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Application of Methanol with an Ignition Promotor in a Medium Speed CI engine

Rasmus F. Cordtz, Chong Cheng, Thomas B. Thomsen and Jesper Schramm

Abstract

This work presents the early findings of methanol application in a direct injected compression ignited combustion engine. A two cylindered and four stroke BUKH DV24 engine with a geometric compression ratio of 18 is used. The engine operates at 1200 rpm, and one cylinder is fired with diesel to provide smooth engine rotation. The other cylinder - the experimental cylinder – is charged with air (at a slightly positive gauge pressure) and fired with methanol blended with a fuel additive. With 5 % (m/m) fuel additive and „early" fuel injections (around 45 crank angles before the top dead center) the fuel-blend provides a reasonable partially premixed combustion process. The combustion is, however, limited in terms of injection timing- and excess air ratio variations. Operation at later „diesel-like" injection timings requires a high compression ratio, which in this work is simulated by heating the charge air. With 5 % (m/m) fuel additive, a charge air temperature above 100 °C provides a less rapid combustion with lower efficiencies compared to the early injections.

1. Introduction

The negative climate impact forces the maritime industry to replace the conventional fuels - based on fossil feedstock's - with „green" carbon neutral fuels. „Electro fuels" are carriers of electrical energy that is stored in the chemical bonds of liquids. These fuels are produced from renewable sources, such as solar and wind power. Methanol is a potential „electro fuel". It is liquefied at atmospheric conditions, but it has a lower energy density per unit volume, and a much lower cetane number than diesel. Fuel properties of methanol and diesel are listed in Table 1. It is noted that the heat of vaporization of methanol is significantly higher than the diesel, which has an impact on its poor cetane rating of less than 5. On the positive side, the octane number is high and the flame temperature is lower than diesel, which potentially reduces thermal NO_x formation. Moreover, it has been reported that methanol-engine operation is essentially soot free [1].

Methanol is a potential sustainable fuel for combustion engines. It is well suited for SI engines due to its high octane number. However, CI-combustion of methanol is more demanding, and will often require an ignition promoter. For instance, diesel pilot fuel injection or specially designed fuel additives. Additives are developed for the application of ethanol fueled CI engines used in Swedish busses [2]. The fuel properties of ethanol and methanol are comparable, and it has been shown that additives also can be applied for methanol fueled CI engines [3], [4].

The advantage of using additives instead of pilot diesel is that the additives can be blended directly into methanol, thus avoiding two different fuel systems, and offer an attractive mono-fuel

solution. Moreover, the additives can be designed precisely for the engine/purpose in question - an advantage compared to diesel pilot injection, where some of the unwanted emissions, like particulates, may be an issue, and the sustainability of the solution can be questioned. On the other hand, diesel is cheaper than the additives.

	Methanol	Diesel
Liquid density [kg/m^3]	798	840
Lower heating value [MJ/kg]	20.1	42.7
Cetane number [-]	< 5	38-53
Octane number [-]	109	15-25
Heat of vaporization [kJ/kg]	1089	250
Adiabatic flame temp* [°C]	1949	≈2100
Stoichiometric air fuel ratio [-]	6.5	14.6
Lower flammability limit [vol %]	6	0.5
Higher flammability limit	36	7.5

Table 1. Fuel properties of methanol and diesel

Methanol has been used as a minor fuel component (MTBE) for SI engines, and for CI engines methanol has been used to produce FAME (a biodiesel) which is produced from vegetable oil, animal waste etc. High concentration methanol blends (M85) can be used in so called FFV's (Flex Fuel Vehicles) that can operate on gasoline as well. In recent time GEM (Gasoline, Ethanol, Methanol blends) has been considered for some markets. The SI engines behind this idea needs similar operation control as FFV's. In China methanol is used in different blends ranging from M5 to M100. When considering methanol use as gasoline component, corrosion inhibitors, co-solvents, and alcohol compatible materials in vehicles are needed to resist phase separation, to maintain stability and safety.

For marine purposes, application of methanol in CI engines is relevant, since most engines in this sector are diesel engines. By modifying these engines to methanol operation, one gain the advantage of being able to run the ship on diesel fuel as a back up. Therefore, it is essential to develop these rather large engines into methanol operation. The largest engines, 2-stroke slow-speed engines are already available [5] and smaller engines are on the way. This study can be seen as part of this process.

An important recent study [6] looked into the possibilities of applying methanol in smaller CI engines. The study was a collaboration between multiple partners conducting different experiments on different engine setups. Common for all experiments was that they were conducted on Scania heavy-duty engines (a modified SCANIA D13 or a SCANIA DC9 alcohol engine with a geometric compression ratio of 27:1.) with slight modifications to work on a test bench. Different types of ignition and methanol engine concepts were tested and compared with traditional diesel combustion. It was found that methanol blended with an additive as ignition improver (MD95) easily can be applied with today's fuel injection patterns to reduce emissions (soot and NOx) and GHG (Green house gasses) for smaller vessels (250-1200 kW). On the longer term a mode-shift to PPC/DI-SI with an oxidation catalyst is suggested. This will further decrease emissions, including GHG's together with the lowest operational cost compared to other fuels tested. The compression ratio (CR) of installed marine engines is normally well below 27:1. Heating the charge air could compensate for this, but it may require additional installations.

This work examines the use of methanol blended with a fuel additive (monofuel) that is burned in a light duty medium speed CI engine with CR of 18. The purpose is to clarify the combustion characteristics and limitations of additivated methanol fuel blends. The research is conducted on a modified BUKH DV24 diesel engine. One of two cylinders - the experimental cylinder - is fuelled with the directly injected methanol fuel blend (typically 5 % m/m additive) The other cylinder overcomes general engine losses, and provides smooth operation. Modifications of the engine include a charge air system, to independently pre-heat and pressurize the charge air, and a gasoline fuel injector for the injection of the fuel-blend into the experimental cylinder.

2. Experimental

The experiments are performed on the two-cylinder and four stroke CI engine specified in Table 2. The drive shaft is connected to an eddy current brake that controls the engine speed. One cylinder - the diesel cylinder – is naturally aspirated, operates at full throttle, and is fueled with diesel through the original manufacturer fuel injector (camshaft controlled injection timings). The power from the diesel cylinder overcomes the inherent engine friction/mechanical losses and provides a stable engine operation. The other cylinder - the experimental cylinder - is fueled with the additivated methanol fuel-blend through a piezo-electric gasoline (Bosch) fuel injector. This type of injector enables the needed fuel mass per cycle at the relatively mild fuel injection pressure of 180 bar. The fuel additive is originally designed/formulated to promote ignition of ethanol in a CI engine. A small 1 liter container (pressurized with 180 bar nitrogen) is filled with the fuel-blend. From the container, the fuel-blend is injected into the experimental cylinder at variable timings, controlled by an ECU. The fuel consumption is measured with a Coriolis mass flow meter. The charge air flow to the experimental cylinder is supplied from a separate compressed air system, and passes an expansion valve to control the inlet pressure. A ceramic air heater is used to heat the charge air to a desired inlet temperature. A Kistler pressure transducer measures the experimental cylinder pressure at a sample resolution of 0.1 crank angle degrees. An average pressure profile (based on 50 consecutive engine cycles) is used for the subsequent data treatment and thermodynamic calculations. The exhaust streams of the two cylinders are separated, in order to analyze the exhaust from the burned methanol fuel-blend.

Rated power	24 HP
Type	Diesel, four stroke
No of cylinders	2
Bore/Stroke	85 mm/85 mm
Geometric compression ratio	18
Engine speed (in this work)	1200 rpm

Table 2. BUKH DV24 engine specifications

3. Calculations

In order to analyze the experimental data, essential properties like the chemical heat release rate (HRR) of the fuel combustion, and the mean cylinder gas temperature T with respect to the crank angle degree (CAD) are calculated. For the purpose, an energy balance of the trapped cylinder

gas (eq. 1) is applied. U is the internal energy of the gas (considered for the part of the cycle where the intake and the exhaust valves are closed) that is modeled as a single homogenous gas volume. The gas is defined by the thermal properties (u, h and c_v) of air. These are determined from thermodynamic polynomials. T is calculated with the differentiated ideal law (eq. 3). The heat transfer/loss to the cylinder walls $\frac{dQ_{wall}}{d\theta}$ is determined from Woschnis heat transfer correlation [7]. The cylinder gas work ($p\frac{dV}{d\theta}$) is determined from the averaged experimental cylinder gas pressure (p), and the cylinder volume gradient ($\frac{dV}{d\theta}$) in eq. 4. A trigonometric function (eq. 5) is used to model the fuel injection rate ($\frac{dm_f}{d\theta}$). It is assumed, that the fuel evaporates and mix instantly with the cylinder gas upon injection. A „blow-by" rate ($\frac{dm_{bb}}{d\theta}$) is approximated (using the same set of equations, where $\frac{dQ_{HRR}}{d\theta}$ = 0) from an experimental motoring pressure profile, and is considered for the mass conservation ($\frac{dm}{d\theta}$) in eq. 6

$$\frac{dU}{d\theta} = \frac{dQ_{HRR}}{d\theta} + \frac{dQ_{wall}}{d\theta} - p\frac{dV}{d\theta} + \frac{dm_f}{d\theta}h_{fuel} - \frac{dm_{bb}}{d\theta}h_{bb} \quad (1)$$

$$\frac{dU}{d\theta} = \frac{d(m \cdot u)}{d\theta} = mc_v\frac{dT}{d\theta} + u\frac{dm}{d\theta} \quad (2)$$

$$\frac{dT}{d\theta} = \frac{p\frac{dV}{d\theta} + V\frac{dp}{d\theta} - R \cdot T\frac{dm}{d\theta}}{R \cdot m} \quad (3)$$

$$\frac{dV}{d\theta} = \frac{\pi B^2}{4}L_R\left(\sin(\theta) + \frac{L_R}{2L_C}\sin(2\theta)\right) \quad (4)$$

$$\frac{dm_f}{d\theta} = m_{f,cycle} \cdot \frac{\pi}{2\Delta\theta}sin\left(\pi\frac{\theta-\theta_0}{\Delta\theta}\right) \quad (5)$$

$$\frac{dm}{d\theta} = \frac{dm_f}{d\theta} - \frac{dm_{bb}}{d\theta} \quad (6)$$

4. Results and Discussions

4.1 Test series 1 - „Diesel like" injection timings (with heated charge air)

To illustrate the hampered „cetane quality" of methanol, fuel-blends with down to 3.5 % (m/m) fuel additive are injected into the experimental cylinder at a „diesel like" injection timing of 10 crank angles before the top dead center (CA BTDC). The charge gas pressure (p_c) is 1.1 bar and the overall excess air ratio (λ) is 3. A lower additive fraction (y_{add}) requires a higher cylinder gas temperature at the start of the fuel injection (SOI) in order to facilitate the auto ignition, and to provide a stable combustion (with acceptably low cycle to cycle variations). The modeled mean cylinder gas temperature at the SOI (T_{SOI}) with respect to y_{add} is depicted in Figure 1. When y_{add} < 5 % (m/m), the required charge air temperature (T_c) exceeds 100 °C. This yields a T_{SOI} of above 1000 K.

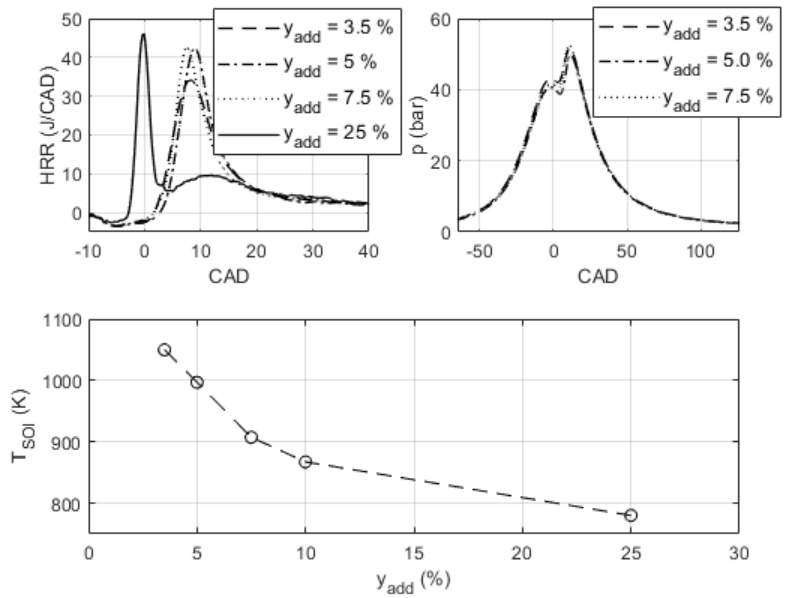

Figure 1. Upper figures: Fuel heat release rates (HRR) and cylinder pressure profiles (p) with respect to the crank angle degree (CAD) at different fuel additive fractions (y_{add}). Lower figure: Mean cylinder gas temperature at the start of the fuel injection (T_{SOI}) versus y_{add}.

Except for the case where y_{add} = 25 % (m/m) the HRR-traces in Figure 1 illustrate that the start of combustion (SOC) is basically after the end of the fuel injection (EOI). The EOI is around -2 CA BTDC and the injection period therefore barely overlaps the HRR-traces. The single coinciding HRR-peaks (for y_{add} ≤ 7.5 %) indicate further, that the fuel-blends mainly burn as a partially premixed combustion (PPC) process, enabled by adequate fuel-air mixing prior to the SOC. Secondary mixing controlled combustion may occur, as well as late combustion of unreacted fuel „pockets" during the expansion stroke. A higher fuel injection pressure, combined with the use of a true diesel injector (instead of the gasoline injector) may improve fuel atomization, fuel air mixing and promote the ignition. Nonetheless, the much lower ignition delay with y_{add} = 25 % indicates, that the delay for the remaining and more „pure" methanol blends, are considerably inhibited by the unfavorable thermodynamic- and ignition properties of methanol. The different fuel-blends yield reasonable cylinder pressure profiles. However, in combination with use of charge air heating the thermal efficiencies are lowered compared to more explicit PPC-operation with earlier fuel injections and „unheated" charge air (presented in the following sections).

4.2 Test series 2 - Start of injection sweep („early" fuel injections)

Test series 2 illustrates how methanol fuel-blends with 5 % (m/m) fuel additive burn at „early" injection timings (p_c= 1.1 bar, λ = 2) and with a T_c of 40 °C. The pressure- and the HRR-traces in Figure 2 are plotted against the crank angle degree (CAD). The indicated- (η_i) and the combustion efficiencies (η_c) are also presented, and are plotted against the SOI. η_i is normalized by the „best efficiency" (at SOI = 45 CA BTDC) and η_c is characterized as the amount CO_2 relative to the amount of CO_2+CO in the exhaust. This is somewhat erroneous, since the exhaust gas also contain carbon in the form of e.g. hydrocarbons - particularly at the latest fuel injection timings. However, the trend of the combustion efficiency is reasonably caught in this way.

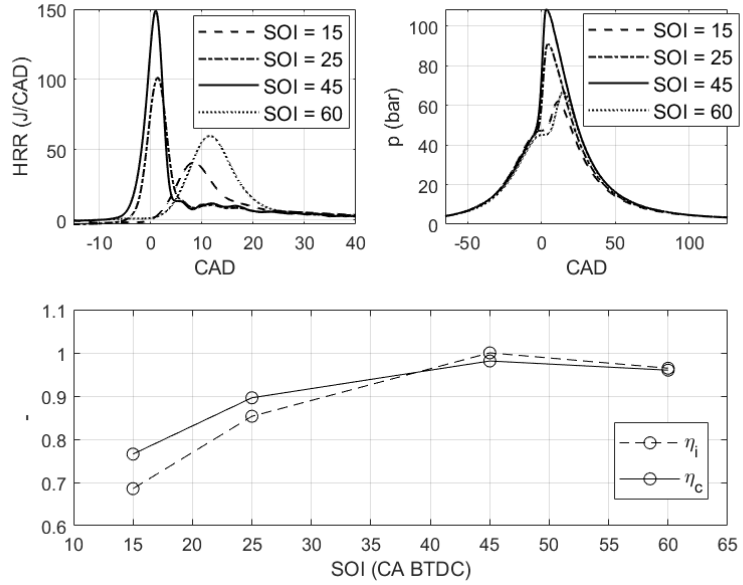

Figure 2. Upper figures: Fuel heat release rates (HRR) and cylinder pressure profiles (p) versus the crank angle degree (CAD) for the different fuel injection timings (SOI) shown in the figure legends. SOI is in CA BTDC. Lower figure: Normalized indicated efficiencies (η_i) and combustion efficiencies (η_c) with respect to SOI. η_i is normalized by the „best efficiency" at SOI = 45 CA DTDC.

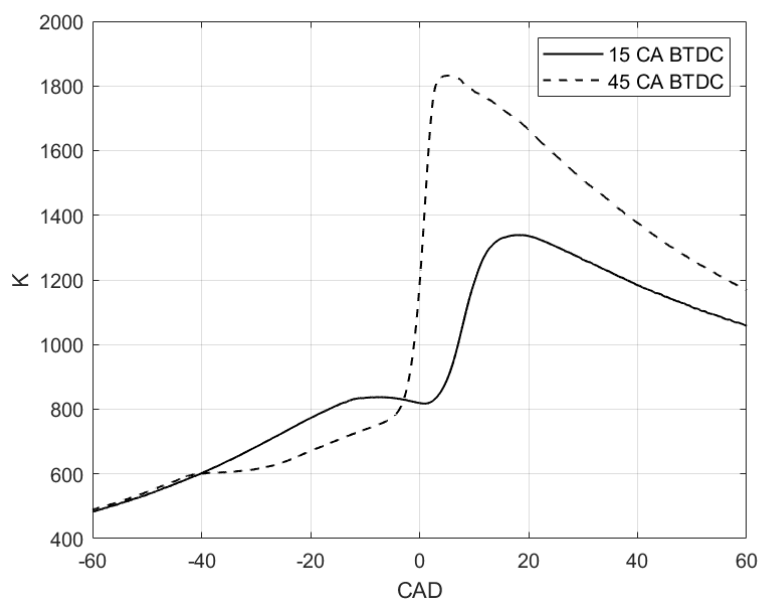

Figure 3. Calculated, mean cylinder gas temperature profiles versus the crank angle degree (CAD) for the injection timings shown in the figure legends.

Figure 3 illustrates the T-profiles for two of the applied fuel injection timings. When SOI = 45 CA BTDC the temperature increment ceases while the fuel injection and evaporation occurs during the compression stroke. The fuel ignites hereafter; a few CAD's before the top dead center (TDC @ 0 CAD) when T is ≈ 765 K. The resulting HRR-trace (Figure 2) is characterized by a single and

tight peak, that indicates a rapid PPC-process, involving a high- peak pressure and pressure rise rate (PRR). But no occurrence of „engine knocking". When the SOI is at 15 CA BTDC, T is higher than 800 K at the point where the fuel starts being injected. Yet, evaporative cooling and insufficient fuel-air mixing still hampers the ignition. The SOC is located after the TDC, which in this case yields a poor fuel utilization (η_c) as well as a low η_i. Advancing the SOI to 60 CA BTDC may provide more time for the fuel-air mixing prior to the SOC. However, this timing yields the longest ignition delay, and a much less rapid combustion relative to the „best case" (at 45 CA BTDC). It is presumed that a more complete fuel-air mixing affects the flammability/reactivity of the combustible mixture.

4.3 Test series 3 - Overall excess air ratio/power sweep („early" fuel injections)

The results of test series 3, in Figure 4, illustrate how methanol fuel-blends with 5 % (m/m) fuel additive burn at altered λ-values between 1.5 to 3 (p_c= 1.2 bar, T_c = 40 °C) when the „optimum" injection timing at 45 CA BTDC (found in series 2) is applied. The λ-value (or fuel mass/engine power) is modified through the fuel injection period. The evaporative cooling of injected fuel per cycle is therefore lowest at higher λ-values. From a simple thermodynamic point of view, this could advance the SOC, due to a higher mean gas temperature. The HRR-profiles show, however, that the ignition delay increases, and the combustion is relatively slow (and inefficient) at λ = 3. This may be explained by imposed flammability limits that impedes low load operation. Rapid partially premixed combustion applies to λ-values ≤ 2.5 where the „coinciding" HRR-traces yield comparable efficiencies. A higher mean gas temperature and rapid PRR at λ = 1.5 seems, however, to increases the heat loss, and reduce the indicated efficiency.

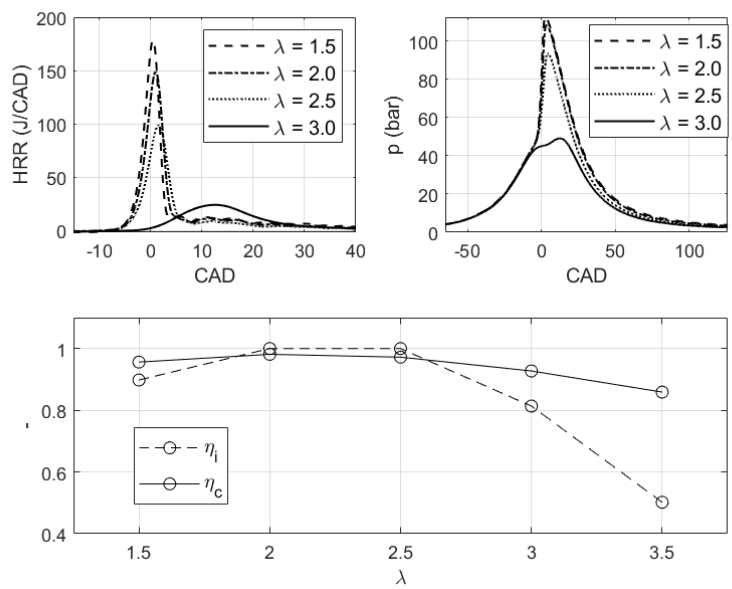

Figure 4. Upper figures: Fuel heat release rates (HRR) and cylinder pressure profiles (p) with respect to the crank angle degree (CAD) at different excess air ratios (λ). Lower figure: Normalized indicated efficiencies (η_i) and combustion efficiencies (η_c) with respect to λ. η_i is normalized by the „best efficiency" at λ = 2.

4.4 Test series 4 – Charge air temperature sweep („early" fuel injections)

The results of test series 4, in Figure 5, illustrate how methanol fuel-blends with 5 % (m/m) fuel additive burn when T_c is altered in the range between 50 - 90 °C (p_c= 1.1 bar, λ = 2.5) and when the „optimum" injection timing (at 45 CA BTDC, found in test series 2) is applied. Since the λ-value is constant, the density of the trapped cylinder gas reduces slightly with T_c. On the other hand, it might be expected that a higher T_c advances the SOC, due to a faster fuel evaporation and improved chemical kinetics. According to the HRR-tracces, the SOC advances only weakly by a higher T_c, and the traces are nearly identic. The weak impact of T_c therefore indicates, that the „preparation/mixing" of the combustible fuel-air vapors essentially determines the ignition delay. η_c improves slightly by elevating the T_c, but it increases the heat loss and lowers η_i a little.

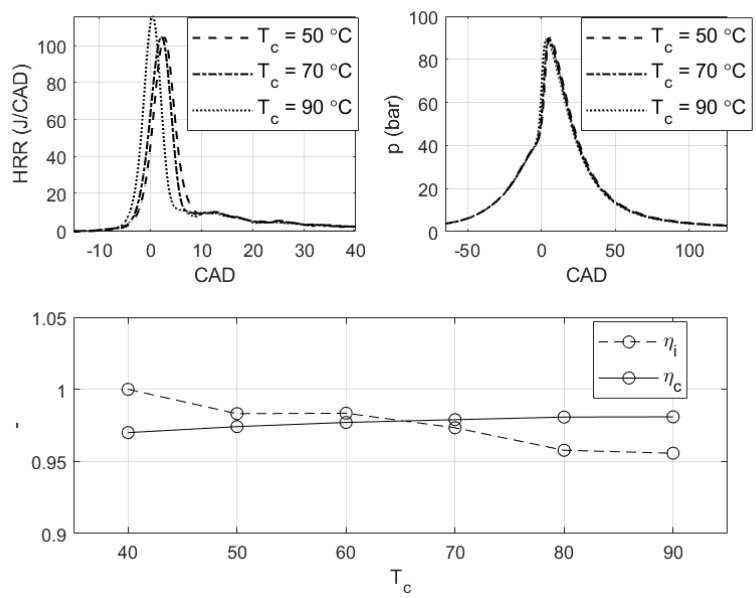

Figure 5. Upper figures: Fuel heat release rates (HRR) and cylinder pressure profiles (p) with respect to the crank angle degree (CAD) at different charge air temperatures (T_c). Lower figure: Normalized indicated efficiencies (η_i) and combustion efficiencies (η_c) with respect to T_c. η_i is normalized by the best efficiency at T_c = 40 °C.

Conclusion

The experiments with a light duty CI engine have shown, that the direct injection of methanol blended with 5 % (m/m) fuel additive (normally used as ignition improver for ethanol operation in CI engines) can work as a mono-fuel solution (with some limitations). At a geometric CR of 18, 1200 rpm, $\lambda \approx 2$, T_c= 40 °C and $p_c \approx$ 1.1 bar, there exists a „best operating point", at a rather early SOI of 45 CA BTDC, that yields a PPC-process. Altered SOI's and λ-values „outside" the „best point" can significantly reduce the thermal- and combustion efficiencies, due to flammability limits and inefficient fuel utilization. „Diesel-like" injection timings (in this case at SOI = 10 CA BTDC) with down to 3.5 % (m/m) fuel additive are shown to burn reasonably. The SOI-timing requires,

however, a T_c of well above 100 °C. The need for heating reduces the indicated efficiency by ≈ 15 % compared to the „best point”.

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Nomenclature

Symbol		unit
B	Cylinder bore	m
c_v	Specific heat capacity of cylinder gas, const. volume	J/kg-K
h	Specific enthalpy of the cylinder gas	J/kg
h_{bb}	Specific enthalpy of blow by gas	J/kg
h_{fuel}	Specific enthalpy of liquid fuel	J/kg
L_R	Length of connecting rod	m
L_C	Length of crank shaft arm	m
m	Mass of cylinder gas	kg
m_{bb}	Blow by mass	kg
m_f	Fuel mass	kg
$m_{f,cycle}$	Injected fuel mass per cycle	kg
p	Cylinder gas pressure	Pa
p_c	Charge air pressure	Pa
Q_{HRR}	Chemical heat release rate of combustion	J/CAD
Q_{wall}	Cylinder gas heat loss	J/CAD
R	Specific gas constant of air	J/kg-K
T	Mean cylinder gas temperature	K
T_c	Charge air temperature	K
T_{SOI}	Mean cylinder gas temp. at the start of fuel injection	K
u	Specific internal energy of cylinder gas	J/kg
U	Internal energy of cylinder gas	J
V	Cylinder gas volume	m^3
y_{add}	Mass fraction of fuel additive in fuel blend	kg/kg
η_c	Combustion efficiency	-
η_i	Relative indicated efficiency	-
θ	Crank angle position	CAD
θ_0	Start of injection	CAD
$\Delta\theta$	Injection period	CAD
λ	Overall excess air ratio	-

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