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Numerical and experimental analysis of gas DI operation on a medium-speed dual-fuel marine engine

Simulation Technologies, Digital Twins and Complex System Simulation

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ABSTRACT

Direct gaseous injection of syn/bio methane has the potential to massively reduce methane slip from dual-fuel engines. Using direct injection prevents methane from being emitted during valve overlap in turbocharged engines. The gas can also be targeted away from the cylinder walls and piston top land area improving combustion and reducing methane emitted as a result of incomplete combustion due to wall quenching. The result is a highly thermally efficient dual-fuel engine with minimal methane slip. This reduces the GHG balance of the engine and capitalises on the low air pollutant potential of dual fuel engines.

3D – CFD simulation is used to analyse the mixture formation process within the cylinder and recommendations are made for homogenous and stratified charge operation. A numerical comparison of today's standard low-pressure PFI operation system for medium-speed engine applications with a solenoid operated gas admission valve (SOGAV) and in contrast a novel medium pressure direct injection (MPDI) operation reveals the benefits of MPDI. These findings are supported by experimental data from a single-cylinder medium-speed dual-fuel 1/34 DF marine engine, which can be operated using either a SOGAV or a novel MPDI direct gas injector. This injector was specially designed for this purpose and was analyzed in a highly sophisticated injection rate analyser (IRA) and a high pressure – high temperature optical spray chamber, the results of which are presented in a separate paper (Paper 174: "Experimental characterization of newly developed medium & high pressure DI gas injectors for 4-stroke applications to reduce methane emissions", Cepelak et. al.). This analysis is used to validate the 3D - CFD models employed.

Engine operation is simulated at 50 % and 100 % load points at constant speed (720 rpm, generator mode). Two different injector nozzles are investigated on the outward opening poppet valve injector, the first emitting a hollow cone gas injection, the second injecting a narrow jet of methane into the combustion chamber. Detailed analysis of the in-cylinder air-fuel ratio distribution (lambda distribution) is used to highlight the benefits and disadvantages of each nozzle type. Differing piston crown forms are simulated to discern their appropriateness for homogenous and stratified charge operation. A variation of the start of injection reveals optimal operating strategies for each nozzle and piston crown combination. The amount of methane residing in the piston top land and the near wall area is analyzed for each variation, showing how injection timing and nozzle/piston crown combination can be used to target methane away from these critical areas. Detailed combustion simulation reveals the effect of these strategies on thermal efficiency, in-cylinder temperatures and emissions formed. The simulation findings are underpinned by experimental results generated on the 1/34 DF single cylinder research engine, one of the largest of its kind in Europe, installed at the University of Rostock.

Gaseous direct injection in four-stroke engines of this size is very rare. Most manufacturers rely on low pressure port fuel injection, as high pressure direct injection (HPDI) comes at too high a cost point. Using medium-pressure direct injection allows a great reduction in methane slip and improvements to efficiency and power density at much better value than HPDI, offering an environmentally friendly and cost-effective solution to the problem of methane slip in dual-fuel marine engines. Combined with the existing onshore infrastructure and the possibility of the production of syn/bio LNG in the future, mediu- pressure direct injection in dual-fuel engines reveals itself as a viable candidate for the road to a carbon neutral world of shipping.

1 INTRODUCTION

The continued growth of global shipping requires novel engineering solutions to cut emissions. Dual-fuel methane / diesel engines allow a theoretical carbon dioxide reduction of around 25 % in comparison to conventional diesel engines, due to the high H/C ratio of methane.

The easy handling of LNG, the well-established on-shore infrastructure, and the possibility of using any proportion of bio or syn-LNG as a drop in fuel makes its future use attractive. The main obstacle to the widespread use of this technology, however, is the issue of methane slip: unburned methane being released into the atmosphere, as methane is 28 times more harmful to the environment than carbon dioxide on a hundred-year time scale. If methane slip can be reduced or even eliminated, this technology becomes more viable.

Methane slip results from differing mechanisms. The first is due to a portion of methane residing in the intake port after port fuel injection (PFI) and then being lost through the exhaust in the following cycle's valve overlap. This can be eradicated by using a gas direct injector, injecting the fuel directly into the combustion chamber after the exhaust valve is closed.

The other source of methane slip is incomplete combustion, in particular in cold areas of the cylinder such as its walls, where flames are extinguished in a process called flame quenching. Using a direct injector, it is possible to target fuel away from the walls using injection timing and the piston crown geometry to contain the methane.

Still more methane can be lost through methane residing in crevices, an area into which a lot of research and development has been carried out in recent years [1]. Modern dual-fuel engines have very little methane slip due to crevices as a result of this development.

The University of Rostock operates a single cylinder dual-fuel medium speed engine and test bed. This paper presents findings from 3D-CFD investigations into the mixture forming processes in this engine and experimental results showing an experimental direct injector being used to create a homogenous combustion regime similar to that created when using a PFI injector.

The simulations and engine tests take place at 50 % and 100 % load using medium pressure direct injection (10 – 80 bar injection pressure) which is compared with engine- standard PFI operation. Medium pressure direct injection provides the benefits of direct injection with relatively low cost in comparison to high pressure direct injection (100 – 600 bar), making it an interesting technology for potential retrofit packages and new builds for ship operators.

Following the investigations covering homogenous operation, options for a targeted stratified charge approach are presented. A simulated start of injection (SOI) sweep results in an analysis of lambda distribution using 3D-CFD.

The results presented here reveal methods of using a direct gas injector to reduce methane slip. The results are promising and with some optimisation could make dual-fuel methane / diesel engines viable for the marine energy transition.

2 THE EXPERIMENTAL TEST BED

With a bore of 340 mm, the Department of Piston Machines and internal Combustion Engines (LKV) at the University of Rostock operates one of the largest four stroke single cylinder medium speed dual fuel engines in Europe.

The following section highlights the test bed's capabilities.

2.1 Engine Parameters

The experimental engine's parameters are shown in table 1.

Table 1. Engine Characteristics

Parameter	
Bore	340 mm
Stroke	460 mm
Compression ratio	12.75*
Nominal operating speed	720 min^{-1}
Power output	> 500 kW

* Differing compression ratios can be attained through the use of compensation plates and varying piston crowns.

2.2 Fuel System

The dual-fuel engine is supplied by

- A diesel common rail system delivering up to 2200 bar injection pressure
- A low-pressure solenoid operated gas admission valve (SOGAV) injecting gas at up to 10 bar when the engine is running in PFI mode
- A medium / high pressure gas direct injector operated at up to 80 or 600 bar accordingly.

The results presented in this paper compare engine operation in PFI mode with medium pressure DI operation. The gas direct injector is mounted centrally in the engine head, the diesel pilot injector is mounted slightly off-axis and tilted by 6.5° as seen in figure 1. The nozzle is adjusted accordingly.

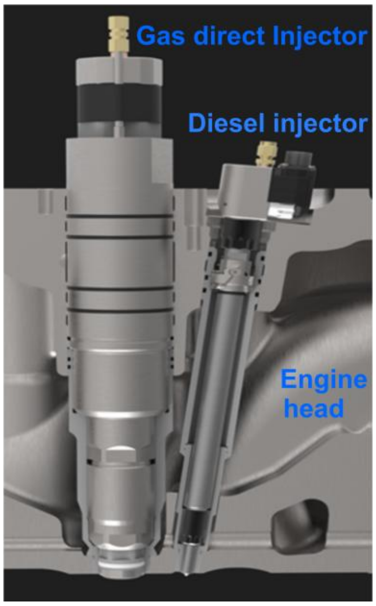

Figure 1. Fuel injectors in engine head

The gas direct injector is of an outward opening poppet valve type with a diameter of 25 mm, producing a hollow cone. A jet forming nozzle can additionally be installed to target the fuel into the centre of the combustion chamber.

2.3 Test Bed Capabilities

The single cylinder engine test bed infrastructure offers the following possibilities:

- Carbon dioxide and propane admixtures to the natural gas, changing the methane number and lower heating value to simulate worldwide gas qualities
- Hydrogen admixtures also possible [2]
- Charge air up to 8.5 bar
- Turbocharger simulation by altering charge air and engine back pressure
- Fully programmable ECU with next cycle control
- FTIR exhaust gas measurement
- EGR System, up to 30 % EGR rate at full load
- Experimental canning for the analysis of catalysts with exhaust gas analysers before and after the catalyst [3].

Figure 2 shows the test bed infrastructure.

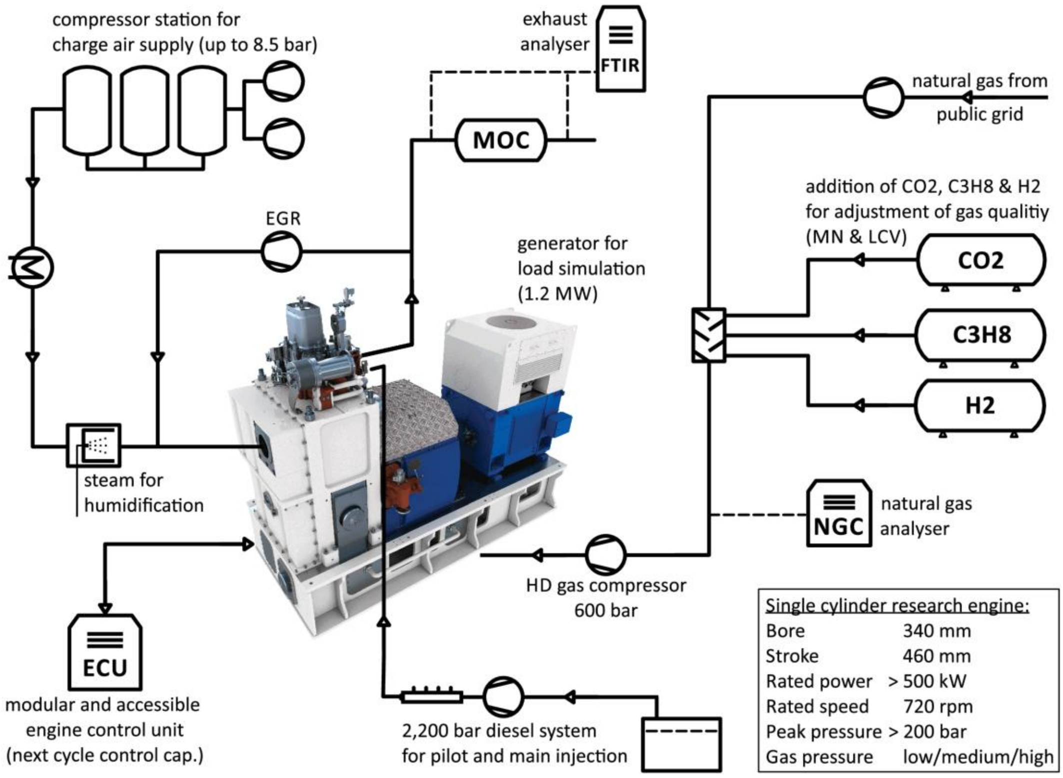

Figure 2. Test bed infrastructure

3 THE CFD MODEL

A 3D-CFD model was developed to represent the experimental engine. A full cylinder mesh including intake and exhaust ports was created using AVL FAME software, seen in figure 3.

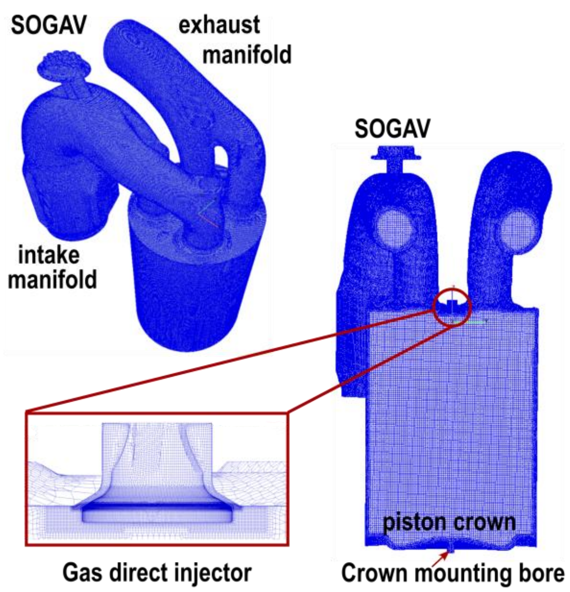

Figure 3. Cylinder Meshes with detail of gas direct injector

The piston top land area and the bore-hole used for mounting the piston crown were also modelled, as these are areas where methane can reside without taking part in combustion.

The full cylinder was modelled due to the asymmetrical nature of the cylinder head and off-centred diesel spray.

A previously conducted sensitivity study (see [4]) revealed that a cell size of 4 mm within the domain would give 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.

Two boundary layers of 0.2 mm thickness were used to resolve near wall flow. During injection, three boundary layers with a thickness of 0.05 mm were employed.

Using these parameters the mesh size ranges 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° crank angle (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 (575° before top dead centre (b. TDC), 50 % load / 550° 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.

The mass flow of methane through the gas direct injector was calculated on the basis of a supercritical injection at a pressure high enough to remain supercritical over the duration of injection, dictated by the engine's motored pressure. Injector opening and closing is modelled using a ramp function, as experimental data showing the exact opening and closing behaviour of the injector was not available at the time of simulation. The injection duration was set to 30° CA, in line with the injector's specifications.

Diesel EN590 is injected using a blob injection scheme, whereafter the droplets are tracked in the Lagrangian manner. The Schiller-Naumann drag law [5] and Dukowicz evaporation model [6] 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 [4].

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

Starting with a comparison of the simulated and experimental pressure traces, this section first presents the use of a direct injector to produce a homogenous combustion regime similar to that created when using a PFI injector. Experimental results for this kind of operation are then shown. The section closes with 3D – CFD simulations of stratified charge operation, demonstrating its ability to keep methane away from cold cylinder walls.

4.1 Comparison of measured and simulated cylinder pressure

The measured cylinder pressure was compared with the simulated cylinder pressure to assess the accuracy of the CFD model. Figure 4 shows the comparison for the exhaust and intake stroke.

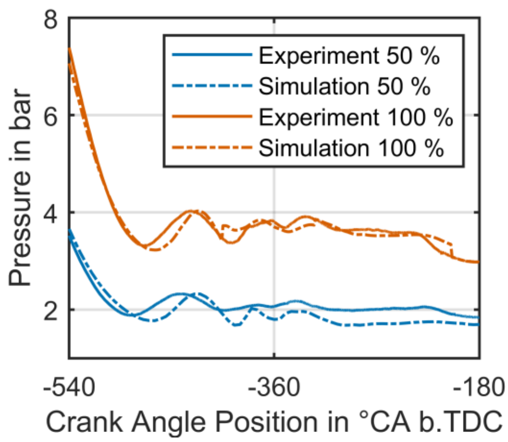

Figure 4. Comparison of measured and simulated pressure in exhaust and compression stroke, 100 % load and 50 % load

The simulated pressure matches the measured cylinder pressure well for both 50 % and 100 % load cases. The peaks and troughs observed in the measured result are represented in the simulated results, although these appear a little delayed in the simulation. This could be the result of pressure sensor position, as the simulated result represents a mean value of all the cells in the domain, whereas the measured result arises from the detected pressure at a specific location within the cylinder. The 50 % load point demonstrates the most discrepancy in pressure, the maximum difference being 0.31 bar. At intake valve closing, however this has been reduced to 0.15 bar difference. The compression stroke can be seen in figure 5.

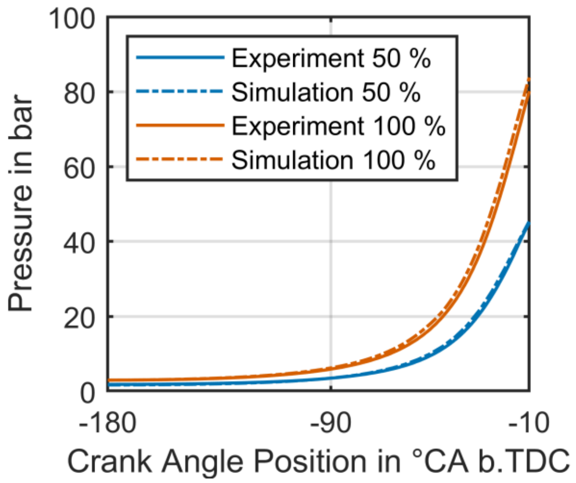

Figure 5. Comparison of measured and simulated pressure in compression stroke, 100 % load and 50 % load

A good agreement between the measured and simulated cylinder pressure is also seen in the compression stroke. In both cases, the simulated pressure is slightly higher than the measured result, demonstrating a pressure difference of 0.26 bar at 10° b. TDC for 50 % load, and 3.72 bar at 100 % load. Engine leakages such as blowby are not taken into account in the simulated result, which gives one reason for the observed offset. Another factor to consider is a possible discrepancy between the actual and specified compression ratio and/or valve lifts of the engine. The error could also be due to heat transfer calculation errors caused by slightly incorrect wall temperatures.

In general, however, the pressure traces are very comparable and give a good basis for the further calculation of injection, mixture formation and combustion.

4.2 Homogenous Operation

Simulations and experiments were used to compare the standard homogenous PFI operation with the novel DI method.

The results are presented in the following. A detailed analysis of the simulated results can also be found in [4] and [7].

4.2.1 Simulation of homogenous operation

The standard PFI gas injection timing was set at 359° b. TDC with a duration of 65° CA. As the start of combustion takes place around 5° b. TDC, this allows a minimum of 291° CA for homogenisation (end of injection (EOI) to start of combustion (SOC).

The earliest possible direct gas injection to avoid methane slip through the exhaust is after exhaust valve closes (EVC). For this reason, the start of gas injection (SOI) was set to 320° b. TDC for full load operation and 336° b. TDC for 50 % load, as the experimental engine utilises cam phasing to apply differing Miller timings for the different load points. The duration of injection was set to 30° CA, in line with the injector specifications. These timings respectively allow 285° CA and 301° CA for homogenisation (EOI to SOC).

Due to the slight possibility of a portion of the directly injected fuel being lost through the intake valve, a further SOI was simulated at 205° CA (100 %) and 195° CA (50 %), after intake valve closing.

Finally, the SOI was set to the start of compression, 180° CA b. TDC for both cases.

These simulations were performed without the jet forming nozzle in order to widely distribute the gas in combustion chamber and achieve better homogenisation. The parameters for the following simulations are shown in table 2.

Table 2. Simulation parameters

Parameter	50 % load	100 % load
Charge air pressure	2.05 bar	4.03 bar
Methane mass	1900 mg	3550 mg

The mass – weighted lambda distribution for the different cases was then compared to reveal the possibilities of homogenous operation using a direct injector.

Figure 6 shows the lambda distributions at 10° b. TDC, shortly before the start of combustion, for the four 100 % load cases.

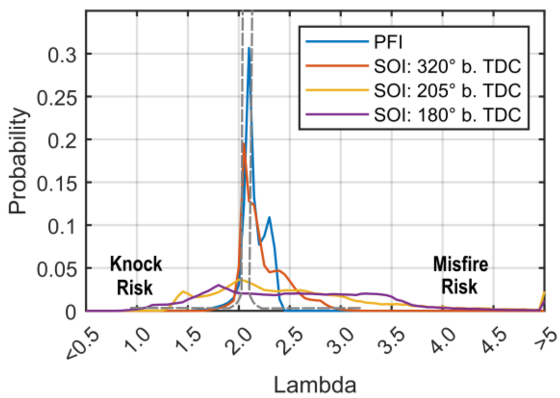

Figure 6. Lambda distribution at 10° b. TDC for various injection timings at 100 % load, PFI and DI operation

A near-perfect homogenisation is represented by the grey dashed line, where all the values in the sample are closely grouped around a single lambda value, similar to a Dirac function. At 100 % load, PFI offers the best degree of homogenisation, with 31 % of the domain receiving a lambda value between 2.05 and 2.15. Homogenisation is also evident in the earliest direct injection (320° b. TDC). The mixture is more heterogeneous, however, with a tendency towards the lean side. This is to be expected due to the excess air increase due to the reduction of displacement effects which take place during PFI injection. The lesser degree of homogenisation can be explained by the reduced mixing time and possibly by the missing turbulence acting on the mixture as it passes through the intake valve. Later direct injection timings result in very heterogeneous lambda distributions with a higher risk of knock due to the prevalence of rich areas. NO_x emissions could also be expected to rise as a result of these rich areas.

The amount of methane present in the intake at intake valve closing was also calculated and is presented in table 3.

Table 3. Methane present in the intake manifold at intake valve closing, 100 % load

Injection Regime	Methane
PFI	376.1 mg (10.6 % of total fuel injection)*
DI, 320° b. TDC	< 0.01 mg
DI, 205° b. TDC	0 mg
DI, 180° b. TDC	0 mg

* Results only representative of experimental engine

The methane present in the intake manifold at intake valve closing represents the potential for methane slip during valve overlap in the following cycle. Not all of this fuel will be lost through the exhaust, as some will remain in the cylinder and be burned in the next cycle. However, minimising the fuel present in the intake manifold will cut methane slip during valve overlap and reduce cyclical variations due to methane from the previous cycle taking part in the next cycle combustion. Direct injection shows great promise even when the fuel is injected synchronously with the intake, as a negligible amount of methane enters the intake manifold.

Direct injection after exhaust valve closing (320° b. TDC) reveals itself as the most promising strategy for achieving a good level of homogenisation while reducing methane slip potential, as is also observed at the 50 % load point. Figure 7 presents the mass-weighted lambda distributions of the 50 % load cases.

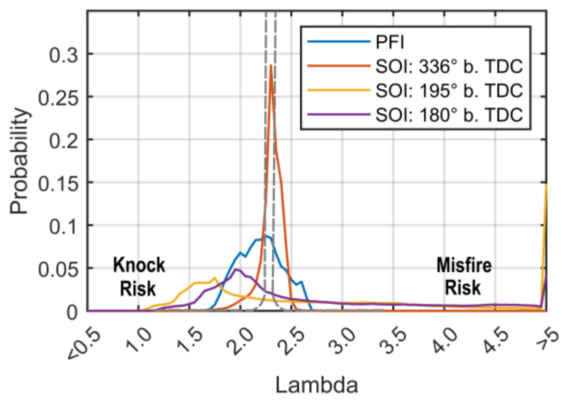

Figure 7. Lambda distribution for various injection timings at 50 % load, PFI and DI operation

At 50 % load, the intake synchronous direct injection (336° b. TDC) results in a better homogenisation than PFI. This is due to the alteration of the valve timings for 50 % load allowing 16° CA more time for DI homogenisation, as the exhaust closes earlier. The degree of homogenisation presented by the PFI scheme at 50 % load is less than at 100 % load due to the comparative lack of turbulence for mixing, as seen in figure 8.

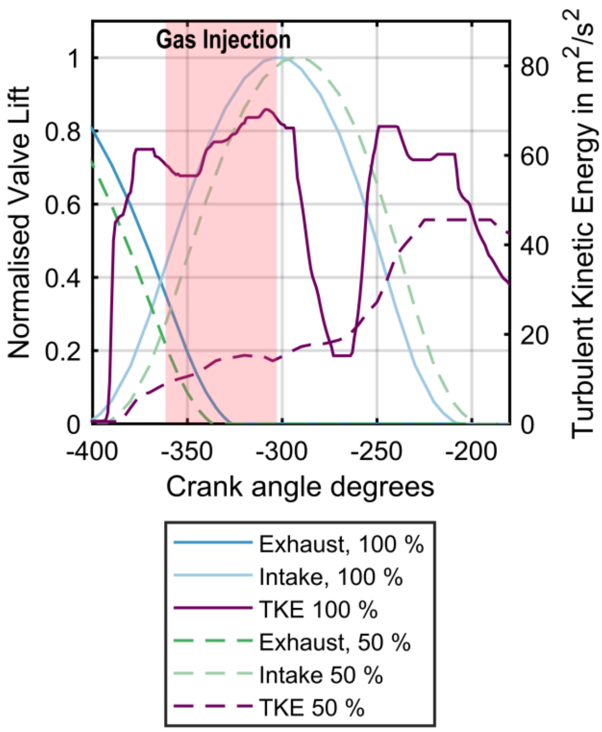

Figure 8. Turbulence Kinetic Energy (TKE) in the cylinder, Valve lift and Gas injection for PFI operation

The simulated turbulence kinetic energy, which represents the small-scale turbulence, is nearly three times higher during gas injection at 100 % load than at 50 %, accounting for the lack of mixing during the intake stroke and the decreased level of homogenisation with PFI observed at 10° b. TDC. The lack of turbulence at 50 % load is likely caused by the reduced valve overlap and lesser charge air pressure.

As seen in figure 7, using DI after exhaust valve closing results in a high level of homogenisation. The later injection timings result in a lesser degree of homogenisation and richer areas, increasing the risk of knock and high NO_x production. Very lean areas in these later timings could also increase the risk of incomplete combustion, leading to methane slip.

As at 100 % load, the methane present in the intake at intake valve closing was calculated and is presented in table 4.

Table 4. Methane present in the intake manifold at intake valve closing, 50 % load

Injection Regime	Methane
PFI	183.5 mg (10.3 % of total fuel injection)*
DI, 320° b. TDC	< 0.01 mg
DI, 205° b. TDC	0 mg
DI, 180° b. TDC	0 mg

* Results only representative of experimental engine

The following conclusions can be drawn from the simulation shown:

- Early, intake synchronous gas injection timings should be chosen for a high degree of homogenisation when using a gas direct injector
- Using a direct injector reduces the methane slip potential during valve overlap to negligible levels
- Later direct injection timings could increase knock risk and NO_x production.

The results of engine testing are presented in the following section.

4.2.2 Engine testing: homogenous DI operation

An SOI sweep was carried out at 50 % and 100 % load. The engine was operated using the parameters shown in table 5.

Table 5. Engine operating parameters

Parameter	50 % load	100 % load
IMEP	10.8 bar	20.8 bar
Engine speed	720 min^{-1}	720 min^{-1}
Charge air pressure	2.05 bar	4.03 bar
Charge air Temperature	45° C	45° C
α_{50}	7.5-8 ° a. TDC	8° a. TDC
Gas injector actuation time	2.0 - 7.8 ms	7.9 – 8.5 ms
Gas pressure	22 bar	30 bar
Gas mass (mean)	2089 mg	3568 mg
Diesel rail pressure	1200 bar	1200 bar

In the following, engine knock, emissions of NO_x and CH_4 and the coefficient of variance for differing SOI's are compared. In each of the figures, the results have been normalised using the highest value of the sample.

Figure 9 shows engine knock levels at 50 % and 100 % load.

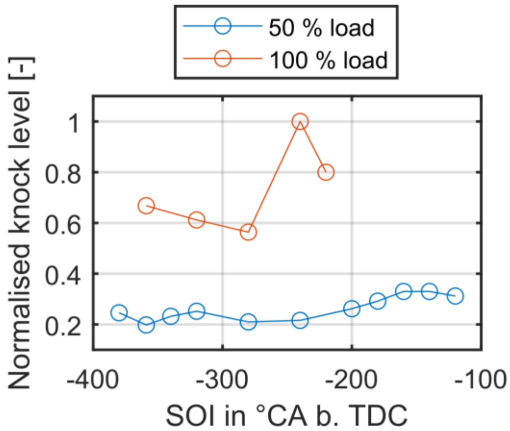

Figure 9. Normalised knock level over SOI sweep for 50 % and 100 % load

The knock level is higher at 100 % than at 50 % load and tends to increase with a later injection of methane. At 100 % load, the knock level was too high to safely complete the SOI sweep, accounting for the missing later injection timings.

The increase in knock is most likely due to the richer areas present in the less homogenous mixture, as demonstrated in 4.2.1. These richer areas should also produce more NO_x, as seen in figure 10.

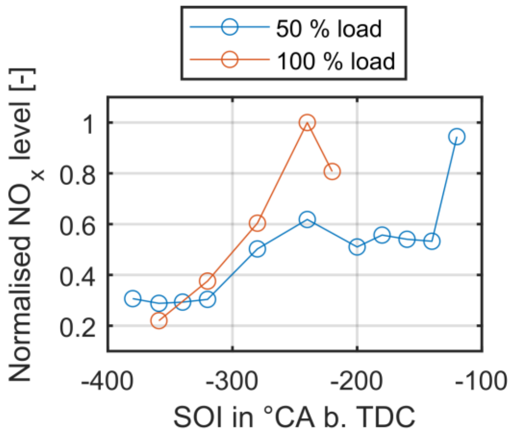

Figure 10. Normalised NO_x levels over SOI sweep for 50 % and 100 % load

The upward trend for NO_x production with later injection is clearly visible. More NO_x is produced at 100 % than 50 % as there is more total fuel distributed within the rich areas in the cylinder. Later injections produce four to five times the amount of NO_x than earlier injections.

Figure 11 shows the emissions of NO_x and CH_4 at 100 % load.

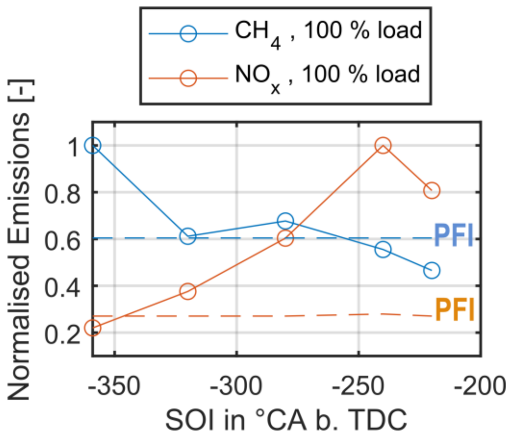

Figure 11. CH_4 and NO_x emissions over SOI sweep at 100 % load, compared to PFI reference case

A typical NO_x - CH_4 trade-off can be seen here. The richer areas in the less homogenous mixtures at later injection timings create higher cylinder temperatures, causing an increase of NO_x and better combustion of the methane. This trade – off is well documented [8] [9] [10].

At 320° b. TDC, methane slip due to incomplete combustion is around that of the complete methane slip (short circuiting and incomplete combustion) in PFI mode. This can be reduced by retarding the injection but at the cost of high NO_x levels (around 4 times as much NO_x at 220° b. TDC). Methane slip could potentially be further reduced by improving the combustion regime via the diesel pilot injection (multiple injections, higher rail pressure).

In addition to emissions and knock, another factor to consider is the stability of operation. In order to assess this, the coefficient of variance (COV) of the indicated mean effective pressure (IMEP) was recorded for 50 % and 100 % load points, as seen in figure 12.

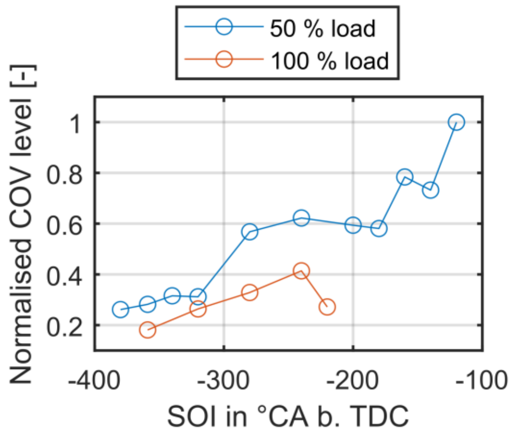

Figure 12. COV of IMEP over SOI sweep at 50 % and 100 % load

This plot reveals that engine running stability decreases with a later injection timing. Variability is higher at 50 % load than 100 % load. At earlier injection timings, engine operation is comparably stable, with the 100 % load point being only slightly more stable than 50% load operation.

Based on these findings, the following recommendations can be made for homogenous engine operation:

- Early injection timings are favourable for reducing knock and NO_x levels and improving engine running stability
- Methane slip caused by incomplete combustion can be reduced by retarding the injection. At 100 % load, 320° b. TDC offers the best methane / NO_x trade-off, which is in line with the simulation presented in 4.2.1.

It has been shown that DI operation offers a viable alternative to PFI. Further benefits, such as the targeting of fuel away from combustion chamber walls, further reducing methane slip, could be realised using a stratified charge approach, as is explored in the next section.

4.3 Stratified Charge Operation

The effect of PFI and single and double direct injections on near-wall methane was investigated for the 50 percent [7] and 100 percent [4] load points, the results of which can be seen in figure 13.

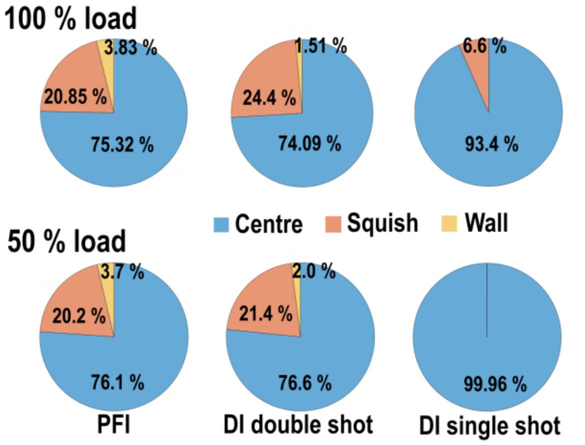

Figure 13: Influence of injection strategy on near-wall methane, distribution of methane at 10° b. TDC for different injection strategies

For this analysis, the chamber was divided into three areas; the centre, squish and near-wall areas, and the amount of methane in these areas tracked. Figure 13 shows the methane distribution at 10° b. TDC, around the time when the start of combustion could be expected. Using either a double shot or single shot direct injection regime results in a reduction of near-wall methane. The single late injection strategy reduces near-wall methane to zero at both load points, minimizing the risk of methane slip by wall quenching. As engine knock is a function of time, as time is required for the cracking of the fuel, it is to be expected that knock levels could be reduced at late injection timings. For these reasons, further investigations were carried out using this injection strategy.

A specialised piston crown was developed for this combustion regime, utilising a higher compression ratio (14.1) and a geometry designed to retain the methane within the bowl while enabling air entrainment. The piston crown has been manufactured. However, at the time of writing, it has not been used experimentally. The piston crown design can be seen in figure 14.

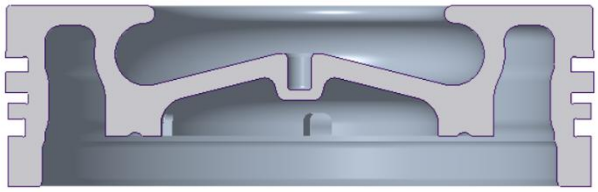

Figure 14. Experimental piston crown for stratified charge

A single, late, supercritical injection of methane was performed at 55° b. TDC, 70° b. TDC and 85° b. TDC. The injection duration and amount of fuel injected was held constant at 30° CA and 3550 mg (100 % load) / 1900 mg (50 % load). The charge air pressure was set to 2.05 bar (50 % load) or 4.03 bar (100 % load). The injector was simulated with the jet forming nozzle to target the fuel into the piston bowl.

Figure 15 shows cutplanes and the methane mass fraction (ω_{CH4}) within these planes for the 100 % load case and a start of injection at 70° b. TDC.

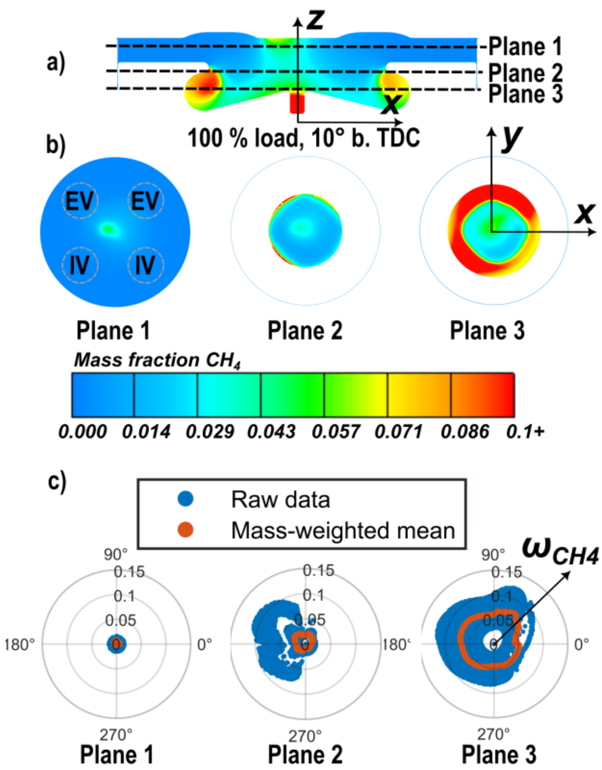

Figure 15. Methane distribution at 10° b. TDC, 100 % load, SOI 70° b. TDC

A cut through the y-axis (a) reveals the methane distribution at 10° b. TDC, around the time of the start of combustion. The methane and diesel are well centred in the combustion chamber, the highest concentrations of methane are found in the pockets of the piston crown and in the central mounting bore. Some methane appears to adhere to the centre of the head, demonstrating the Coandă effect. Three horizontal planes were then chosen for analysis, the first in the centre of the clearance volume (plane 1), the second at the mouth of the piston crown (plane 2) and the third in the centre of the piston bowl (plane 3), as seen in b). The position of the intake valves (IV) and exhaust valves (EV) are also shown here. The ring around the central portion of the plane 2 and plane 3 cuts is the piston top land area.

A mass-weighted mean mass fraction of methane was then calculated for each 5° section of the combustion chamber, as seen in c). The orientation of the polar lambda plots matches the orientation of the planes shown in b). As can be seen, each consecutive plane contains a higher amount of methane, confirming the ability of the piston crown to retain the fuel charge and create a stratified charge along the vertical axis.

The portion of the combustion chamber under the intake valves (180° - 359°, lower half of the circle) is somewhat leaner than the portion under the exhaust valves in all three planes. The polar plot offers a good basis for comparison of the methane distribution at different loads and injection timings, as seen in figure 16.

The positions of the intake and exhaust valves are marked in the top left polar plot. Dashed and dotted lines mark the mass fraction of methane corresponding to lambda values of one and two.

The area under the intake valves is leaner at 100 % load. In comparison, at 50 % load, the area between intake and exhaust valve appears leaner for all cases. Comparing the 100 % load points, injection at 70° b. TDC creates a richer mix in the piston bowl (plane 3) due to the reduced time for mixing. It would be expected that an injection at 55° b. TDC should result in an even richer mix in plane two and three, however, these are leaner. Plane one, however, contains a richer area around 0° which is also observed in plane 2. As the gas injector closes, the gas jet becomes weaker, becoming more susceptible to being wafted by the in-cylinder gas movement. In this case, a portion of the weakening fuel jet is blown to the right-hand side of the cylinder, causing the locally rich area seen in plane one. This fuel is then missing in planes two and three, causing them to appear leaner. At 50 % load, more of the fuel is present in the upper planes (planes one and two), due to the lower cylinder pressure, as seen in the comparison of 100 % load and 50 % load injections shown in figure 17.

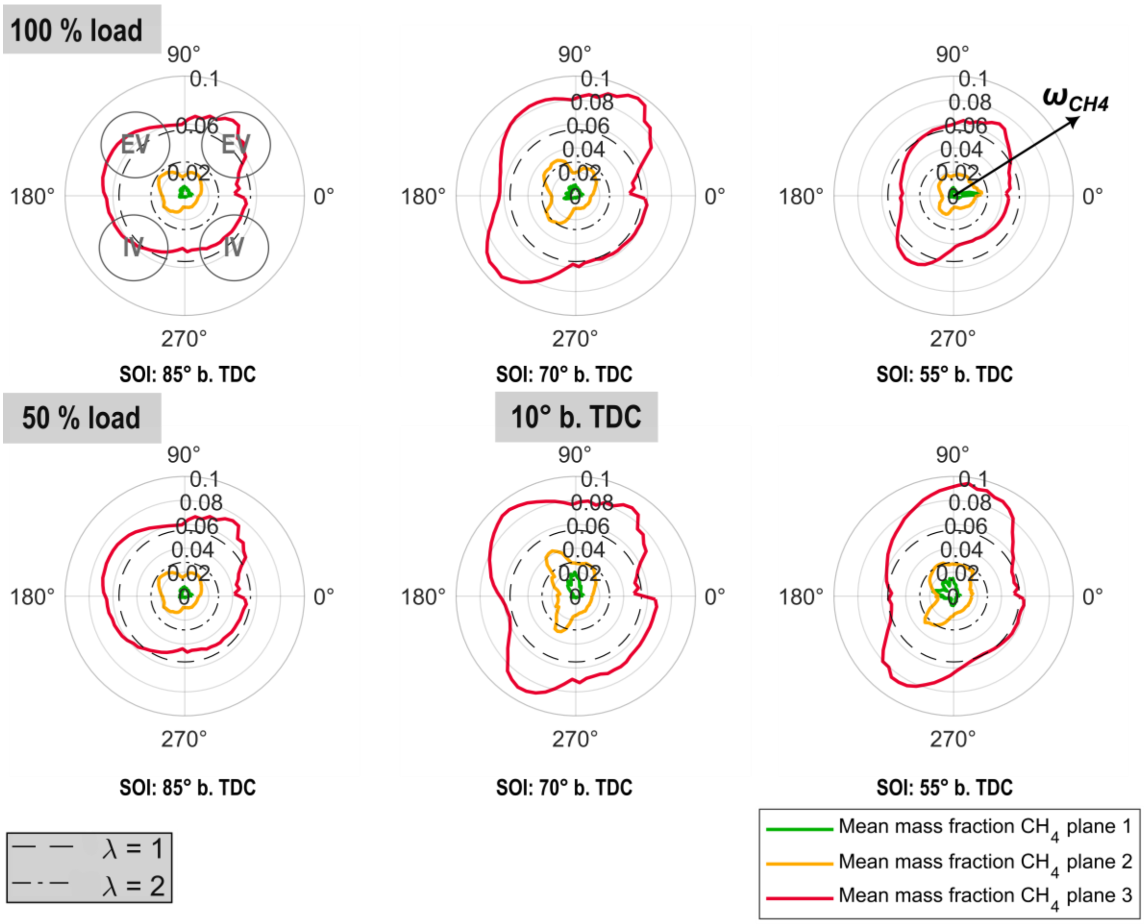

Figure 16. Comparison of methane mass fraction at 10° b. TDC, 100% and 50 % load over SOI sweep

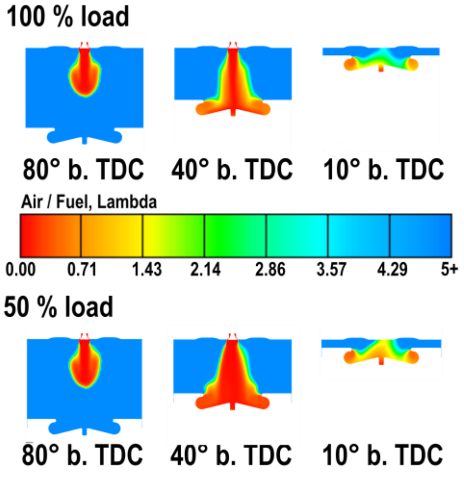

Figure 17: Comparison of 100 % and 50 % gas injection and mixing, SOI at 85° b. TDC

The lower cylinder pressure at 50 % load creates a wider fuel jet and allows the methane to splash out of the piston bowl into the central area of the combustion chamber, making the top two planes richer than at 100 %. The upward flow is slightly stronger on the exhaust side, carrying more of the fuel into the upper planes, seen in the richer areas around 90° in all three 50 % load cases (figure 16). At 100 % load, the fuel is nestled in the piston bowl with little spillage to the central area of the cylinder.

To assess the effects on combustion produced by the differing injections at different loads, experimental testing or simulation with a parametrised combustion model is necessary. At the time of writing, this model had not been fully parametrized and engine testing had not yet been carried out. In general, however, a later injection should decrease the chance of knock, due to the reduced time for fuel-cracking.

5 CONCLUSIONS

This paper aimed to demonstrate how a gas direct injector could be used to decrease methane slip on a gas/diesel dual-fuel medium speed engine.

3D-CFD simulation showed that homogenous operation at 50 % and 100 % load was possible, the best homogenisation being attained from gas injection while the intake valve is open but after exhaust valve closes to eradicate methane slip during valve overlap. Using this strategy, the mixture at 50 % load was more homogenous than when using the engine standard PFI injection approach. At 100 % engine load, the level of homogenisation was less than when using PFI, nevertheless homogenous enough for a low-NO_x operation as was shown in experimental testing.

The experimental testing revealed an optimal injection timing for 100 % load operation. With the start of activation set at 320° b. TDC, similar levels of NO_x and CH_4 emissions were achieved as when using PFI. Potential remains for further reducing methane emissions due to incomplete combustion. Multiple diesel injections and/or using a higher diesel rail pressure could result in a better combustion of fuel. Stable operation was attained at both 50 % and 100 % load.

For homogenous operation, simulation and experiments point to the following operating strategy: using an early direct injection after exhaust valve closing to utilise the turbulence from the intake air for good mixing. A better degree of homogenisation results in less NO_x and less engine knock, as richer areas within the cylinder are avoided. The engine was also found to run with increased stability using early injection timings.

Stratified charge operation using a single late injection has been shown to reduce near-wall methane to zero, which should reduce methane slip by wall quenching. 3D –CFD simulations of an SOI sweep using such a strategy at 50 % and 100 % load revealed that at 100 % load, the fuel remains nestled in the piston bowl, whereas at 50 % load, the fuel spills out of the mouth of the piston bowl into the central portion of the chamber and up to the underside of the cylinder head. It was observed that the lambda distribution within the cylinder was not rotationally symmetrical, but leaner on the intake valve side at 100 % load and leaner between exhaust and intake valves at 50 % load.

A parametrised combustion model and / or experimental analysis of this injection strategy is necessary to determine its effects on combustion and emission formation. Parametrising a standard combustion model to deal with the very long ignition delays of the small pilot fuel injection quantities which are typical for diesel / gas dual-fuel engines and are governed by different effects than standard diesel combustion, combined with the slow combustion following ignition in the low turbulence combustion chamber has proved challenging.

Direct injection shows potential for reducing methane slip. By injecting the methane after exhaust valve closing, methane emissions via short-circuiting during valve overlap are avoided completely. However, methane slip resulting from incomplete combustion remains comparable with PFI operation, meaning that further reducing methane emissions by reducing wall quenching and achieving a more complete combustion of methane is key to the future success of this technology.

6 DEFINITIONS, ACRONYMS, ABBREVIATIONS

3D-CFD: Three dimensional computational fluid dynamics
$\boldsymbol{\alpha_{50}}$: Centre of combustion
ω_{CH4}: Mass fraction of methane
°CA: Crank angle degrees
a. TDC: after top dead centre
b.TDC: before top dead centre
COV: Coefficient of variance
DI: Direct injection
ECU: Electronic control unit
EGR: Exhaust gas recirculation
EOI: End of injection
EOS: Equation of state
EV: Exhaust valve
EVO: Exhaust valve opening
FTIR: Fourier transform infra-red
IMEP: Indicated mean effective pressure
IV: Intake Valve
MOC: Methane oxidisation catalyst
NO_x: Nitrous oxides
PFI: Port fuel injection
RANS: Reynolds averaged Navier Stokes
SOC: Start of combustion
SOI: Start of injection
TDC: Top dead centre
TKE: Turbulence kinetic energy

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8 REFERENCES AND BIBLIOGRAPHY

[1] MAN Energy Solutions. 2025, 13th January. "Managing methane slip" https://www.man-es.com/campaigns/download-Q1-2025/Download/managing-methane-slip/d34a34a1-cc03-4d99-a4e1-30385cf12518/Managing-Methan-Slip

[2] Glauner, M. et. al. 2025. Paper No. 105, Lowest GHG emissions on medium speed engines by higher epsilon and H2-admission to the fuel gas. *CIMAC Congress 2025*, Zürich

[3] Seipel, P. et. al. 2025. Experimental Long-Term Study of a Methane Oxidation Catalyst on a Medium-Speed Dual-Fuel Engine. *CIMAC Congress 2025*, Zürich

[4] Dinwoodie, J. et. al. 2024. 3D-CFD Simulation of Direct Gaseous Injection for a Medium Speed Dual-Fuel Engine, *8th RGMT*, Rostock, 1: 334-359.

[5] Schiller, L. and Naumann, A. 1935. "A Drag Coefficient Correlation." *Zeitschrift des Vereins Deutscher Ingenieure*, 77: 318-320

[6] Dukowicz, J.K. 1979. "Quasi-steady droplet change in the presence of convection", *informal report, Los Alamos Scientific Laboratory,* LA799MS,

[7] Dinwoodie, J. et. al. 2024. Numerical Analysis of a Medium Pressure Gaseous Direct Injector on a Marine Medium Speed Dual-Fuel Engine, *MTZ Heavy-duty- On- und Off-Highway-Motoren 2024*, Eisenach

[8] Edward W. Carr et. al, 2023. *Options for Reducing Methane Emissions from New and Existing LNG-Fueled Ships*; report for the International Council on Clean Transportation (ICCT)

[9] Schleef, K. 2023. *Analyse und Optimierung des Verbrennungsprozesses von Dual-Fuel-motoren bei Betrieb mit schwankenden Brenngasqualitäten*. Dissertation, Universität Rostock

[10] Yan, F. et. al. 2017. Application of hydrogen enriched natural gas in spark ignition IC engines: From fundamental fuel properties to engine performances and emissions. *Renewable and Sustainable Energy Reviews*

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