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The potential of methanol as a fuel for flex-fuel and dedicated spark-ignition engines

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HIGHLIGHTS

- ▶ Methanol and gasoline operation are compared on two atmospheric flex-fuel engines.
- ▶ Methanol enables a relative efficiency increase of 10% while reducing NO_x and CO_2.
- ▶ Throttleless load control strategies using lean-burn and EGR are evaluated for methanol.
- ▶ EGR strategy allows to increase part load efficiency while maintaining low emissions.
- ▶ A high CR, turbo engine with this strategy reaches diesel-like efficiencies on methanol.

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ABSTRACT

Using light alcohols in spark-ignition engines can improve energy security and offers the prospect of carbon neutral transport. The properties of these fuels enable considerable improvements in engine performance and pollutant emissions. Whereas most experimental studies have focused on ethanol, this paper provides experimental results gathered on various methanol-fuelled engines. A comparison against gasoline on two flex-fuel engines yielded relative efficiency benefits of about 10% for methanol thanks to more isochoric combustion, less pumping, cooling and dissociation losses. Lower combustion temperatures allowed to reduce engine-out NO_x by 5–10 g/kWh. The CO_2 values dropped by more than 10%. Alternative load control strategies, employing mixture richness or exhaust gas recirculation (EGR) to control load while keeping the throttle wide open, were compared on a single cylinder engine. The EGR strategy seems preferable as it allows to increase part load efficiency up to 5% without sacrificing in terms of tailpipe emissions. Finally, this load control strategy of choice was applied to a turbocharged, high compression ratio engine to demonstrate that methanol can be used in dedicated engines with diesel-like efficiencies (up to 42%) and emission levels comparable to or lower than gasoline engines.

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1. Introduction

1.1. Renewable transportation fuels

Hydrogen and electrification are two approaches to de-carbonizing transport that receive a lot of attention these days. However, their inherently low energy densities and high associated infrastructure costs make it unlikely that these solutions will become competitive with liquid fuels in the near future. Conversely, sustainable liquid alcohols, such as ethanol and methanol, are largely compatible with the existing fuelling and distribution infrastructure and are easily stored in a vehicle.

Biofuels, such as ethanol, can only constitute part of our energy supply because of the limited area of arable land [1]. Methanol, on the other hand, can be produced from a wide variety of renewable sources (e.g. gasification of wood, agricultural by-products and waste products [2]) and alternative fossil fuel based feed stocks (e.g. coal and natural gas [3]). A number of workers have even proposed a sustainable closed-carbon cycle where methanol is synthesized from renewable hydrogen and CO_2 from power plants [4] or the atmosphere [5]. Methanol can be used in low-cost internal combustion engines with only minor adjustments to ensure material compatibility [6] and enables increased engine performance compared to gasoline, as will be explained below.

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1.2. Methanol as a fuel for internal combustion engines

Methanol has the potential to increase engine performance and efficiency, thanks to a variety of interesting properties. Properties of gasoline, methanol and ethanol relevant to their use in internal combustion engines are summarized in Table 1. The main favourable properties of light alcohols include:

- High heat of vaporization, which in combination with the low stoichiometric air to fuel ratio leads to high degrees of intake charge cooling as the injected fuel evaporates.
- Elevated knock resistance, which is partly due to the considerable cooling effect. This opens opportunities for increased power and efficiency by applying higher compression ratios, optimal spark timing and aggressive downsizing.
- High flame speeds, which enable qualitative load control using mixture richness or varying amounts of EGR.

These properties and their favourable effects are most pronounced for methanol. A more extensive discussion on their implications for engine performance and emissions can be found in earlier publications [7,8].

Despite its interesting properties, the use of methanol as a fuel has met resistance due to toxicological and fire safety concerns. As discussed in several recent reviews [3,9], this issue is often overstated as methanol's toxicity (to human health and the environment) is on the same order as other fuels being considered as gasoline and diesel substitutes. In terms of fire safety methanol is substantially less hazardous than gasoline and for this reason it has been the preferred racing fuel in the US for many years.

1.3. Published work on alcohol engines

Published experimental work on alcohol engines indicates that the increase in power and efficiency depends on whether an engine is designed for alcohol operation only or for flexible fuel operation on both gasoline and alcohol.

In dedicated alcohol engines, the elevated knock resistance can be used to raise the compression ratio (CR) (to levels of 12:1 and above) without the need for spark retarding to avoid knock. Thanks to this design change Ford was able to obtain 20% more power and 15% higher efficiency from their M85 (a mixture of 85 vol.% methanol and gasoline) Escort model compared to its gasoline equivalent, and this was in 1981 [10]. Clemente et al. reported similar figures for a more recent dedicated ethanol engine designed for the Brazilian market [11].

The elevated flame speed and wide flammability limits of alcohols open some alternative options for load control, especially for methanol. Pannone and Johnson [12] have published results from an experimental turbocharged lean-burn methanol engine. The reported brake thermal efficiencies are up to 14% better than for stoichiometrically fuelled engines with throttled load control [15]. Engine-out CO emissions were reduced by over 50%, while unburned fuel emissions mildly increased. The tailpipe NO_x penalty of the lean burn strategy reached up to 150%, making the practical use of such a strategy questionable.

More interesting is to exploit the wide dilution limits of alcohols in a strategy using stoichiometric fuelling and exhaust gas recirculation (EGR) to control the load, thus reducing throttling losses and enabling three-way catalyst aftertreatment. Brusstar et al. demonstrated this using a 1.9 l turbocharged diesel engine with a CR of 19:1 that was converted for SI operation on methanol [13]. The high compression ratio enabled peak brake thermal efficiencies higher than the baseline diesel engine (40%) for operation on methanol (42%). Elevated levels of EGR (up to 50%) were used to spread the high efficiency regions to part-load operating points. Throttleless operation was possible down to a BMEP (brake mean effective pressure) of 6 bar.

Flexible fuel vehicles (FFVs) were developed during the 1980s to avoid the chicken and egg problem associated with the lack of alcohol refuelling stations. The lower knock resistance of gasoline meant the CR could no longer be increased a lot. Still FFVs attained about 5% more power and efficiency due to increased volumetric efficiency, lower flow losses and more isochoric combustion [10]. Today, active knock control and aggressive spark retarding make it possible to combine high CR and flexible fuel operation.

Bergström et al. took full advantage of the evaporative cooling effect by using E85 in a production turbocharged flex-fuel engine with direct injection [14]. Operation on E85 enabled the application of optimal ignition timing, increasing the engine's power by 20%. The mean brake thermal efficiency over a NEDC (New European Driving Cycle) was improved by over 5% compared to operation on gasoline.

As might be clear from the cited references, most of the recent work has focused on ethanol, whereas quantitative data for methanol-fuelled engines remains scarce. The present paper aims to demonstrate the potential of neat methanol-fuelled engines by analysing existing [15,16] and new, unpublished experimental results gathered on various engine test benches in terms of power, efficiency, greenhouse gas and pollutant emissions. In the first part of this paper the efficiency and noxious emissions are compared between gasoline and neat methanol operation on two normally aspirated flex-fuel engines. Next, the potential of two alternative load strategies are tested on a normally aspirated single cylinder flex-fuel engine. The strategies under consideration are wide open throttle lean burn operation and wide open throttle, stoichiometric operation with varying amounts of EGR to control load. Finally the wide open throttle EGR strategy is applied to a high compression ratio engine, representing the potential of dedicated methanol engines.

Table 1
Properties of typical gasoline, methanol, ethanol and hydrogen relevant to internal combustion engines [28,29,8].

Property	Gasoline	Methanol	Ethanol
Chemical formula	Various	CH_3OH	C_2H_5OH
Oxygen content by mass (%)	0	50	34.8
Density at NTP (kg/l)	0.74	0.79	0.79
Lower heating value (MJ/kg)	42.9	20.09	26.95
Volumetric energy content (MJ/l)	31.7	15.9	21.3
Stoichiometric air to fuel ratio (kg/kg)	14.7	6.5	9
Energy per unit mass of air (MJ/kg)	2.95	3.12	3.01
Research octane number (RON)	95	109	109
Motor octane number (MON)	85	88.6	89.7
Sensitivity (RON–MON)	10	20.4	19.3
Boiling point at 1 bar (°C)	25–215	65	79
Heat of vaporization (kJ/kg)	180–350	1100	838
Reid vapour pressure (psi)	7	4.6	2.3
Mole ratio of products to reactants[a]	0.937	1.061	1.065
Flammability limits in air (λ)	0.26–1.60	0.23–1.81	0.28–1.91
Laminar flame speed at NTP, λ = 1(cm/s)	28	42	40
Adiabatic flame temperature (°C)	2002	1870	1920
Specific CO_2 emissions (g/MJ)	73.95	68.44	70.99

[a] Includes atmospheric nitrogen. NA: not available. NTP: normal temperature (293 K) and pressure (101325 Pa).

2. Experimental set-up and procedures

The main specifications of the three engines and the corresponding measurement equipment used in the current study are summarized in Tables 2 and 3.

Table 2
Engine specifications.

Engine type	Volvo 1.8 l	Audi/NSU	VW 1.9L TDI
Cylinders	4 in-line	1	4 in-line
Valves	16	2	8
Valvetrain	DOHC	OHC	OHC
Bore	83 mm	77.5 mm	79.5 mm
Stroke	82.4 mm	86.4 mm	95.5 mm
Displacement	1783 cc	407.3 cc	1896 cc
CR	10.3:1	10.17:1	19.5:1
Injection	PFI	PFI	PFI
Induction	Atmospheric	Atmospheric	Turbocharged
ECU	MoTeC M800	MoTeC M4	MoTeC M800

Table 3
Measurement equipment.

Engine type	Volvo 1.8 l	Audi/NSU	VW 1.9L TDI
Cylinder pressure	Kistler 6118AFD13	AVL QC34C	AVL GH12P
Intake pressure	Kistler 4075A10	Kistler 4075A10	Kistler 404515V200S
Exhaust pressure	Kistler 4075A20	Kistler 4075A20	Kistler 4049A10S
Crank angle encoder	Kistler COM2611	Kistler COM2611	AVL 365C01
Fuel flow	gravimetric	gravimetric	Bronkhorst M55-AGD-55-0
Air flow	Bosch MAF	Bronkhorst F-106BZ-HD-01-V	Bronkhorst F-106CI-ABD-01
O_2	Maihak Oxor-P S710	Maihak Oxor-P S710	–
CO, CO_2, NO_x	Maihak Multor 610	Maihak Multor 610	Hermann HGA 400

2.1. Test engines

2.1.1. 1.8 l four-cylinder engine

A production type Volvo four cylinder gasoline engine with maximum power output 88 kW (120 HP) at 5800 rpm and maximum torque 170 Nm at 4000 rpm was converted to flex-fuel operation by mounting fuel injectors with increased flow capacity (Racetronix 48INJL), stainless steel fuel rail and fuel lines to ensure methanol compatibility. The standard spark plugs were replaced by colder ones to avoid pre-ignition issues on methanol [17].

2.1.2. Single cylinder engine

The single cylinder engine used in this study is based on an Audi/NSU research DI diesel engine. The diesel injectors were replaced by spark plugs and the engine was converted for flex-fuel operation using similar adjustments as for the Volvo 1.8 l.

The setup is equipped with a cooled EGR line and a supercharging system using a volumetric claw compressor (Busch type MM1102 BP AQUA) with intercooler. The amount of EGR is controlled by a valve with an adjustable opening frequency and duty cycle. Additionally a backpressure valve can be used to force additional exhaust gases through the EGR line in situations where high EGR contents are desired.

2.1.3. 1.9 l four-cylinder turbocharged engine

A third engine used in this work was derived from a VW 1.9L AXR TDI automotive diesel engine. The original diesel injectors were replaced with spark plugs with a cold heat grade to avoid pre-ignition issues (NGK C9E). The original manifold was modified to accommodate port fuel injectors. Several injector types were tested and the current results were obtained using 6-hole Bosch injectors with an extended tip (0280157000). The original fuel supply system was replaced by a methanol compatible one. As reported by Brusstar et al. this type of engine has inlet ports which give a swirl ratio of about 2, reducing the tendency for knock [13]. Furthermore the combustion chamber was modified to eliminate potential pre-ignition sites.

A variable geometry turbocharger (Garrett VNT15) maintained the intake manifold pressure. EGR was directed from the low-pressure side of the turbine to the low-pressure side of the compressor. The amount of EGR was controlled by a throttle in the EGR pipe in combination with a backpressure valve in the exhaust. The EGR and fresh air were cooled after the compressor using a stock air-to-air intercooler.

2.2. Measurement equipment

All engines were equipped with a fully programmable engine control unit to control ignition timing, start of injection and injection duration.

Cylinder pressure measurements were done using piezo-electric pressure sensors and piezo-resistive sensors were placed in the intake and exhaust for pegging the cylinder pressure (see Table 3). The crank angle was recorded using crank angle encoders. For the four-cylinder engines, a spark plug pressure sensor was used in one of the cylinders, but for the single cylinder research engine fitting the cylinder pressure sensor posed a particular challenge.

Earlier attempts to measure the cylinder pressure through the glow plug passage had failed because of excessive mechanical vibrations and pressure oscillations in the passage. Therefore, a new passage was created so the sensor could be fitted level with the cylinder head. The position of this passage was determined using X-ray Computed Tomography (CT). CT is a non-destructive technique which enables visualization of the internal structure of materials in three dimensions (3D). The scan was performed at the modular micro-CT set-up at the Centre for X-ray Tomography of the Ghent University [18] and the obtained projections were reconstructed using the software package Octopus [19]. The interior of the cylinder head was scanned, so as not to damage any structural, lubrication or cooling related features (see Fig. 1).

A Bosch wide band sensor and digital air/fuel metre were used to give a direct reading of the air-to-fuel equivalence ratio λ. The fuel consumption measurements were done either gravimetrically or with a Coriolis type flow sensor (see Table 3). The mass air flow was measured with an automotive type MAF sensor and/or a

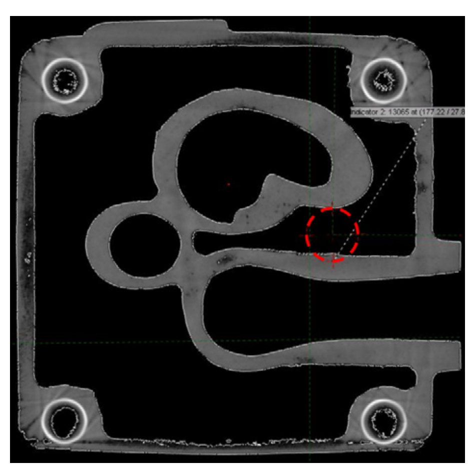


Fig. 1. Reconstructed cross-section of the micro-CT images, showing the internal structure of the Audi/NSU single cylinder engine's cylinder head (top view). The position of the pressure sensor between intake and exhaust port is indicated by the dashed circle.

research type air flow metre. The torque was measured with a load cell and temperatures were recorded at several locations using K-type thermocouples.

For the 1.8 l four-cylinder and the single cylinder engine, the exhaust gas components O_2, CO, CO_2 and NO_x were measured (O_2: Maihak Oxor-P S710, paramagnetic; CO, CO_2, NO, NO_2: Maihak Multor 610, non-dispersive infra-red). For the 1.9 l VW engine, concentrations of CO, CO_2, HC and NO_x in the exhaust gases were measured using a universal exhaust gas analyzer (Hermann HGA 400). This device gives fast readings but the accuracy is rather limited. It can only be used to discern trends.

2.3. Procedures

The results presented in this work were obtained during steady state operating conditions at various engine loads and speeds. For knock-limited operating conditions, BLD (border line detonation) spark timing was used, for all other conditions MBT (Minimum spark advance for best torque) timing was applied. The effect of pressure charging was not considered on the single cylinder engine.

All measurements were done with cooled EGR (to 25–30 °C), although the potential increase in thermal efficiency is reported to be higher using hot EGR [20]. This was done in order to protect the inlet of the compressor of the single cylinder engine, which is not designed for temperatures above 80 °C. Additionally, cooled EGR is more effective in lowering NO_x emissions. The EGR mass flow was calculated based on measured CO_2 concentrations [16].

Indicated quantities were averaged over 50 consecutive pressure cycles, in order to level out cyclic variation.

3. Results and discussion

3.1. Comparison between gasoline and methanol operation in flex-fuel engines

A well known advantage of using methanol in spark-ignition engines is the increase in maximum achievable engine power. As mentioned above, the high degree of charge cooling as the fuel is injected leads to increased volumetric efficiency. This, in combination with the elevated knock resistance, enables a power increase of more than 10% as the results obtained on the Audi engine demonstrate (see Fig. 2).

Apart from increased power, methanol also leads to superior brake thermal efficiency thanks to a number of interesting properties. To compare the distinct features of neat methanol and gasoline the measured cylinder pressure curves from the 1.8 four-cylinder engine are compared at a single operating point: 40 Nm and 1500 rpm. This would correspond to a low load, steady cruise in high gear. Table 4 shows the engine settings and resulting brake thermal efficiency (BTE) for methanol and gasoline. Fig. 3 shows the corresponding diagrams of cylinder pressure versus volume, in logarithmic coordinates.

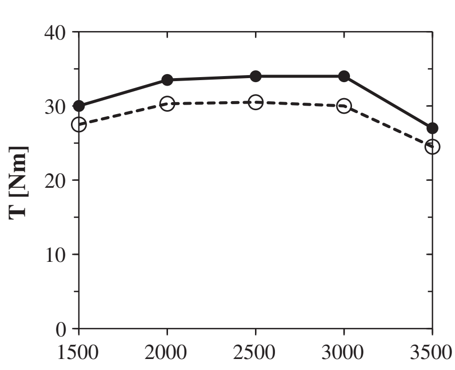


Fig. 2. Maximum torque from the Audi/NSU engine in atmospheric operation. Gasoline – open symbols, Methanol (M100) – closed symbols.

Table 4
Efficiency comparison between methanol and gasoline at 1500 rpm and 40 Nm.

	Methanol (M100)	Gasoline
λ	1	1
Throttle position (%)	8.5	14
Ignition timing (°ca BTDC)	17	25
Mass air flow (g/s)	9.4	10.3
Brake thermal efficiency (%)	26.1	24.0

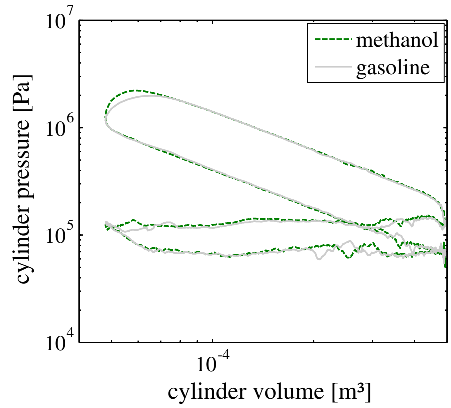


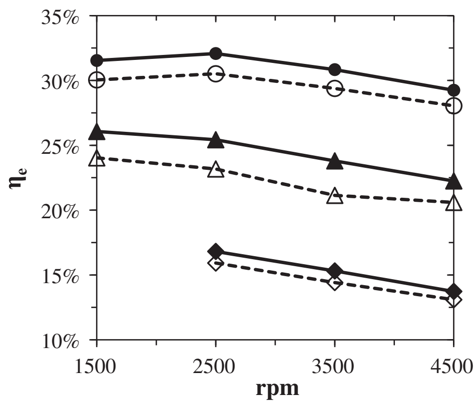
Fig. 3. Volvo engine – cylinder pressure versus cylinder volume at 1500 rpm and 40 Nm. Gasoline – full line, Methanol (M100) – dashed line.

Table 5
Measurement points for the 1.8 l four-cylinder Volvo and single cylinder Audi/NSU engine.

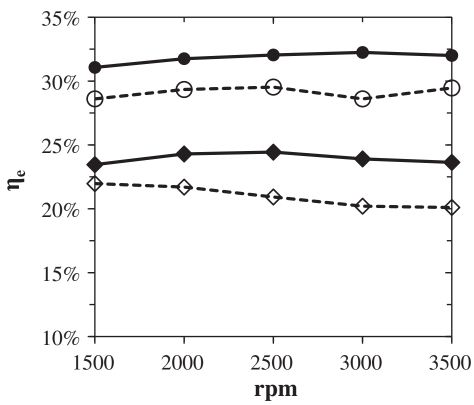
Volvo 1.8 l					
rpm	1500	2500	3500	4500	
Torque (Nm)	20	40	60	80	
BMEP (bar)	1.41	2.82	4.23	5.63	
Audi/NSU single cylinder					
rpm	1500	2000	2500	3000	3500
Torque (Nm)	10	15	20	25	
BMEP (bar)	3.09	4.63	6.17	7.71	

When comparing the pressure traces for methanol and gasoline (Fig. 3), it can be seen that operation on methanol takes advantage of its higher burning velocity. Additional proof for this faster burning can be found by comparing the optimal spark timing for both fuels (Table 4). The flow losses on methanol operation are slightly lower than for gasoline. On the one hand the airflow on methanol operation is lower due to the lower mixture energy per unit mass of air (see Table 1). On the other hand the throttle opening on methanol operation is smaller (Table 4). Although the volumetric (molar) mixture energy content is smaller than for gasoline, the large charge cooling upon methanol injection results in substantial increase in volumetric efficiency. Consequently the throttle opening to achieve a certain torque is smaller on methanol than on gasoline, which increases throttling losses.

Next, the efficiencies on methanol and gasoline operation on both the 1.8 l four-cylinder and the single cylinder engine are compared at different fixed torque settings and for a range of engine speeds (see Table 5). The influence of gas dynamics (flow losses)

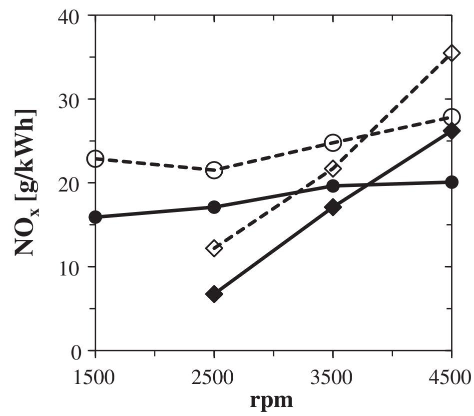


(a) 1.8l four-cylinder engine - 20 Nm (diamonds), 40 Nm (triangles), 80 Nm (circles)

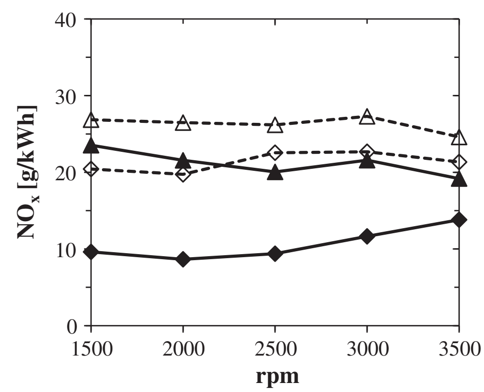


(b) Single cylinder engine - 10 Nm (diamonds), 25 Nm (circles)

Fig. 4. Brake thermal efficiency as a function of engine speed for different fixed brake torques. Gasoline – open symbols, Methanol (M100) – closed symbols.



(a) 1.8l four-cylinder engine - 20 Nm (diamonds), 80 Nm (circles)



(b) Single cylinder engine - 10 Nm (diamonds), 20 Nm (triangles)

Fig. 5. Engine-out NO_x (g/kWh) as a function of engine speed for different fixed brake torques. Gasoline – open symbols, Methanol (M100) – closed symbols.

is expected to rise with increasing rpm. At each point, optimal spark timing was used except for gasoline operation at 25 Nm on the single cylinder engine. There, the spark timing had to be slightly retarded to avoid knock.

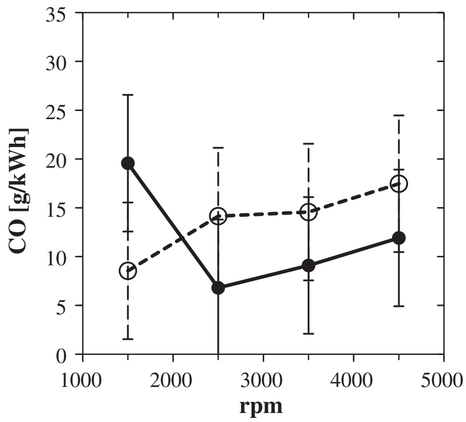
Fig. 4 shows the brake thermal efficiency as a function of engine speed and for different fixed torque outputs measured on both engines. A leaking throttle on the 1.8 l four-cylinder engine made it difficult to reach stable engine operation at 1500 rpm and 20 Nm. The error bars are not shown the figure, but were below 0.5% for the four-cylinder and below 1% for the single cylinder engine measurements respectively. It can be seen that the BTE on methanol is about 10% higher relative to gasoline. This is partly explained by the higher burning velocity of methanol, which can be more than twice that of gasoline. As mentioned before, another contributing factor is the reduced flow losses on methanol due to lower airflow. This advantage is partly lost however because of the smaller throttle opening. The in-cylinder cooling losses are also smaller on methanol compared to gasoline operation. Not only does the charge cooling due to methanol evaporation reduce the unburned mixture temperature, but the high heat capacity of burned methanol also reduces the flame and exhaust gas temperatures compared to gasoline. Other minor factors contributing to the BTE rise on methanol might be a slight increase in expansion work due to a higher ratio of C_p–C_v.

Fig. 5 compares the engine-out NO_x emissions on methanol and gasoline at different fixed loads. The estimated uncertainty is below 1 g/kWh for all measurement points. The values are consistently 5–10 g/kWh lower on methanol. This is explained by the cooler combustion temperatures on methanol compared to on gasoline. Additional evidence for these lower combustion temperatures can be found when comparing the mean temperature of the exhaust gases, which are 2–7% lower on methanol relative to gasoline (not shown here).

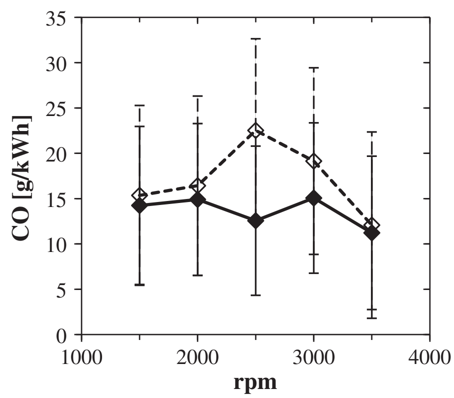
For the results obtained on the 1.8 l four-cylinder engine, the strong influence of engine speed on the amount of NO_x emissions at 20 Nm might be caused by elevated levels of internal EGR at low rpm. At this low load the vacuum in the intake due to throttling is quite considerable [21]. At 80 Nm the internal EGR levels are much lower, which explains the higher NO_x emissions for gasoline and methanol.

Fig. 6 compares the engine-out CO emissions for gasoline and methanol. Results and the associated error bars are displayed for one load only, so as not to overload the figures. The trends are similar for other considered load points. It can be seen that the emissions are slightly lower for methanol for most measurement points. According to some authors this is due to the oxygenated nature of methanol which might cause a more complete combustion [22]. However, the differences in CO emissions between the two fuels are not significant in our case. Also, slight deviations from stoichiometric operation might have much more influence on the CO emissions than the fuel type.

Fig. 7 compares the CO_2 emissions for gasoline and methanol. The results indicate that methanol operation reduces the specific CO_2 emissions by more than 10% compared to gasoline. This is

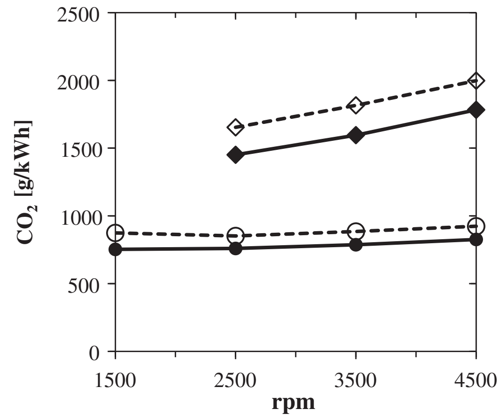


(a) 1.8l four-cylinder engine - 80 Nm (circles)

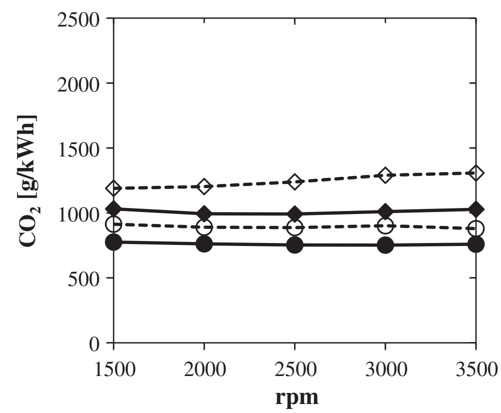


(b) Single cylinder engine - 10 Nm (diamonds)

Fig. 6. Engine-out CO (g/kWh) as a function of engine speed for different fixed brake torques. Gasoline – open symbols, Methanol (M100) – closed symbols.



(a) 1.8l four-cylinder engine - 20 Nm (diamonds), 80 Nm (circles)



(b) Single cylinder engine - 10 Nm (diamonds), 25 Nm (circles)

Fig. 7. Engine-out CO_2 (g/kWh) as a function of engine speed for different fixed brake torques. Gasoline – open symbols, Methanol (M100) – closed symbols.

partly due to the lower CO_2 formation per unit of energy when burning methanol (68.4 g/MJ compared to 74 g/MJ for gasoline). Another reason is the higher brake thermal efficiency when running on methanol.

Emissions of unburned hydrocarbons have not been included since flame ionization detectors (as used in our measurements) are reported to have a slow response time to oxygenated species [23]. Oxygenated species such as unburned methanol and formaldehyde are commonly found in the exhaust gases of methanol engines [10]. Using a flame ionization detector might thus lead to an underestimation of the total unburned hydrocarbons on methanol operation.

3.2. Alternative load control strategies on methanol

As mentioned above the elevated flame speed and wide flammability limits of methanol open some alternative options for load control. We have examined wide open throttle (WOT) lean burn operation and WOT EGR operation. These strategies respectively employ variable mixture richness or stoichiometric mixtures and variable amounts of external exhaust gas recirculation to control the load while the throttle position is fixed, preferably on wide open throttle.

The application of these strategies is interesting for a number of reasons. The reduction of throttling losses by working with the throttle wide open is beneficial for part load efficiencies [13,20,24,25]. Further efficiency improvements are obtained through the reduction of cooling losses and endothermic dissociation losses thanks to lower in-cylinder temperatures associated with the dilution of excess air or exhaust gases [20,24,25]. Additionally, lean operation can shift the combustion reaction closer to completion, releasing more energy per mole of fuel. Another interesting feature of lean operation is that the value of $\kappa = C_p/C_v$ increases, which leads to slightly higher efficiencies [12,26].

Compared to dilution with air, EGR dilution is more effective in reducing the temperatures since its heat capacity is higher than air. The temperature reductions can also help to avoid abnormal combustion phenomena such as pre-ignition or knock. This makes it possible to retain optimal spark timing over a wider range of operating conditions, thereby improving efficiency [13].

Lower peak in-cylinder temperatures are also interesting from an emissions point-of-view. The formation of nitrogen oxides (NO_x) is mainly triggered by temperatures above 1800 K, which means any reduction of in-cylinder temperatures will help to decrease the level of engine-out NO_x emissions [13,20,24]. However, it is important to note that in lean conditions the engine-out NO_x emissions should be low enough to offset the loss in three-way catalyst conversion efficiency.

For the WOT EGR strategy the amount of unburned hydrocarbons (UHC) generally increases due to the less complete combustion associated with adding excess exhaust gas. CO emissions remain unaffected, for these are mainly influenced by the mixture equivalence ratio [20,24]. For the WOT lean burn strategy the level of UHC and CO emissions usually decreases due to more complete combustion in lean conditions. However, as the lean limit is approached misfires and incomplete combustion produce an increase in these emissions [12,26].

Unfortunately, the application of excess air or external EGR also has some negative side-effects. The most important disadvantage is that the combustion becomes increasingly unstable as higher levels of dilution are applied. This elevated cyclic variation not only reduces the mean efficiency of the engine, but can even make it unfit for driving purposes [24]. A common measure for cycle-to-cycle variation is given by the Coefficient of Variation (CoV), which is defined as the standard deviation on the indicated mean effective pressure (imep) of the different engine cycles (σ_{imep}) divided by the mean imep (μ_{imep}).

$$\mathrm{CoV} = \sigma_{imep}/\mu_{imep} \tag{1}$$

In accordance with Heywood, the CoV was kept below a value of 10% in this work [27]. Threshold values of 5% or less are also commonly used [13,24].

Diluting the mixture will reduce the turbulent burning velocity inside the cylinder, rendering the combustion less isochoric. This will decrease indicated efficiency. An additional effect of increased combustion duration is that the hot combustion gases are in contact with the cylinder walls for a longer time, which partly offsets the reduction in heat losses through dilution.

Increasing the level of in-cylinder turbulence can help to increase the flame speed to a certain degree, but will also negatively affect the heat losses. Others ways of improving dilution tolerance include turbocharging and increasing the compression ratio of the engine, as will be demonstrated in the next chapter of this paper [15].

A fuel with wide flammability limits, elevated EGR tolerance and high laminar burning velocities such as methanol (see Table 1) will be less prone to the negative side-effects of dilution [24]. Our measurements on the single cylinder engine indicated that EGR levels up to 30% could be applied before the CoV surpassed the threshold value of 10%, whereas for gasoline operation this happened at 10% EGR. The lean burn limits were $\lambda = 1.5$ and $\lambda = 1.25$ on methanol and gasoline respectively [16].

Fig. 8 illustrates some of the key differences between throttled stoichiometric, WOT lean burn and WOT EGR operation on methanol in a $\log p$–$\log V$ diagram. The associated engine operating conditions are summarized in Table 6. The pumping loop in the WOT lean burn and WOT EGR operation is considerably smaller due to the absence of throttling losses. However, the excess air and EGR flow also brings about some minor additional flow losses in the intake and EGR pipes. Also visible on the diagram is the adverse influence of air and EGR dilution through less isochoric combustion. This effect is slightly less pronounced for lean operation.

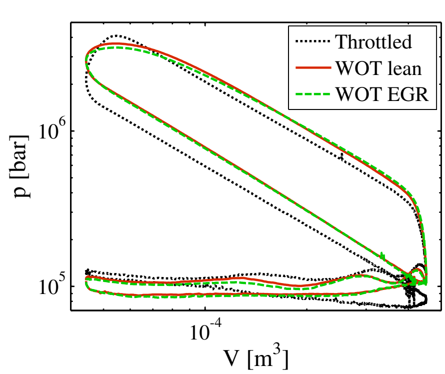


Fig. 8. Cylinder pressure versus cylinder volume at 1500 rpm and 25 Nm. Throttled – dotted line, WOT EGR – dashed line, WOT lean – full line.

Table 6
Efficiency comparison between throttled, WOT EGR and WOT lean operation at 1500 rpm and 25 Nm.

	Throttled	WOT EGR	WOT lean
λ	1	1	1.14
Throttle position (TP) (%)	30.5	100	100
Ignition timing (°ca BTDC)	16	23	19
EGR (mass%)	0	16.4	0
Indicated thermal eff. (%)	32.8	35.1	35.2

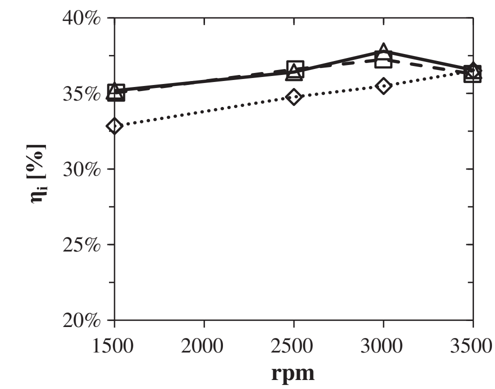


Fig. 9. Indicated thermal efficiency as a function of engine speed for a fixed brake torque of 25 Nm. Throttled – diamonds, WOT EGR – squares, WOT lean – triangles.

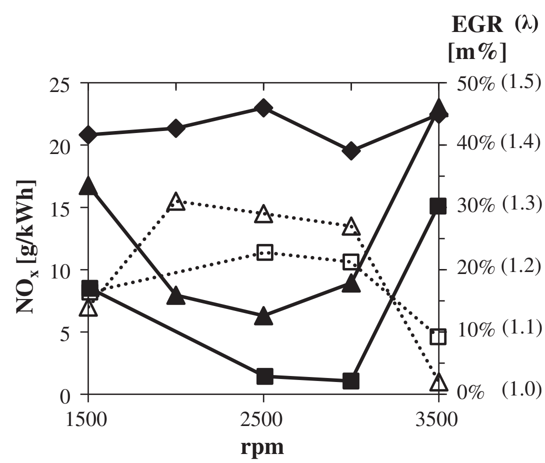


Fig. 10. Engine-out NO_x (closed symbols) and associated EGR%/λ (open symbols) as a function of engine speed for a fixed brake torques of 25 Nm. Throttled – diamonds, WOT EGR-squares, WOT lean-triangles.

Fig. 9 illustrates that although the lean burn strategy theoretically has an edge on the EGR strategy in terms of efficiency, in practice both strategies produce almost identical indicated thermal efficiencies when running on methanol. The relative improvements compared to throttled operation are in the order of 5%. At this load (20 Nm) there were also considerable cyclic variations when running on methanol (CoV $\approx$ 6%), which leads to a rather high standard deviation on the indicated thermal efficiency (2% absolute). At 25 Nm, the CoV was below 1% on methanol and the same trends were observed.

From Fig. 10 it can be seen that larger decreases in engine-out NO_x emissions are obtained when diluting the mixture with EGR rather than excess air. This is mainly due to the higher specific heat of the recirculated combustion gases. There is a strong correspondence between the amount of excess air (proportional to λ) or EGR (EGR%) and the decrease of NO_x.

As opposed to EGR dilution, using excess air also implies that the engine-out NO_x emissions should be low enough to compensate for the loss in three-way catalyst conversion efficiency in lean conditions. A common threshold value is 100 ppm [27]. The load points in Fig. 10 exceed this threshold value, so they cannot be part of a realistic control strategy. Pannone and Johnson [12] suggest that for methanol further leaning of the mixture and compensating the power loss by turbocharging (which also increases lean limits) might drop the engine-out NO_x low enough to eliminate the need for NO_x aftertreatment.

The results for engine-out CO emissions did not indicate a significant difference between the load control strategies. This is not in agreement with the trends observed by Pannone and Johnson [12]: leaning the mixture decreases the CO emissions. For WOT EGR operation no change in CO emissions was expected, as these are mainly driven by air-to-fuel equivalence ratio [13,17].

Our measurements demonstrate that both the WOT EGR and lean burn load strategy have the potential to increase engine efficiency compared to throttled stoichiometric operation. Theoretically the lean burn strategy can have a minor edge on the WOT EGR strategy in terms of brake thermal efficiency, but this was not reflected in our measurement results.

With regard to NO_x emissions, the current results indicate that the WOT EGR strategy leads to lower specific engine-out values thanks to the higher thermal capacity of exhaust gases compared to excess air. More importantly, the WOT EGR strategy employs stoichiometric combustion, which means conventional three-way catalyst aftertreatment can be used to lower the tailpipe NO_x emissions to appropriate values. This is not the case for the lean burn strategy, which limits its practical use.

In conclusion, the EGR WOT strategy seems more promising as it allows to significantly increase the part load efficiency without sacrificing in terms of tailpipe emissions. Despite methanol's high burning velocity and dilution tolerance, the load range in which this load strategy could be applied was quite limited for the Audi engine. At 6 bar BMEP, CoV values were near the threshold limit of 10%. For this reason the WOT EGR strategy was further investigated on another engine, whose properties are more appropriate for EGR dilution.

3.3. Application of the WOT EGR strategy to a turbocharged, high compression ratio engine

The VW TDI engine has an elevated compression ratio of 19.5:1, is turbocharged and has high levels of in-cylinder turbulence thanks to swirl created by the intake. These properties boost the burning rate inside the engine, which is desirable if one wants to use elevated levels of EGR to control load. The combination of high compression ratios and external EGR is especially interesting since EGR reduces some of the disadvantages related to high compression ratios:

- High compression ratios increase the chance of abnormal combustion phenomena such as knock and pre-ignition.
- The increased temperatures associated with high CR lead to higher cooling and dissociation losses.
- These increased in-cylinder temperatures also lead to more NO_x formation.

Dilution with EGR reduces the in-cylinder temperatures, thereby lessening the chance of knock and allowing optimal spark timing to be retained. Also cooling and dissociation losses and the formation of NO_x are reduced in this way.

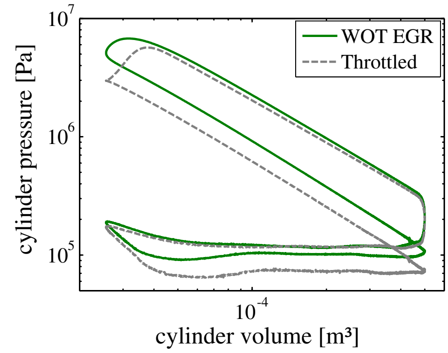
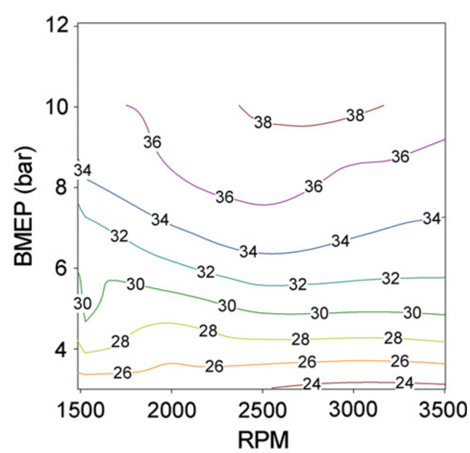
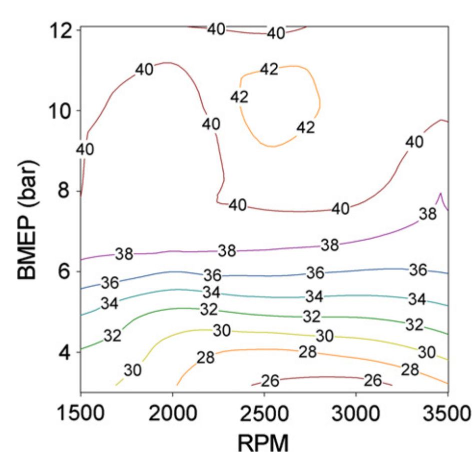


Fig. 11. Cylinder pressure versus cylinder volume at 2000 rpm and 100 Nm. Throttled – dotted line, WOT EGR – full line.



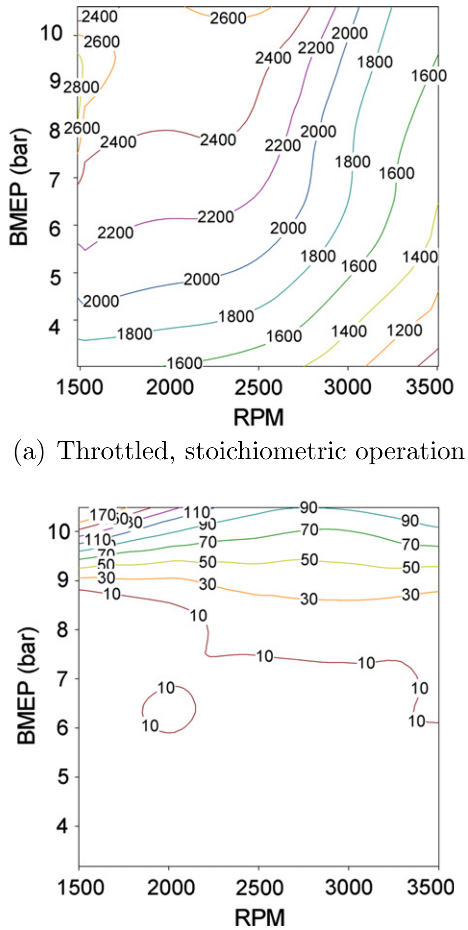
(a) Throttled, stoichiometric operation



(b) WOT EGR operation

Fig. 12. Brake thermal efficiency (%) as a function of engine speed and brake mean effective pressure (bar).

Two operation strategies on methanol were compared on the engine: normally aspirated throttled operation and turbocharged WOT operation with EGR. In both cases the mixture was combusted stoichiometrically. Measurements were done at loads between 50 Nm and 175 Nm (corresponding to 3.31 and 11.60 bar BMEP) and at speeds between 1500 and 3500 rpm. In throttled stoichiometric operation the maximum engine torque was knock limited to about 160 Nm. In much of the high load



(a) Throttled, stoichiometric operation

(b) WOT EGR operation

Fig. 13. Engine-out NO_x (ppm) as a function of engine speed and brake mean effective pressure (bar).

points the spark timing had to be retarded using this operating strategy. For the WOT EGR strategy this was not the case thanks to the cooling effect of EGR.

Thanks to the high burning rates in this engine, throttleless operation was possible down to 3.3 bar BMEP without unacceptable cycle-to-cycle variations (CoV < 10%). This corresponds to an EGR tolerance of nearly 50% EGR by mass [13,16].

Fig. 11 compares the pressure traces at 100 Nm and 2000 rpm between throttled stoichiometric and WOT EGR operation. The same features can be seen as in Fig. 8. In WOT EGR operation the flow losses are considerably reduced by keeping the throttle wide open. The difference between both load strategies is even larger than in Fig. 8 as this is a four-cylinder engine with a much higher mass airflow. The burning rate is reduced due to the EGR dilution, but thanks to the high CR this effect is weaker than in Fig. 8. It is interesting to note that in the case of Fig. 8 only 14% EGR was added, whereas the EGR content in Fig. 11 is more than 31% by mass.

Fig. 12 compares the brake thermal efficiency obtained using the throttled stoichiometric strategy (a) to that obtained using the turbocharged WOT EGR strategy (b). Both peak brake thermal efficiency and part load efficiencies can be seen to be considerably higher when using the WOT EGR strategy.

In both cases the peak brake thermal efficiencies are very high thanks to the elevated compression ratio of the engine. This compression ratio of 19.5:1 is not necessarily the optimal value, which is expected to be around 15:1 [27]. Whereas the spark timing is knock limited in the throttled stoichiometric case, MBT timing can be retained when using EGR. Additionally the reduced peak cylinder temperatures lead to lower cooling and dissociation losses. This explains the difference in peak brake thermal efficiency between the two strategies. Brusstar et al. reported the same value for peak brake thermal efficiency (42%) when employing the WOT EGR strategy on a similar engine [13].

The absence of throttling losses provides an additional efficiency benefit for the EGR WOT strategy at part loads. In these points the relative difference between both operating strategies can mount to 20% and higher. The gains are most pronounced at low engine speeds as the relative importance of throttling losses is largest in these conditions.

Due to the limited accuracy of the employed exhaust gas analyzer, the results for engine-out NO_x emissions shown in Fig. 13 are only indicative. They do suggest that the high levels of exhaust gas dilution applied in this engine enable vast reductions in engine-out NO_x emissions. More research will be executed to be able to estimate the improvements over an entire driving cycle.

4. Conclusions

This work experimentally assessed the potential efficiency benefits and emission reductions by employing methanol in flex-fuel and dedicated engines.

In the first section, gasoline and methanol operation were compared on two flex-fuel engines. On methanol, relative efficiency benefits of about 10% were obtained over the entire load range thanks to more isochoric combustion, less flow, cooling and dissociation losses. Lower combustion temperatures allowed to reduce the engine-out NO_x levels by 5–10 g/kWh compared to gasoline. No significant difference in CO emissions was observed, but the CO_2 values dropped by more than 10% partly thanks to the increased efficiency.

The second section of the paper focused on alternative load control strategies for methanol engines. Throttled stoichiometric operation was compared to WOT lean burn operation and WOT stoichiometric operation using EGR to control load. Both alternative strategies led to efficiency improvements up to 5% relative to throttled operation thanks to reduced throttling, cooling and dissociation losses. The decrease in combustion temperatures through dilution also enables a significant diminution in engine-out NO_x. This is most pronounced for EGR dilution, since the heat capacity of exhaust gases is higher than that of air. For lean burn operation the reduction is not sufficient to justify the loss in three-way catalyst efficiency associated with lean combustion. In this respect, the WOT EGR strategy seems of more practical use to methanol engines.

In the last section the WOT EGR strategy was applied on a turbocharged engine with an elevated compression ratio and high levels of in-cylinder turbulence. These features increased the burning rates enough to allow throttleless operation down to 3 bar BMEP. Thanks to the compression ratio of 19.5:1, peak brake thermal efficiencies of 42% were obtained on methanol. Gasoline operation was not possible due to heavy knock. At part loads, the absence of throttling losses and benefits associated with lower in-cylinder temperatures enabled relative efficiency improvements up to 20% compared to throttled operation. The high levels of EGR dilution at these loads (up to 50%) also allowed to reduce the engine-out NO_x emissions to negligible levels. These results demonstrate that methanol can be used in dedicated engines with diesel-like efficiencies and emission levels comparable to or lower than gasoline.

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