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A 3D – CFD Analysis of Gas Direct Injection for a Dual – Fuel Medium Speed Marine Engine

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ABSTRACT

The University of Rostock operates one of the largest single-cylinder dual fuel marine engines of its kind in Europe, for which a novel gas direct injector for methane has been developed. Using medium pressure direct injection (MPDI) instead of port fuel injection (PFI) offers a high potential for the minimisation of methane slip while improving efficiency and power density at lower cost than high pressure direct injection. Analysis using three-dimensional computational fluid dynamics (3D – CFD) simulation was carried out in order to better understand the mixture formation process within the cylinder, as knowledge of the interaction of the gas jet and air motion within the cylinder is crucial to designing a successful combustion regime. Different possibilities for the use of MPDI are shown, covering homogenous and inhomogeneous operation, analysed using 3D – CFD.

Keywords: Gas direct injection, methane slip, LNG, dual fuel, medium speed engine, 3D-CFD Simulation.

1 INTRODUCTION

The growth of the world shipping fleet and the need for stringent regulation of marine emissions in order to meet climate goals requires the use of novel marine engine technologies. One established modern technology is the dual-fuel methane/diesel engine, which uses methane as its main energy source and around five energy percent of diesel to ignite the methane. Due to methane's high H/C ratio, this results in a saving of around 25 % carbon dioxide, e.g. Sremec et al. (2017).
Four-stroke turbocharged methane/diesel engines often suffer from methane slip- the escape of unburned methane through the engine's exhaust. Methane has a global warming potential of 29.8 over a hundred year period, see IPCC (2024). In order for these engines to fully unfold their climate friendly potential, this issue must be solved using technological solutions.
One such solution is the use of a methane direct injector, which eliminates methane slip occurring from valve overlap. To this end, a novel methane direct injector has been developed for a single cylinder dual-fuel marine engine in the TEME2030+ project at the University of Rostock.
In order to characterise the behaviour of the injector and identify potential injection strategies, a 3D-CFD model of the engine cylinder and injector was created. This allows the analysis of gas mixing during the injection and compression strokes.
Following a brief overview of the engine, injector and CFD model, this paper presents an investigation into the effect of variation of the start of injection in the intake and compression strokes on the gas injection itself and on the following homogenisation process.

2 Experimental Setup

The engine and direct gas injector which form the basis for the 3D-CFD modelling are briefly presented here.

2.1. Engine and Test Bed

With its 340 mm bore, 460 mm stroke and power output greater than 500 kW, the University of Rostock operates one of the largest single cylinder research engines of its kind in Europe. Multiple cylinder versions of these engines are used onboard ships for power generation. The single cylinder engine is run on the generator curve at a rated speed of 720 min^{-1}.

The engine test bed offers a high level of flexibility for various engine tests, including:

- Charge air pressure up to 8.5 bar (dry air)
- Humidification of charge air
- Adjustable engine back pressure to simulate turbocharging
- An EGR system (30 % EGR rate at full load)
- Canning for experimental methane oxidation catalysts
- Detailed exhaust gas measurement (FTIR Sesam i60)
- Gas admixtures of propane, carbon dioxide and hydrogen to the main natural gas fuel
- Medium pressure (up to 80 bar) and high pressure (up to 600 bar) fuel gas supply
- Solenoid operated gas admission valve (SOGAV) for port fuel injection (PFI) at up to 10 bar
- Gas direct injector (up to 80 bar for medium pressure, up to 600 bar for high pressure direct injection)
- Diesel pilot injection up to 2200 bar
- Fully programmable research ECU.

2.2 Gas Direct Injector

Serial production engines of this type rely on SOGAV gas injection into the intake manifold, which has proved itself as a safe and reliable technology. The disadvantage of this strategy is the potential for methane slip during valve overlap. A portion of fuel remains in the intake manifold until the next cycle and escapes through the exhaust while intake and exhaust valves are simultaneously opened, which is necessary to improve the scavenging process and cool the engine components. To avoid these fuel losses, a novel gas direct injector was developed for the engine. This outward opening poppet valve type injector injects the methane fuel directly into the combustion chamber at injection pressures up to 80 bar after the exhaust valve closes, eliminating methane slip during valve overlap. The nominal injection duration is 30° crank angle (CA), according to the injector specification

It is possible to operate the injector with a gas jet forming nozzle, creating a narrow jet of fuel within the combustion chamber. The following CFD investigations were carried out in order to characterise this injector's behaviour when used on the experimental engine without the jet-forming nozzle. The next section details the numerical models employed for this simulation.

3 Numerical Model

A full cylinder mesh including intake and exhaust ports was created using AVL FAME software, as seen in Fig. 1.

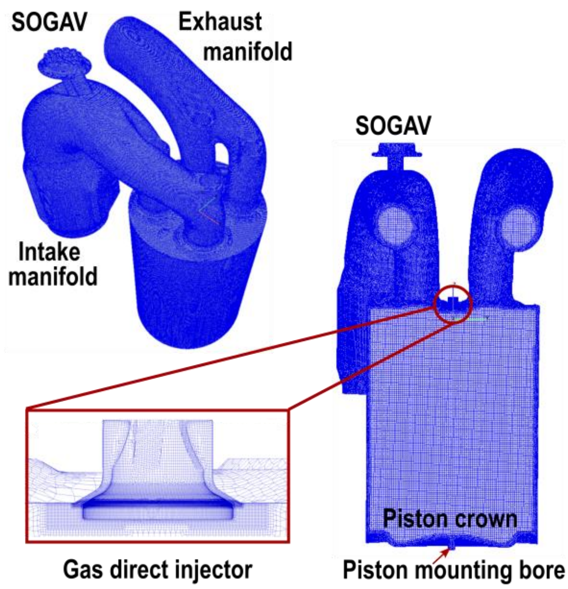

Figure 1: Cylinder meshes with detail of gas direct injector

Due to the asymmetrical nature of the cylinder head and an off-centre diesel spray, the complete geometry of the combustion chamber including the piston top land area and piston mounting bore was modelled. These latter areas were included as they are often sources of methane slip.
A previously conducted sensitivity study (Dinwoodie et. al. 2024a) revealed that a cell size of 4 mm within the domain would give sufficiently accurate results. Local refinements were added to the injector nozzle area, resulting in cell sizes of 0.25 mm in the direct vicinity of the injector and 0.0625 mm in a thin band located around the valve exit. During the diesel injection the mesh is further refined, resulting in mesh sizes ranging from 603 000 to 4 million cells depending on the level of refinement and the simulated piston position.

AVL Fire M software was used to solve the Reynolds averaged Navier Stokes (RANS) equations, Peng-Robinson real gas equation of state (EOS) and the $k - \zeta - f$ turbulence model. An unsteady solver with a variable time step was used, the nominal time step being 0.5° CA. The time step was reduced to 0.1° CA during injections of methane and diesel using a ramp function to maintain numerical stability. The calculations were started shortly after exhaust valve opening (550° before top dead centre (b. TDC), 100 % load) and ended in the power stroke.

Experimental data and previous 0D-1D modelling were used to produce the necessary boundary conditions such as wall temperatures and the pressures at intake and outlet. Walls were treated as non-slip boundaries with constant temperatures.
A supercritical mass flow of methane was calculated based on the engine's motored pressure and applied to the simulated injector. Injector opening and closing is modelled using a ramp function as experimental data was not available at the time of simulation. In line with the injector specifications, the injection duration was set to 30° CA.
Diesel EN590 is injected using a blob injection scheme, whereafter the droplets are tracked in the Lagrangian manner. The Schiller-Naumann drag law (Schiller and Naumann, 1935) and Dukowicz evaporation model (Dukowicz, 1979) are used. Diesel parcels follow a wave-tab secondary breakup scheme. N-Heptane is used as a surrogate fuel for the evaporated diesel. Measurements from previous spray chamber tests were used to parametrise the diesel spray model (Dinwoodie et. al. 2024a).

This CFD model was used to simulate four strokes of the experimental engine's cycle using a direct injector, as is seen in the following section.

4 Results

The results presented in this paper arise from simulations of the 100 % load point. For further details of the validation process, see Dinwoodie et. al. (2024a). The timing of the start of the direct gas injection was varied to investigate the gas jet properties and mixing of the fuel and air. The duration of injection was held constant at 30° CA, in line with the injector's specification. The charge air pressure was set to 4.03 bar, and the fuel mass to 3550 mg, according to full load operation. The gas injection pressure is calculated to be 51 bar.

As the standard engine operates using PFI to maximise the homogenisation of the fuel and air mixture, early injection timings starting from 320° before top dead centre (b. TDC) were first investigated. The air – fuel equivalence ratio, λ , is given as

$$\lambda = \frac{m_a}{m_f \cdot a_{min}} \quad \text{Eq. 4.1}$$

with m_a the actual mass of air present in the cylinder, m_f the mass of fuel and a_{min} the minimum air requirement for stoichiometric combustion. Lambda values smaller than one denote fuel-rich areas, values greater than one denote fuel-lean areas. Figure 2 shows the development of lambda values in 5° crank angle steps over the duration of injection.

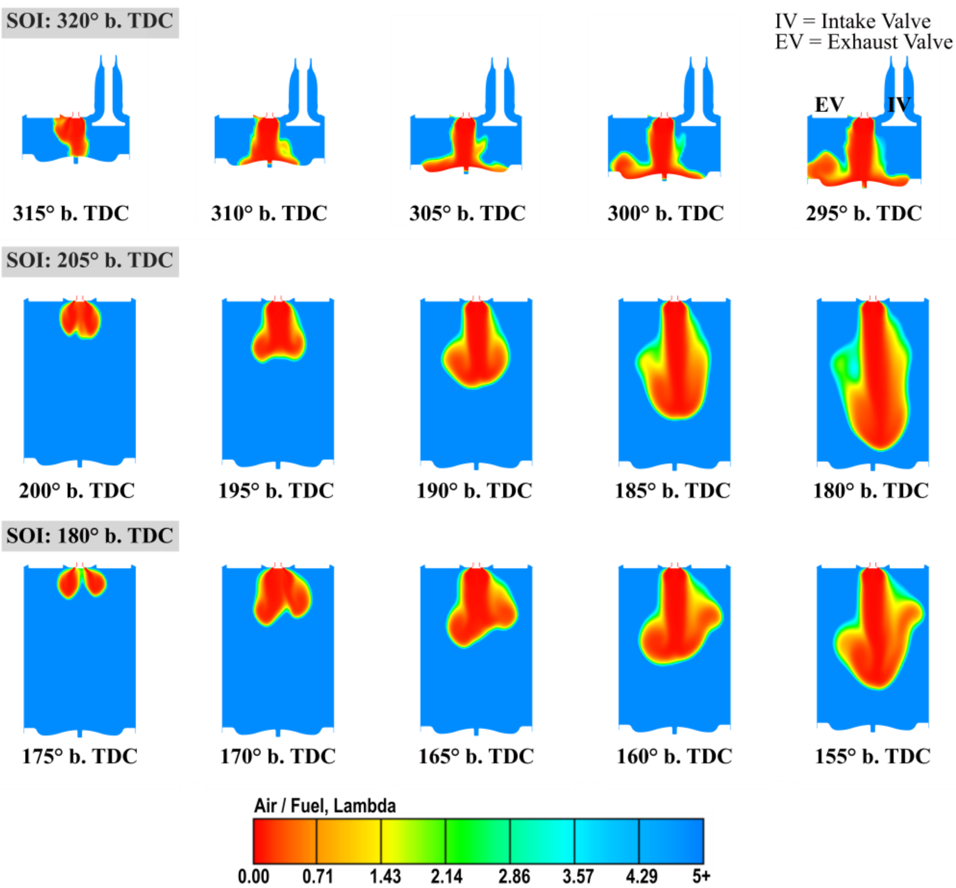

Figure 2: Lambda development for differing injection timings within the intake stroke at 100 % load

With early injections (SOI = 320° / 205° b. TDC), the in-cylinder flow created by the intake airflow influences the shape of the gas jet, creating a narrow and fast-penetrating injection into the centre of the cylinder. This downward flow becomes less after 205° b. TDC with the intake valve closed (Miller cycle). Injecting the fuel later (SOI = 180° b. TDC) results in a wider and slower penetrating gas jet. This trend is also observed when injecting the fuel successively later in the compression stroke, as seen in Fig. 3.

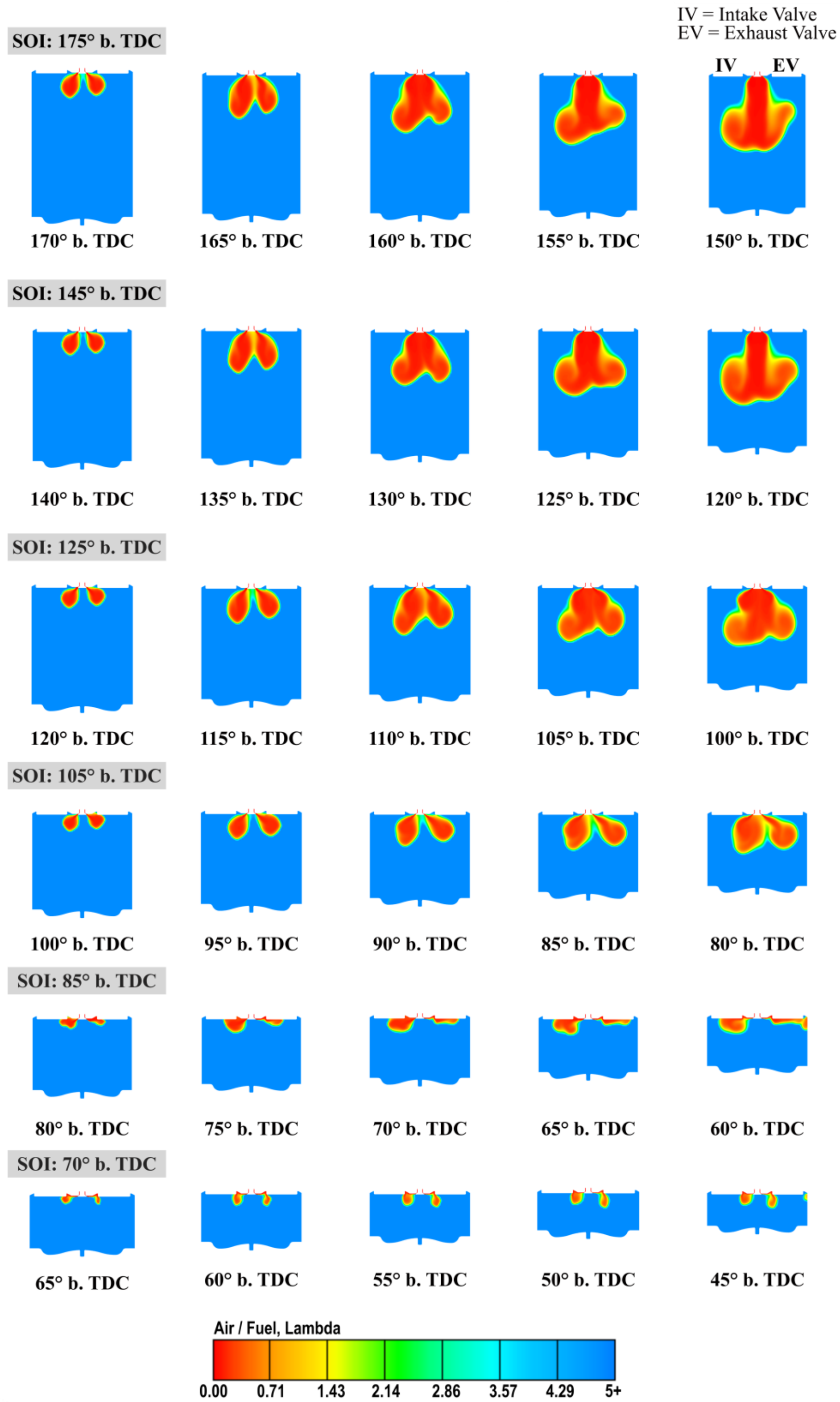

Figure 3: Injection in the compression stroke. Variation of the start of injection at 100 % load

As can be seen, further retarding the injection creates a wider cone angle and slower penetration. Injections after 125° b. TDC, result in a large hollow cone form, different to the jet seen in previous variations. Injections beginning after 85° b. TDC result in the gas jet adhering to and travelling along the cylinder head and being diverted by the closed valves as in the injection at 70° b. TDC. Figure 4 shows the development of penetration length and cone angle over the duration of the injection for the injections in the compression stroke, whereby the ½ cone angle is defined as the angle between the centre line of the cylinder and the outmost extremity of the methane jet.

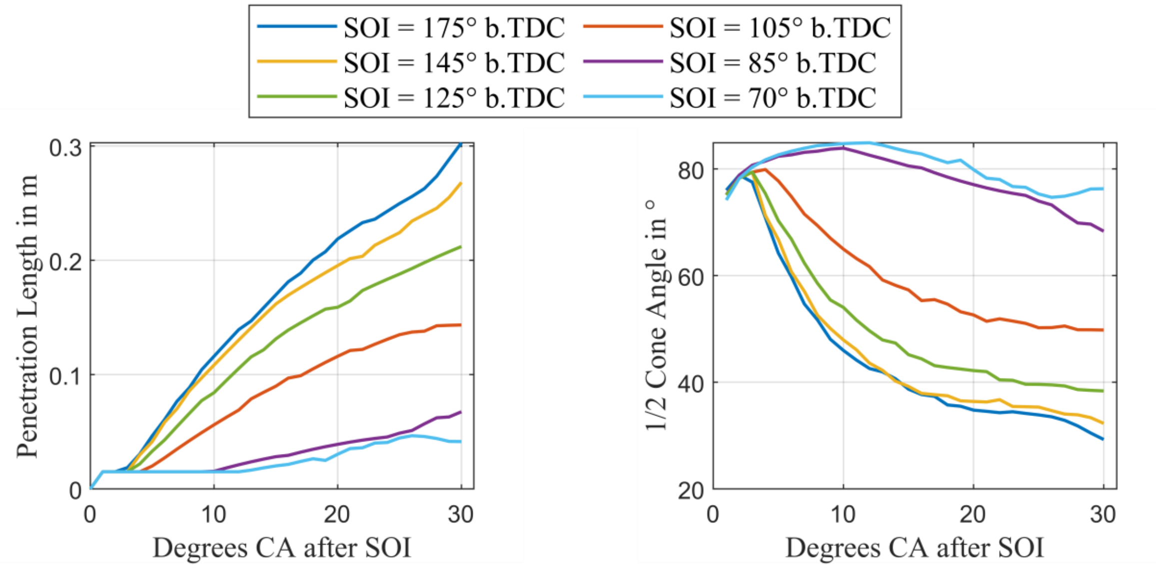

Figure 4: Development of penetration length and cone angle over the duration of injection for injections in the compression stroke

The effect of the higher density within the combustion chamber at later injection timings can clearly be seen in both the penetration length and cone angle. As the start of injection moves closer to TDC, the pressure within the chamber is higher, resulting in a slower penetration and a wider cone angle. This behaviour is further encouraged by the upward motion of the cylinder air. The last two injections, as already observed in Fig. 3, have virtually no penetration, travelling along the cylinder head until the gas is diverted by the walls of the cylinder or the closed intake and exhaust valves. The half cone angle for these points is almost 90°. In order to explore this phenomenon in greater depth, a detailed analysis of the beginning of the latest injection is given in Figs. 5 and 6.

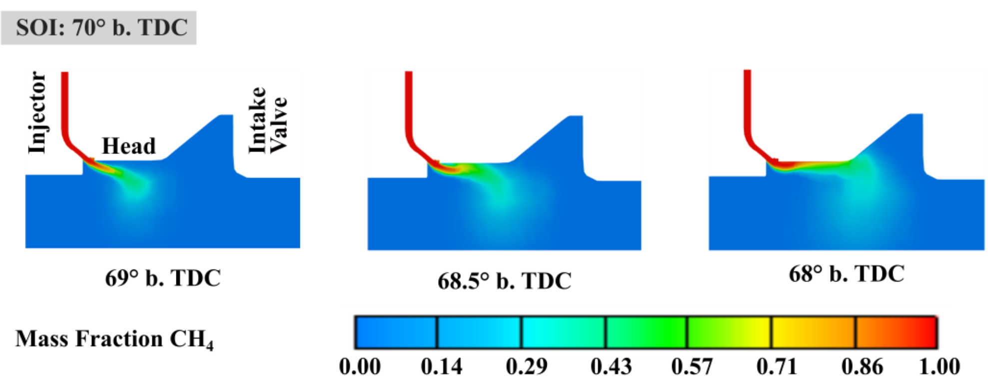

Figure 5. Methane mass fraction at the start of injection, 70° b. TDC

One crank angle degree after the start of injection, the methane jet flows freely from the injector without impingement or adherence to the cylinder head. At 68.5° b. TDC, 0.5° CA later (0.12 ms at 720min^{-1}), the flow has been diverted and the gas adheres to the head. This continues for the remainder of the injection.

Figure 6 shows the effect of temperature and pressure on the jet at 69° b. TDC, shortly before the jet's adherence to the head.

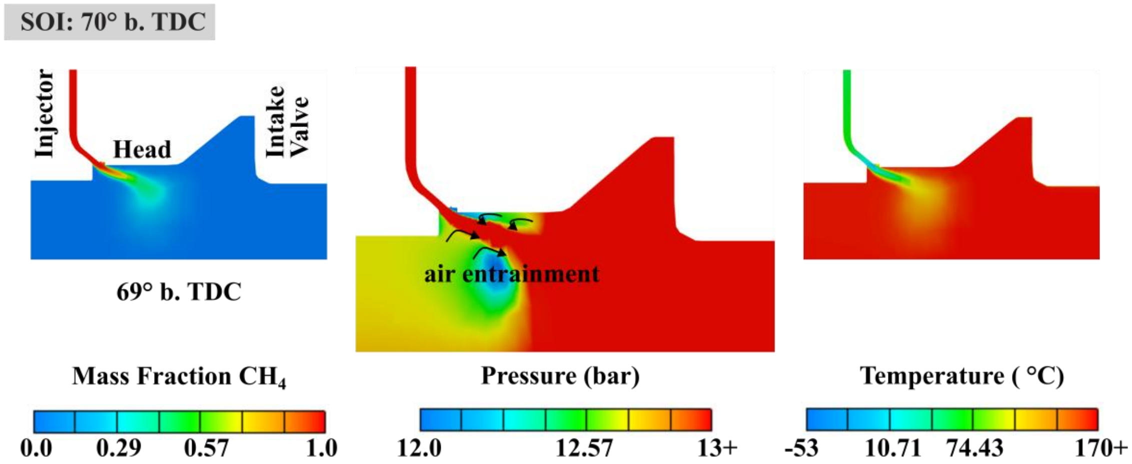

Figure 6: Effect of temperature and pressure on jet development, SOI 70° b. TDC

The jet adheres to the head due to an area of low pressure which is formed between the jet and the cylinder head. Low temperatures of -53°C are calculated within the nozzle opening due to the rapid expansion of gas (right). The temperature is, however, not responsible for the area of low pressure, as the local temperature in this area remains similar to the rest of the chamber. As the gas jet enters the chamber, air is entrained, creating the low pressure areas on either side of the jet and causing the jet to adhere to the cylinder head due to the partial vacuum between the jet and the surface. This is known as the Coandă effect and is well documented, e.g. in Seboldt et.al. (2016), Zanforlin and Boretti (2014) or Sankesh et. al. (2018).

The degree of homogenisation resulting from the various injection strategies was also investigated, as seen in Fig. 7.

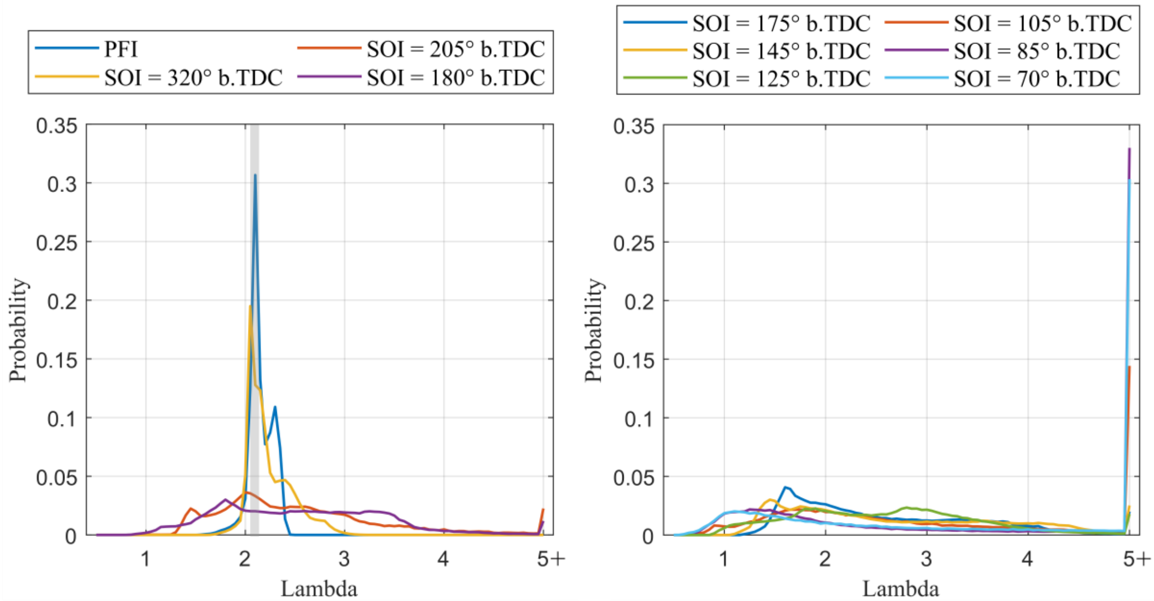

Figure 7: Mass - weighted lambda distribution of the cells within the combustion chamber at 10° b. TDC for differing injection timings at 100 % load. The grey shaded area denotes an ideal homogenisation.

The probability density function of the air-fuel equivalence ratio the cylinder was plotted using the mass-weighted lambda values from the simulation at the end of the compression stroke, 10° b. TDC, shortly before the start of combustion. An ideal homogenisation would result in all values in the chamber occupying a small band of lambda values, as denoted by the grey shaded area in the left-hand plot. As can be seen, the highest homogenisation is achieved using the serial PFI fuel injection. A high degree of homogenisation is also achievable using an intake synchronous direct fuel injection (SOI = 320° b. TDC). Later injections in the intake and compression stroke lead to increasingly heterogenous mixtures and extremely lean areas (lambda > 5). These later injections are likely to create more nitrous oxides (NO_x) when burnt and be more susceptible to engine knock, uncontrolled local autoignitions of fuel after initial ignition due to the rich areas present in these mixtures. The added fact that late injections using the injector without the jet forming nozzle direct methane towards the head and walls of the cylinder as seen in Fig. 3 makes later injection with this injector / piston crown inadvisable, which has also been experimentally validated, see Dinwoodie et. al. (2025). Late injections using the jet forming guide and an appropriate piston bowl geometry can however be used to create a stratified charge combustion regime, as investigated in e.g. Dinwoodie et. al. (2024b) or Dinwoodie et. al. (2025).

5 CONCLUSIONS

3D-CFD simulations of a novel outward opening poppet valve type gas direct injector without its jet-forming nozzle operated on a single cylinder medium-speed dual fuel marine engine were carried out using AVL Fire M software. The purpose of this simulation was to characterise the behaviour of the injector under differing injection strategies.

It was found that when operated in the injection stroke, high gas injection penetration is achieved, which creates a good level of homogenisation towards the end of the compression stroke where ignition occurs, especially at the early, intake synchronous injection timing. Injection in the compression stroke results in successively lower penetration lengths as the injection is moved closer to TDC, due to the higher in-cylinder gas density. The cone angle was also found to increase with increasing chamber pressures.

Between a start of injection of 105° b. TDC and 85° b. TDC a regime change takes place in which the injected fuel adheres to the cylinder head due to the partial vacuum created by the entrainment of air between the jet and the cylinder head (Coandă effect).

For best homogenisation of the fuel and air mix, earlier injection in the intake stroke is the most promising strategy. Injections in the early compression stroke are also possible, but will exhibit less homogenous mixing. Injection after 105° b. TDC should be avoided, as this will lead to large amounts of fuel in near-wall areas which will be affected by wall quenching and further raise methane slip. Later injection timings are possible using the jet-forming nozzle, see e.g. Dinwoodie et. al. (2024a, 2024b, 2025). Fuel injection synchronous with the intake and directly after exhaust valve closing gives the highest degree of homogenisation.

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