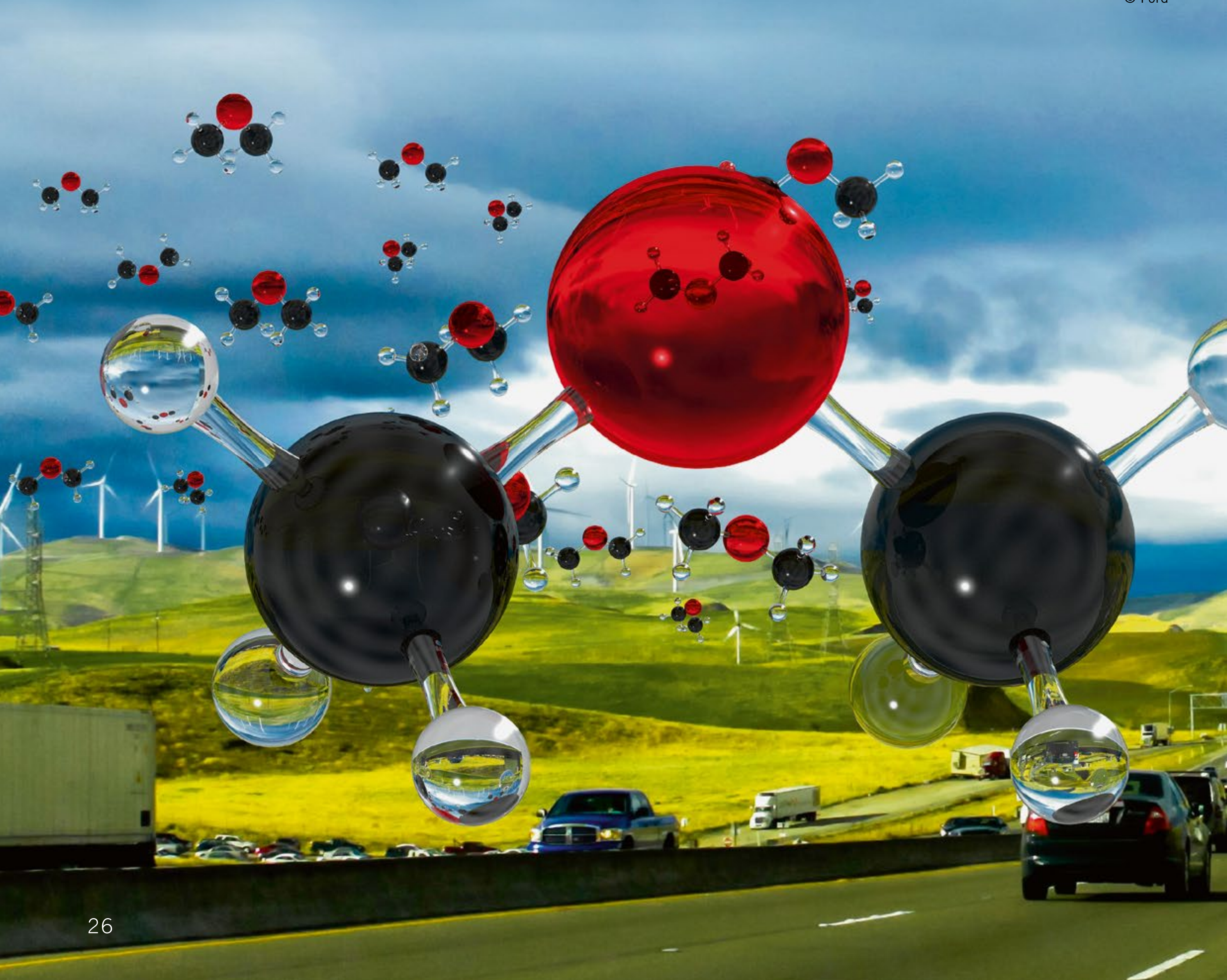
Oxygenated Fuels in Compression Ignition Engines

Internal combustion engine technology can also be applied in order to meet future greenhouse gas and pollutant emission targets, if the fuel characteristics are included as free parameters within the optimization of the powertrain. Therefore, a BMWi-funded consortium investigated oxygenated low-carbon fuels for the application in compression ignition engines, in order to study their combustion and pollutant formation properties on the engine and within a vehicle application.



MOTIVATION

The reduction and avoidance of Greenhouse Gases (GHG) has become one of the most important technology drivers for industrial applications worldwide to limit global warming. In particular the transport sector, which is one of the major contributors to GHG emissions, is in the focus of political and social debates. Additional efforts are required in order to reduce CO_2 concentration in the atmosphere and thus limit global warming. Evolutionary approaches based on internal combustion engines powered by synthetic fuels from renewable sources are promising in addition to disruptive concepts such as battery electric vehicles.

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REQUIREMENTS FOR SUSTAINABLE SYNTHETIC FUELS

Renewable synthetic fuels have become an important subject to powertrain research in recent years. However, selection the most promising fuel candidates for future vehicle applications requires specific criteria to bundle research and development resources. Those are:

- Actual CO_2 reduction: Synthetic fuels must show CO_2 reduction on the vehicle usage for short-term introduction, as CO_2 legislation is currently tank-to-wheel-based. Applications of synthetic fuels should therefore not exceed the CO_2 emission of current fossil fuels (diesel as benchmark).
- Reduction of pollutants: New fuels should not only avoid CO_2 emission, but also pollutants.
- Availability and costs: CO_2 avoidance is a problem which requires immidiate action. The development of standards for new fuels and the introduction into type approval regulations require time and effort. Therefore, standardized fuels, which are available globally and at reasonable costs, should be favored.

Hence, dimethyl ether (DME, CH_3-O-CH_3) and oxymethylene ether (OME_1, CH_3-O-CH_2-O-CH_3) were identified as suitable and promising fuels for applications in diesel engines. Thus, they were selected for the investigations.

DME AS AN ALTERNATIVE FUEL FOR COMPRESSION IGNITION ENGINES

DME was investigated from fundamental high-pressure chamber tests over single/multi-cylinder engine tests up to vehicle demonstration in order to assess its potential as diesel fuel replacement within the framework of the project.
In the following, the physical and chemical properties of DME as the most promising methyl ether will be discussed.

PHYSICAL AND CHEMICAL PROPERTIES OF DME

It should be noted that due to the high oxygen content (34.8 % m/m) and the absence of C-C bonds for DME, an almost soot-free combustion can be achieved. Consequently, higher Exhaust Gas Recirculation rates (EGR) can be utilized in order to reduce NO_x emission simultaneously. **TABLE 1** shows the physical properties of DME compared to diesel (EN590). However, the differences in calorific value and density as well as lubricity and viscosity require technical adjustments to the tank and injection system for DME usage.

DME is a liquid gas with a vapor pressure curve comparable to LPG. Therefore, LPG-compatible tanks and components can be used for the low-pressure fuel supply. The components of the system have been tested for material compatibility with DME and adapted accordingly (sealing material). DME is stored in the saturated state at a tank pressure between 2 and 7 bar depending on the ambient temperature. The liquid DME is fed from the tank into a fuel conditioning circuit at 20 to 30 bar. The fuel within the circuit is conditioned to a temperature of < 40 °C to avoid a phase change. The fuel is then fed to the injection system.

–	Unit	EN590 diesel	DME
Boiling point	°C	180–350	-24.8
Cetane number	-	51–54	55–60
Density (15 °C)	kg/m³	830	671
Oxygen content	% m/m	< 1	34.8
Latent vaporization heat (25 °C)	kJ/kg	≈ 350	466.9
Kinematic viscosity (40 °C)	mm²/s	≈ 3	0.184 (25 °C)
Wear scar diameter (HFRR)	µm	460	1180–1500 (60 °C)
Lower heating value	MJ/kg	43	27.6

TABLE 1 Fuel properties of EN590 diesel and DME (© Ford)

DME-COMMON RAIL INJECTION SYSTEM

A diesel common-rail injection system was adapted to the specific fuel properties of DME for the research work discussed in the publication. In the past, the low compression modulus and viscosity of the DME have limited the maximum injection pressure to 600 bar to minimize the risk of leakage in the high-pressure fuel pump. The poor lubricating properties of DME require an oil lubricated high-pressure pump. The reduction of the dead volumes allows an increase of the rail pressure up to 1000 bar. The magnetic servo valves are dry running to avoid wetting and dissolution of the resins. The sealing rings are made of DME-resistant FFKM material.

TABLE 2 shows an overview of the nozzle specifications that were used for the component and engine tests. The lower values for heating value and density of DME compared to diesel fuel do not allow injecting the same energy into the combustion chamber in the same time. An increase of the cross-sectional area of the nozzle is therefore necessary. The hydraulic flow rate of the injection nozzle is increased by a factor of 1.8 (PC-1.8) relative to the reference diesel nozzle (PC-D). The lower achievable rail pressure of 1000 bar compared to diesel also requires an adjustment of the flow behavior which was realized with the help of an additional nozzle scaled by a factor of 2.5 (PC-2.5). Special nozzles for the investigations in the high-pressure chamber with three holes were designed in order to improve the optical access of the individual sprays. Moreover, an additional nozzle was provided based on the results of the CFD optimizations (PC-2.1) for the single-cylinder engine investigations.

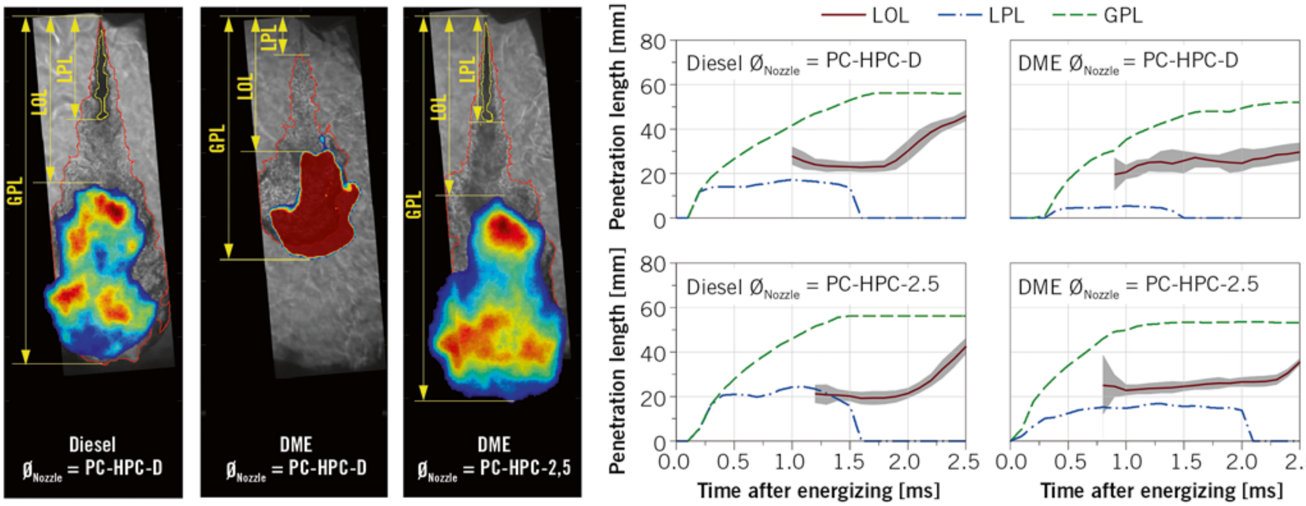
In order to characterize the influence of the increased nozzle hole diameter investigations were carried out in a High-pressure Chamber (HPC). The previously described scaled nozzles for DME were investigated in comparison to the reference nozzle (PC-HPC-D) of the passenger car diesel configuration.

The temporal development of the macroscopic spray characteristics can be observed by means of optical measurements in the high-pressure chamber. The flame lift off height (Lift-off Length, LOL) was determined in the measurements in addition to the Gas and Liquid Penetration Length (GPL and LPL). Schlieren images were used to determine the GPL and the radiation of the excited hydroxyl radical (OH*) for the LOL. The LPL of diesel fuel was determined by Schlieren images as well, while a Mie scattered light method was used for DME due to its lower density. **FIGURE 1** shows the results of the HPC measurements at a chamber pressure of p_a = 50 bar and a temperature of T_a = 840 K for diesel fuel and DME. The ambient conditions within the chamber were derived from load point-depending in-cylinder conditions close to start of injection of the reference series engine.

Nozzle	High Pressure Chamber [No. holes × spray angle/HFR (Name)]	Engine Tests [No. holes × spray angle/HFR (Name)]
Reference nozzle for diesel	3 × 154.5° / 217.5 cc/min (PC-HPC-D)	8 × 154.4° / 580 cc/min (PC-D)
Compensated by density and lower heating value (scaling factor = 1.8)	–	8 × 154.4° / 1044 cc/min (PC-1.8)
Compensated by density, lower heating value and injection pressure (scaling factor = 2.5)	3 × 154.5° / 554 cc/min (PC-HPC-2.5)	8 × 154.4° / 1476 cc/min (PC-2.5)
Nozzle derived from CFD optimization for improved DME combustion	–	8 × 145.3° / 1200 cc/min (PC-2.1)

TABLE 2 Nozzle specification for injection testing and engine testing (© Denso)

FIGURE 1 Optical investigations of the mixture preparation in the high-pressure chamber [2] (© RWTH Aachen University)



The actuation duration of the injector was t_{en} = 1000 µs and the rail pressure was set to 1000 bar. Diesel fuel shows a significantly longer liquid jet than DME when comparing the LPL using the reference nozzle (PC-HPC-D). This can be related to the lower boiling point and vapor pressure of DME. Here, the vapor pressure is very close to the ambient pressure in the chamber. The phase change of DME from liquid to gaseous occurs almost immediately, resulting in a shortened LPL. A significant influence on the LPL at DME was observed for the nozzle with the larger hole diameter (PC-HPC-2.5). While the penetration depth of the liquid diesel jet showed only a small increase, significant differences could be observed for DME due to the stronger cooling effect of the evaporating jet.

SINGLE-CYLINDER ENGINE INVESTIGATIONS

For the experimental combustion investigations, a single-cylinder engine was derived from the corresponding multi-cylinder engine. It was equipped with the adapted injection system components. Characteristic load points based on mapping data from the series diesel engine were selected. In the following, an operating point at medium load for diesel and DME will be analyzed in detail. The engine boundary conditions were not changed for the different fuels and nozzle configurations. The start of actuation and duration of the injector as well as the EGR rate were adapted depending on the fuel and nozzle size in order to keep the center of heat release constant. Only a single main injection was applied for the tests on the single-cylinder engine, in contrast to the standard mapping of the multi-cylinder engine. **FIGURE 2** shows the comparison between the diesel reference measurement and DME with two different nozzle hole diameters. The diesel baseline was measured with the reference nozzle (PC-D) of the series configuration. The adapted nozzles for compensation of the physical properties and injection pressure (PC-1.8 and PC-2.5) were investigated for DME. The EGR rate

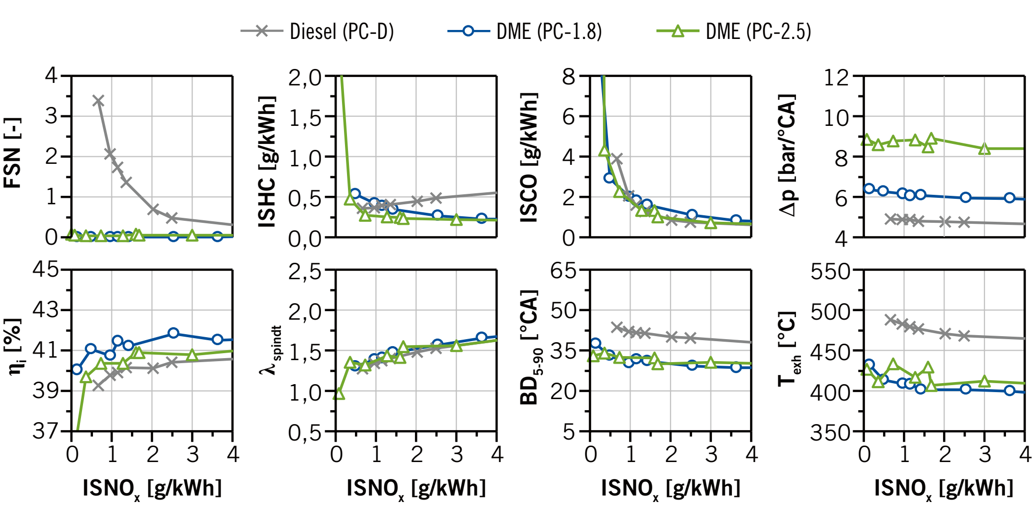


FIGURE 2 EGR variation (n = 1750 rpm; IMEP = 8.6 bar) of diesel baseline (PC-D), DME with scaled nozzle (PC-1.8 und PC-2.5) (© RWTH Aachen University)

Engine specification	Unit	Ford 1.5-l I4 88 kW TDCi
Combustion system	–	Compression ignition, four-stroke
Engine displacement	cm³	1499
Rated power	kW at rpm	88 at 3600
Maximum torque	Nm at rpm	270 at 1500–2500
Compression ratio	–	16.0
Stroke	mm	88.3
Bore	mm	73.5
Boosting system	–	VTG (single-stage)
Exhaust gas recirculation system	–	High-pressure, cooled

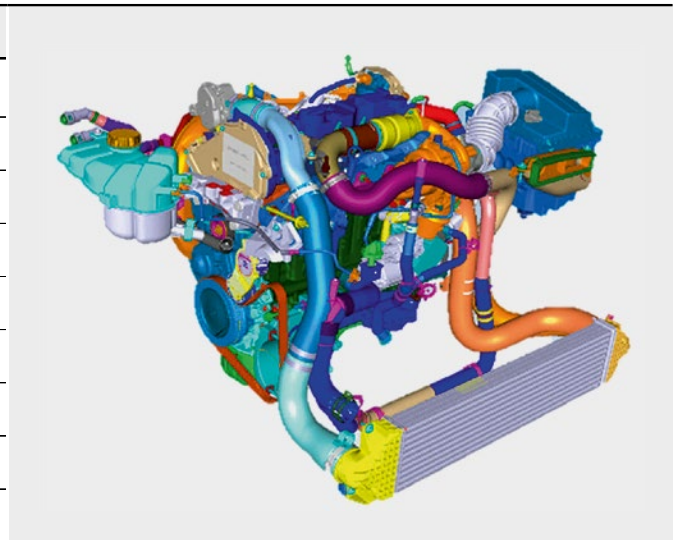


TABLE 3 Specification of the test engine (© Ford)

was adjusted to achieve different specific NO_x emission levels ($ISNO_X$).

No soot emission corresponding to the Filter Smoke Number (FSN) could be detected even at minimal NO_x emission level for both nozzle configurations with DME, **FIGURE 2**. This behavior is due to the fast mixture formation, the high oxygen content and the missing C-C bonds [1]. The missing C-C bonds lead to the formation of CO instead of soot precursors in case of high-temperature pyrolysis [3].

The specific HC (ISHC) and CO emissions (ISCO) are of similar magnitude for all three cases. The HC emission of DME is lower compared to diesel fuel only at low EGR rates. However, the maximum pressure rise rate shows differences between the three configurations. The measurements with DME show higher values despite the higher cetane number, which usually leads to a lower pressure rise. This behavior can be explained by the shorter burning duration ($BD_{5\text{-}90}$). However, the smaller nozzle (PC-1.8) showed higher EGR tolerances compared to the larger one (PC-2.5). Thus, it appears more advantageous for usage. Another big difference between DME and diesel lies in the difference regarding indicated efficiency (η_i), **FIGURE 2**. DME has a higher efficiency relative to diesel, due to the

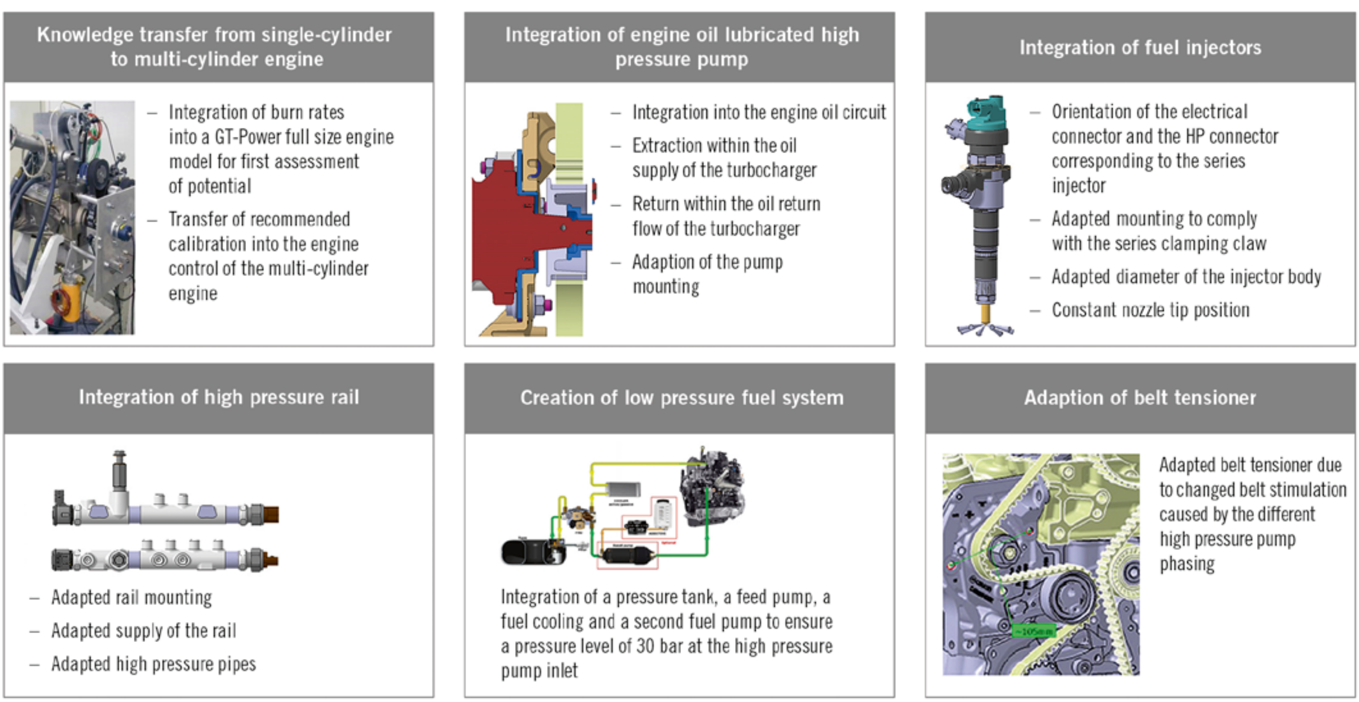


FIGURE 3 Required modifications of the engine hardware (© IAV)

shorter burning duration and lower exhaust gas temperature (T_{exh}).

MULTI-CYLINDER ENGINE INVESTIGATIONS FOR THE PASSENGER CAR DME APPLICATION

A Ford 1.5-l diesel engine was used for the engine tests on the dyno and within the vehicle. The technical specifications of the multi-cylinder engine are shown in **TABLE 3**. The engine hardware had to be adapted to the DME injection components, **FIGURE 3**. The greatest challenge was the integration of the high-pressure pump. The pump was lubricated by the engine oil due to the lack of lubricity of DME. The belt drive had to be adapted due to the different phasing position of the pump. Moreover, new sensors and actuators for controlling the high-pressure fuel system had to be integrated into the engine control system to enable engine operation under steady-state and transient conditions. The engine control system, except for the injection parameters, remained in the serial engine control unit whereas a separate second unit controlled the DME injection system. The communication between both control units is ensured by a specific CAN interface.

The measurements on the multi-cylinder engine were conducted at ten part load and three full load operating points. A representative operating point (n = 2000 rpm, BMEP = 7 bar) will be discussed for DME operation to show exemplarily the approach for calibration optimization, **FIGURE 4**. Variations of the rail pressure, the center of heat release and the combustion air-to-fuel ratio at constant NO_x emission have been conducted, while roughly maintaining the diesel reference calibration and boundary conditions. The tests with DME have been done without the use of a pilot injection in contrast to the base measurement with diesel fuel, as the single-cylinder engine test results showed promising trends from an engine noise point of view by applying low rail pressure in combination with high EGR rates, when using the smaller nozzle hole diameter.

The aim of the individual variations was to achieve a maximum possible efficiency with lowest CO emission. The turbocharger was one of the limiting factors because of the shorter burning duration and the resulting lower exhaust gas temperature of DME relative to diesel fuel. Therefore, the optimization had to be performed at a relatively high specific NO_x emission (2.0 g/kWh) in order to ensure enough flexibility regarding the VTG position.

A rail pressure reduction to 350 bar with simultaneous adjustment of start of injection showed advantages in terms of CO and efficiency. The indicated high-pressure cycle efficiency decreased with reduced rail pressure as expected, but the effective engine efficiency increased. This can be explained on the one hand by reduced friction losses due to lower drive power demand of the high pressure pump. On the other hand, the combustion duration increased in comparison to the base measurement with 720 bar. Consequently, the exhaust enthalpy increased enabling a higher turbocharger efficiency. Thus, the gas exchange losses decreased. A further reduction in rail pressure to values smaller than 350 bar did not result in further benefits. A similar trend could be observed for the variation of the center of heat release, as an advanced timing helped to increase the efficiency at similar CO emission level. The variation of the air-to-fuel ratio showed that lean combustion should take place at least at λ = 1.5. Stoichiometric combustion comparable to gasoline engines could not be achieved, due to the decrease in efficiency and the significantly increased CO emission. The latter was well above the value of modern DI gasoline engines. Additional exhaust gas components were measured with a particle counter and a Fourier Transform Infrared Spectrometer (FTIR). The CH_4 emission downstream of the diesel oxidation catalyst was not affected by the variation of rail pressure and center of heat release. However, it increased with decreasing air-to-fuel ratio and found its maximum of about 140 ppm at λ = 1.1. The further reduction of the air-fuel ratio to λ = 1.0 led to a

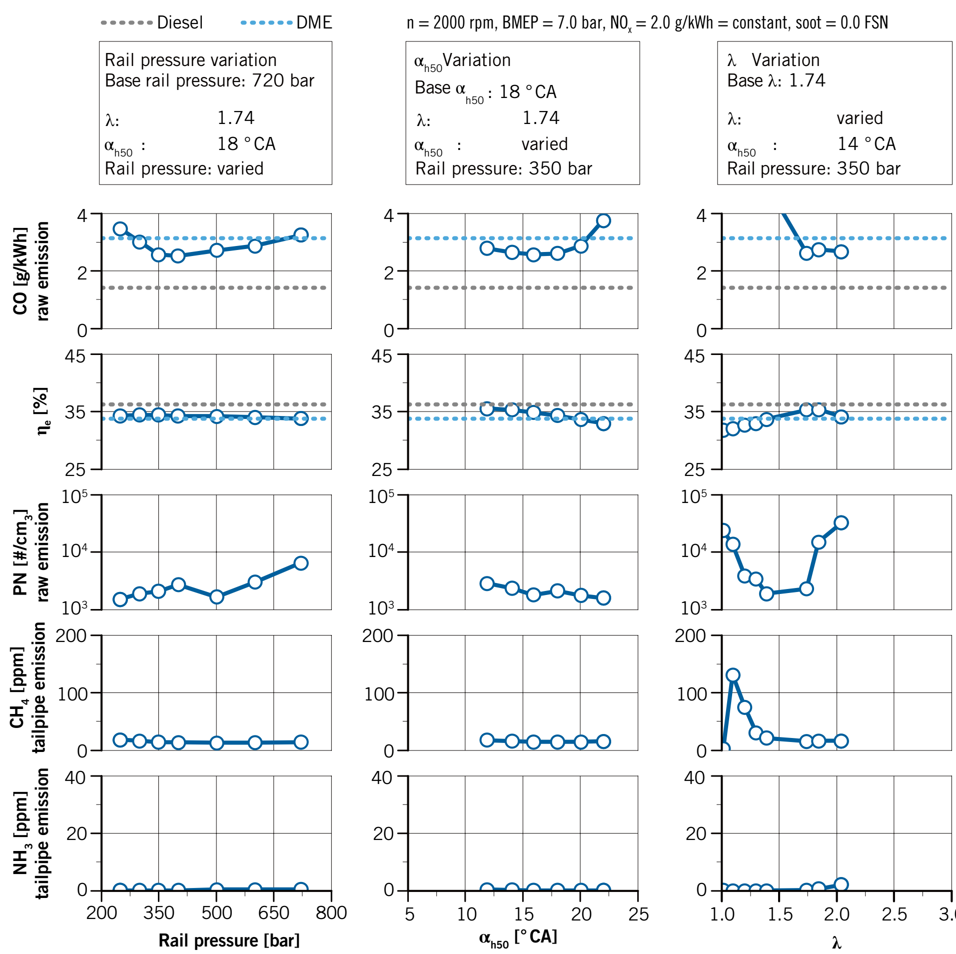


FIGURE 4 Optimization for n = 2000 rpm, BMEP = 7.0 bar at constant NO_x emission (© IAV)

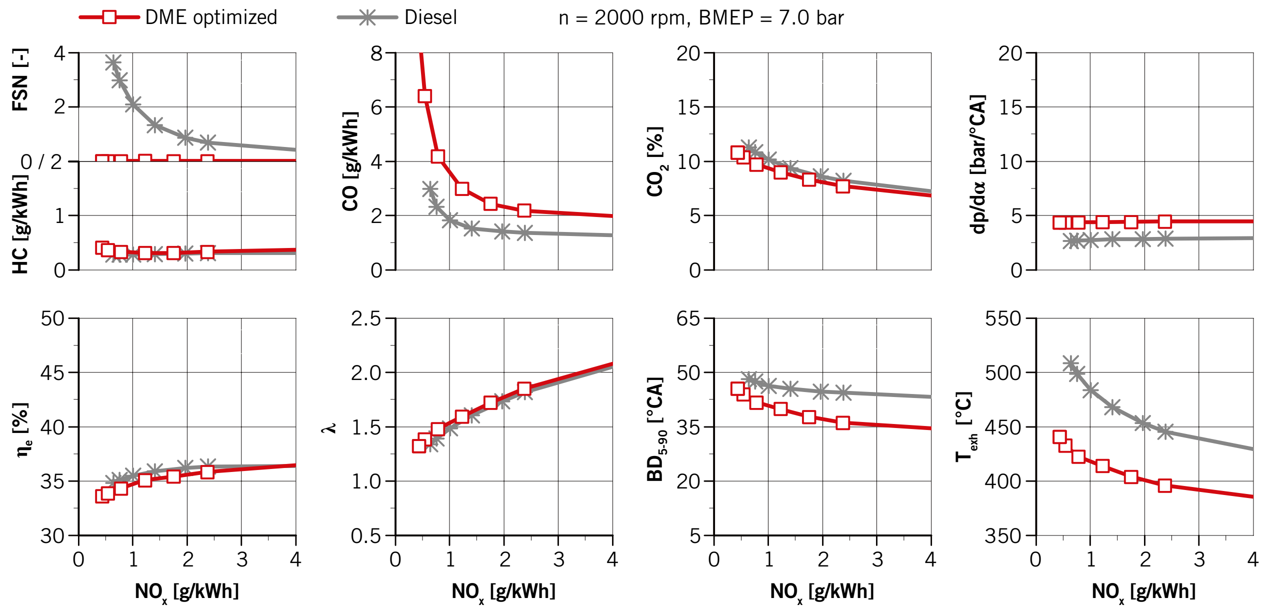


FIGURE 5 EGR-variation after optimization for n = 2000 rpm, BMEP = 7.0 bar (© IAV)

reduction of CH_4 to an emission level near the detection limit. Here, the exhaust gas temperature increased substantially (approximately 550 °C) so that the threshold value for CH_4 conversion in the catalyst was exceeded. However, this threshold value could not be reached at low loads, due to the overall lower exhaust gas temperature level. The raw emission of the Particulate Number (PN) remained well below of modern DI gasoline engines but tended to be of same magnitude or even higher at other operating points, **FIGURE 4**.

The final EGR variation was carried out based on the findings of the optimization, **FIGURE 5**. The effective engine efficiency with optimized calibration was almost the same as that of the diesel-powered counterpart. DME offered an advantage in terms of CO_2 emission due to the more favorable H/C-ratio relative to diesel fuel. The CO emission was still high. The map-wide calibration based on the optimization of the steady-state operating points, which implies the use of a deNO$_x$ system and a diesel particulate filter, is shown in **FIGURE 6**.

A demonstrator vehicle based on a Ford Mondeo with adapted engine hardware and tank system was developed for the final prove of concept. It exhib-

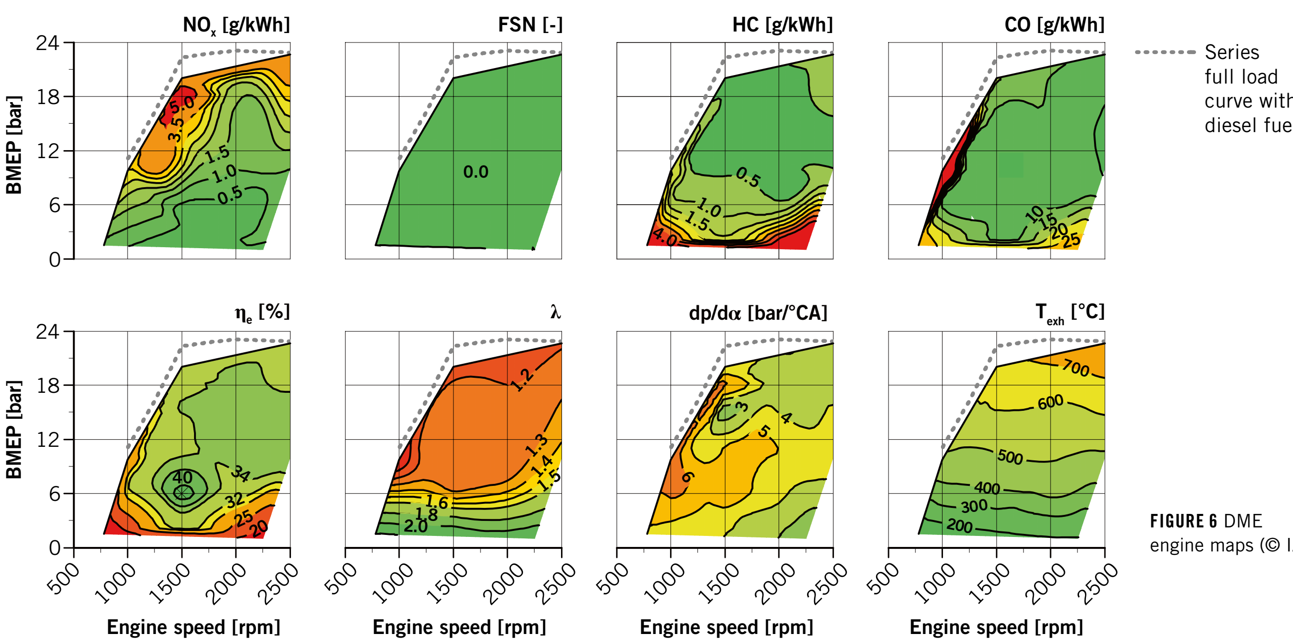


FIGURE 6 DME engine maps (© IAV)

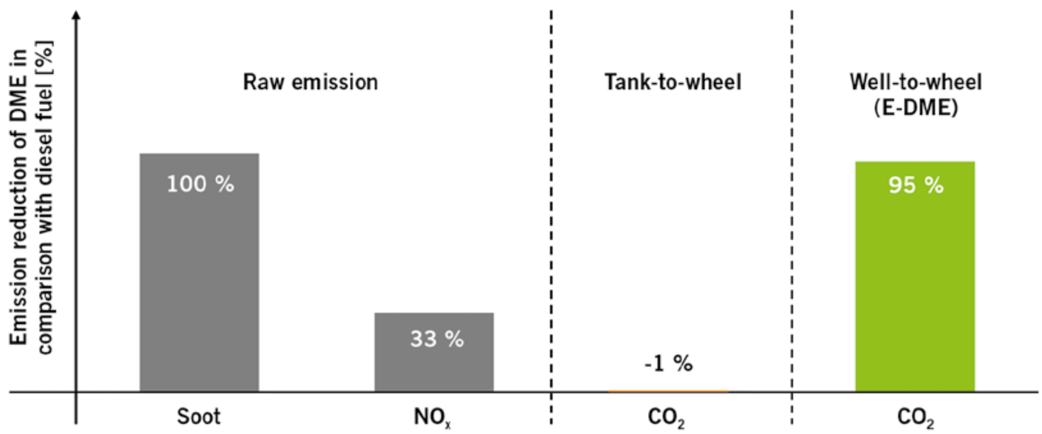


FIGURE 7 Emission improvement of DME relative to diesel in WLTC (© IAV)

ited the findings from the multi-cylinder engine investigations.

The emission potential of DME compared to diesel fuel in WLTC is shown in **FIGURE 7** and evaluated with regard to raw emissions of particulates and NO_x as well as for CO_2 emission downstream of the exhaust aftertreatment system. The soot/NO_x trade-off of the classic diesel combustion was no longer present for DME combustion. The particulate mass could be brought to zero emission level, while NO_x emission relative to diesel operation could also be reduced significantly (-33%). The CO_2 emission level remained almost the same. A low pollutant emission and nearly CO_2-neutral operation could be achieved by providing the fuel from regenerative sources (E-DME) and a well-to-wheel consideration.

CONCLUSION

As part of the technical program „Neue Fahrzeug- und Systemtechnologien" funded by the German Federal Ministry of Economics and Energy, methyl ether fuels (DME/OME_1) were investigated as sustainable fuels for diesel engine applications in passenger cars and commercial vehicles in the "XME-diesel" project. DME in particular proved to be a promising candidate as a substitute for fossil diesel fuel, considering different requirements for future sustainable fuel solutions for combustion engines (TtW CO_2 and emission reduction, available standards, global availability and low costs). The performance of DME from basic injection chamber measurements to single-cylinder and multi-cylinder tests to a demonstrator vehicles (Ford Mondeo) were analyzed and demonstrated within the project.

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