertheless, it was deemed interesting to report (with dashed-dotted line) the results recorded in this test in order to highlight important aspects related to this technology. In particular, even with direct injection, ammonia tends to burn with difficulty and develop a higher CoV than other fuels (> 1%), despite using a richer dosage than the reference (EAR = 1.7).

In any case, although the slow combustion of ammonia helps to limit NO_x (about half as much as diesel ref., i.e. about as much as 100% MeOH), there remain significant amounts of residual NH_3 in the exhaust and also limited fractions of N_2O which (having a much higher greenhouse effect than CO_2) weighs heavily on CO_2 equivalent emissions, being worse than the PFI solution. Again, increasing the diesel fraction leads to a progressive increase in the temperature in the spray (which should accelerate the combustion of ammonia) but also leads to a progressive depletion of the oxygen available for ammonia combustion, with the result that overall, an increase in the diesel fraction tends to lengthen the combustion duration and slightly worsen engine efficiency.

5 CONCLUSIONS

This paper presents engine test results that have allowed OMT to evaluate the potential and critical aspects of injection technology (DI vs. PFI) and related combustion processes of methanol and ammonia. In this study we found that:

Direct injection of methanol allowed stable diffusive combustion with a low quantity of diesel pilot injection (5%) yielding very good efficiency, comparable to diesel operation, and less than half of the NO_x emissions of diesel operation. Using methanol from renewable sources, this technology would enable the reduction of fossil CO_2 emissions by more than 90%, when compared to a modern diesel engine delivering the same power output, where the remaining emissions originate from the pilot fuel. Hence, further fossil CO_2 emission reductions could be achieved using biodiesel for this function. By minimizing the quantity of pilot fuel needed to initiate methanol combustion, direct injection represents the most promising technology to reach the net-zero emission target, considering that the global supply of biodiesel is extremely small, with limited potential for scaleup.

Premixed combustion of methanol realised through port injection showed considerable knocking issues, which can be limited only by using a higher fraction of diesel pilot fuel (>30%) and an excess air ratio (EAR > 2.3), a combination that increases fuel consumption (+38% compared to diffusive combustion). It was hypothesised that this was partly caused by incomplete fuel evaporation in the intake manifold, due to the high latent heat of the fuel, which caused spots of richer mixture in the combustion chamber. This can be improved somewhat with further engine tuning and refined port injector design. While this combustion concept offers a promising solution for rapidly and economically retrofitting existing engine fleets to reach the first IMO GHG reduction targets in 2030, it is not as interesting for the long term as high-pressure injection and diffusive combustion.

Conversely, premixed combustion of ammonia resulted in a more uniform air-fuel mixture, achieved by introducing ammonia in gaseous state into the intake manifold and because of the slower burn rate of ammonia, which inherently reduces the chances of knocking. In fact, the tests showed that, to improve the combustion speed and the rate of heat release, it was necessary to significantly reduce EAR down to 1.4. Engine efficiency was low (comparable with methanol PFI), and emissions of NO_x and NH_3 were higher. This can be mitigated by increasing the fraction of diesel pilot energy, but this leads to increased CO_2 emissions.

Direct injection and diffusive combustion of ammonia could only be partially explored in this study, because of the limitations of the laboratory fuel supply system used to feed the injector on the engine test rig. This highlighted the complexity of reliably handling ammonia, in terms of pressurisation, safe confinement, and cavitation erosion on fuel system parts. Nevertheless, the experience gained in the process proved valuable in developing and maturing OMT's injection technology for new fuels.

This study showed that operating marine engines with new fuels requires the development of new combustion concepts, and that the pros and cons of each solution will likely lead to a variegated landscape, where the most convenient fuel and injection technology will be adopted for each use case. For these reasons, OMT is actively developing tailormade solutions for DI and PFI of methanol and ammonia, to be able to support its customers in their engine development efforts.

In this paper, focus was placed on direct injection, and specifically on two different products that were developed using different technological solutions. These had the common goal of delivering, through the same injector, both methanol/ammonia and the diesel fuel needed to ignite them, while at the same time providing full diesel mode operation capability. This led, for example, to the challenge of placing the two fuel nozzles as close as possible to each other, to ensure a good interaction between pilot and main fuel, in order to yield reliable ignition, while, at the same time, preventing the jets of one fuel from hitting the nozzle of the other fuel, because this would lead to significant unburnt fuel emissions.

The design of these dual fuel injectors was driven by customer specifications and led to the adoption of different actuation concepts for carbon-neutral fuel nozzles. In one case, for maximum initial reliability, a hydraulic oil was used for control and sealing purposes, to avoid the provision of a pressurised, double walled, fuel return line, and also to avoid operating the control valve with highly cavitating fluids, such as methanol and ammonia. In a second case, a product was developed for a new engine, focusing on layout simplicity and reduced cost. This employed a single fluid actuation concept and a design that did not need a sealing oil, which further eliminated possible fuel/oil compatibility issues. However, it required a dedicated design for limiting cavitation in the valve.

The impact on injection performance of the two actuation strategies (i.e. control oil vs. single fluid) was investigated. It was shown that these different concepts lead to different sensitivities to fuel pressure fluctuations in the supply line. An injected mass reduction of 5% was calculated for a pressure reduction of 50 bar when using fuel as actuation fluid. It was increased to 8% for separate control oil. On the other hand, when operating at partial loads, where fuel pressure is reduced to prolong injection duration, in order to yield optimal heat release, actuation with control oil offers the possibility of modulating its pressure to increase injector responsiveness. This yields better performance than the fuel actuated injector would display in the same conditions.

In conclusion, an injector controlled using the sealing oil could present a larger dispersion of operation between cylinders. On the other hand, the flexibility, of being able to apply a different pressure to fuel vs. control fluid the allows better injector operation performance to be maintained over a wider range of injection pressures.

6 DEFINITIONS, ACRONYMS, ABBREVIATIONS

°CA: Crank angle degrees

CFD: Computational Fluid Dynamics

CO: Carbon monoxide

CO_2: Carbon dioxide

CoV: Coefficient of Variation

DI: Direct injection

EAR: Excess air ratio

FTIR: Fourier transform infrared spectroscopy

GHG: Greenhouse gas

HFO: Heavy fuel oil

HPDI: High pressure direct injection

IMEP: Indicated mean effective pressure

IMO: International Maritime Organization

MDO: Marine diesel oil

MeOH: Methanol

N_2O: Nitrous oxide

NH_3: Ammonia

NO_x: Nitrogen oxides

PFI: Port fuel injection

PLN: Pump line nozzle (injection system)

SCE: Single cylinder engine

SoI: Start of injection

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