

High Efficiency and Low Emissions from a Port-Injected Engine with Neat Alcohol Fuels

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ABSTRACT

Ongoing work with methanol- and ethanol-fueled engines at the EPA's National Vehicle and Fuel Emissions Laboratory has demonstrated improved brake thermal efficiencies over the baseline diesel engine and low steady state NO_x, HC and CO, along with inherently low PM emissions. In addition, the engine is expected to have significant system cost advantages compared with a similar diesel, mainly by virtue of its low-pressure port fuel injection (PFI) system. While recognizing the considerable challenge associated with cold start, the alcohol-fueled engine nonetheless offers the advantages of being a more efficient, cleaner alternative to gasoline and diesel engines.

The unique EPA engine used for this work is a turbocharged, PFI spark-ignited 1.9L, 4-cylinder engine with 19.5:1 compression ratio. The engine operates unthrottled using stoichiometric fueling from full power to near idle conditions, using exhaust gas recirculation (EGR) and intake manifold pressure to modulate engine load. As a result, the engine, operating on methanol fuel, demonstrates better than 40% brake thermal efficiency from 6.5 to 15 bar BMEP at speeds ranging from 1200 to 3500 rpm, while achieving low steady state emissions using conventional aftertreatment strategies. Similar emissions levels were realized with ethanol fuel, but with slightly higher BSFC due to reduced spark authority at this compression ratio. These characteristics make the engine attractive for hybrid vehicle applications, for which it was initially developed, yet the significant expansion of the high-efficiency islands suggest that it may have broader appeal to conventional powertrain systems. With further refinement, this clean, more efficient and less expensive alternative to today's petroleum-based IC engines should be considered as a bridging technology to the possible future of hydrogen as a transportation fuel.

INTRODUCTION

Alternative fuels, especially alcohol fuels, offer potential to mitigate national security and economic concerns over fuel supplies as well as environmental concerns over tailpipe emissions and resource sustainability. As a result, there has been continuing interest in alternative fuels, heightened recently over proposed legislation that would mandate increases in the use of renewable transportation fuels. Over the last thirty years of automotive research, a variety of alcohol fuels—primarily methanol, ethanol and blends with hydrocarbon fuels—have demonstrated improved emissions of oxides of nitrogen (NO_x) and particulate matter (PM) as well as moderately improved brake thermal efficiency [1-4]. Despite this, infrastructure barriers as well as technical challenges, notably cold starting, have limited the widespread use of neat alcohol-fueled vehicles.

The benefits and challenges of neat alcohol fuels in PFI applications have been demonstrated in numerous earlier works. Benefits such as higher efficiency and specific power and lower emissions may be realized with alcohols: their high octane number gives the ability to operate at higher compression ratio without preignition [5]; their greater latent heat of vaporization gives a higher charge density [1-3, 6]; and their higher laminar flame speed allows them to be run with leaner, or more dilute, air/fuel mixtures [7]. In addition, alcohols generally give lower fuel heat release rates, resulting in lower NO_x emissions and reduced combustion noise [2]. The engine described in the present work uses these inherent advantages of alcohol fuels as the basis for its design and control, thereby enabling attainment of efficiency levels exceeding that of the diesel, with low emissions.

One of the main challenges with neat alcohol fuels is cold start emissions, especially in PFI engines [8]. In such applications, the low vapor pressure and low

cetane number must be overcome with higher-energy ignition systems or higher compression ratio [9]. Further, the increased wetting of the intake manifold, cylinder walls and spark plugs must be addressed in the design of the combustion chamber and in the control of transient fueling during startup [8, 10]. Because of these issues, earlier works with PFI SI methanol engines commonly report starting problems below ambient temperatures of about 10°C [8, 11]. With extended periods of cold cranking (i.e., 60 seconds or more), successful starting has been achieved at ambient temperatures as low as -6.5°C [12]. However, these studies were generally performed with lower compression ratio engines, derived from their gasoline counterparts, which are therefore not necessarily optimized for use with neat alcohol fuels. The ongoing work at EPA with high-compression ratio single cylinder PFI SI engines [13], for example, demonstrates the ability to fire on neat methanol during relatively brief cranking at higher speeds, at temperatures as low as 0°C.

Earlier work at EPA with alcohol-fueled multi-cylinder engines examined both PFI and DI configurations, generally demonstrating improvements in fuel economy and power, as well as promising cold start emissions. Initial work with a methanol-fueled PFI SI engine [14, 15] yielded fuel economy and emissions that were similar to, but not significantly better than, the baseline gasoline engine. Cold starting was not addressed in that early program, but was instead examined in a follow-on project with a turbocharged, DI, glow-plug-ignited, stratified charge engine [16], based on earlier works showing good cold start performance down as low as -29°C [17]. This project was largely successful, demonstrating good startability and driveability down to -29°C, and producing low FTP emissions of NO_x (0.3 g/mi), HC (<0.01 g/mi), CO (0.2 g/mi), PM (0.02 g/mi) and aldehydes (0.002 g/mi) while running lean with a single-stage oxidation catalyst. The measured fuel economy was between 7%-22% better than the baseline gasoline engine, but still slightly lower than the turbocharged diesel.

The ongoing PFI alcohol work presented here builds on our earlier experience, and demonstrates better steady state efficiency than the baseline diesel and low emissions with conventional aftertreatment systems, at a significantly lower cost than the diesel. The engine described below runs unthrottled over most of its load range, much like the diesel, but operates with high EGR dilution ratios at stoichiometric fueling, rather than lean and stratified. This strategy takes advantage of the favorable dilute flammability limits of alcohol fuels to operate with lower pumping losses, and uses the high levels of EGR to control knock at high compression ratio. As a result, the engine demonstrates its potential as an efficient, lower cost, renewable fuels alternative to the diesel.

EXPERIMENTAL SETUP

The research described below is being conducted under EPA's Clean Automotive Technology Program, in order to demonstrate feasibility of cleaner, more efficient technologies. The primary focus of the work is on methanol fuel, since it represents the limiting case of oxygenated fuels, at 50% oxygen by mass. Also, its physical properties lend some performance advantages over other alcohols, discussed below. For comparison, however, brake thermal efficiency data with ethanol fuel is also given below, demonstrating similar benefits.

ENGINE AND TEST DESCRIPTION

The engine designed for this work is derived from the 1.9L Volkswagen TDI automotive diesel engine, modified suitably to accommodate port fuel injectors and spark plugs. The stock inlet ports give a swirl ratio of about 2.0, a factor that has been demonstrated to reduce the tendency for knock [18]. Knock was further reduced by modifying the stock combustion chamber to eliminate potential preignition sites. A range of compression ratios from 17:1 to 22:1 were tested in this engine with methanol fuel, although the results reported below were conducted at a nominal compression ratio of 19.5:1.

Intake manifold pressure was maintained with a variable geometry turbocharger, which, in turn, also varied the exhaust backpressure on the engine. EGR was metered from the low-pressure side of the turbine to the low-pressure side of the compressor, using a variable backpressure device in the exhaust. The EGR temperature was reduced with a stock Volkswagen water-to-air cooler before the compressor, and the EGR and fresh air were cooled after the compressor with a stock air-to-air intercooler. Together, these compact heat exchangers were able to maintain intake manifold temperatures in the vicinity of 30°C.

At least four different types of port fuel injectors were evaluated for measured engine brake thermal efficiency as well as spray characteristics with methanol, verified with high-speed planar laser imaging. The best-atomizing injectors among the group were racing-style, 36 lb/hr, 12-hole port fuel injectors manufactured by Holley, operating at 4 bar rail pressure. For best startup and transient performance, the injector tip was targeted at the back of the intake valve, from a distance of approximately 80 mm.

The ignition system consisted of a production Toyota coil with a Champion dual electrode, recessed gap spark plug. High load operation, with a combination of high cylinder pressures and smaller spark advance, placed great demand on both the plugs and coils. Together with higher corrosive properties of methanol, spark plug durability was somewhat of an issue in this testing, as had been witnessed in earlier works [9].

Table 1: EPA alcohol engine specifications

Engine Type	4 cyl., 4-stroke
Combustion Type	PFI, SI
Displacement	1.9L
Valves per cylinder	2
Bore	79.5 mm
Stroke	95.6 mm
Compression Ratio	19.5:1
IVO	-344° ATDC*
IVC	-155° ATDC*
EVO	152° ATDC*
EVC	341° ATDC*
Bowl Volume	18 cc
Clearance volume	26.4cc
Swirl Ratio	2.0
Injectors	Holley, 36 lb/hr, 12-hole nozzle
Rail Pressure	4 bar
Spark Plugs	Champion recessed gap, dual electrode
Turbocharger type	Variable geometry
Exhaust	Ford FFV 2-stage, three-way catalyst

*-relative to fired TDC

The engine was run with anhydrous chemical-grade methanol and ethanol fuels, and batch chemical analyses were performed to verify the heating value and density. NO_x emissions were measured with a chemiluminescent NO_x analyzer, while CO emissions were measured with a non-dispersive infrared analyzer. Unburned hydrocarbon (HC) emissions were measured with a heated flame ionization detector calibrated with propane, but corrected separately for response to methanol and ethanol. A two-stage, three-way Ford FFV catalyst was used for exhaust aftertreatment, and was aged approximately 10 hours at high, variable load prior to testing.

ENGINE CONTROLS DESCRIPTION

The engine controller was a Rapid Prototype Engine Control System (RPECS) provided under contract from Southwest Research Institute. The EPA operating strategy was based on three fundamental principles: (1) High compression ratio, in order to give an expanded dilute operating range; (2) Turbocharging with high levels of EGR, for primary load control and low NO_x emissions; (3) Stoichiometric fueling (based on oxygen to fuel), to permit operation with a three-way catalyst. The performance and/or emissions benefits of individual components of this strategy have been demonstrated in earlier works, discussed below. Taken together, however, the present strategy is unique, and presents a path for attaining high levels of efficiency and low emissions in a practical, feasible system.

Methanol and ethanol have relatively high octane numbers compared with gasoline; published RON values for methanol and ethanol are between 105-109, compared with about 91-99 for gasoline [19, 20]. As a result, they may be run at a much higher compression ratio, thereby yielding higher engine thermal efficiency. Earlier works with single-cylinder SI methanol engines [5], for example, showed 16% improvement in brake efficiency when raising the compression ratio from 8.0 to 18.0, while still achieving minimum best torque (MBT) spark timing with only light knock. A compression ratio of 19.5:1 was chosen for this work based on earlier experience with a wider range of compression ratios, which showed this to be the best compromise between full spark authority without knock at high load and dilute combustion range at light load. The full spark authority at high load is enabled partly by the relatively high levels of EGR, which has been shown in earlier works to suppress knock at higher compression ratio [21]. Light load stability, meanwhile, is improved by the high compression ratio, which raises the temperature of compression and enhances the already comparatively high flame propagation velocities of the alcohol fuels. As a result, earlier works [21, 22] have demonstrated the ability to operate satisfactorily with as much of 33%-40% EGR with methanol, even with a relatively low compression ratio of 8-8.5. Using a higher compression ratio, the present work was able to achieve nearly 50% EGR without unacceptable cycle-to-cycle combustion variability, using a production spark ignition system.

The main objective of the engine load control strategy was to exploit the physical properties of the alcohol fuels in order to run unthrottled, and therefore more efficiently, over a relatively wide range of loads. Methanol-fueled engines using high levels of EGR to modulate load [21-23] have demonstrated efficiency gains of greater than 10% over throttled engines, while giving considerably lower NO_x emissions. Combining variable EGR rates with variable intake manifold pressure allows for a wider range of load control. This strategy has also been shown as an effective means of achieving NO_x levels below 1.0 g/kW-hr and peak efficiency around 42% in DI, lean stratified-charge methanol engines [23] and similar improvements in PFI lean burn methanol engines [24]. In the present engine, EGR and boost levels are maintained to achieve the best NO_x and efficiency, and still enabling MBT (or near MBT) spark timing at high loads. Manifold absolute pressure (MAP) was varied between 1.0-1.5 bar, while the maximum dilution level was limited to about 50% EGR. Throttling, meanwhile, was used only to achieve near-idle loads.

The engine is controlled to stoichiometric fueling, enabling use of a three-way catalyst for attainment of emissions at the levels required to achieve Federal Tier II LDV standards. Earlier experience operating lean with an oxidation catalyst [16] showed the ability to achieve Tier II-level emissions on a methanol vehicle for all but NO_x, pointing to the need for a three-way catalyst.

Operating at stoichiometric has the added benefit of enabling a higher specific power than a similar lean, stratified engine.

This strategy was successfully employed to achieve the steady state efficiency and emissions results shown below.

RESULTS AND DISCUSSION

Given below are efficiency and emissions test results for the engine operating with methanol and ethanol. Following this, a brief overview of preliminary cold start testing with methanol is presented.

BRAKE THERMAL EFFICIENCY (BTE)

The measured BTE of the engine operating with methanol fuel is given below in Figure 1, which may be compared with the BTE of the baseline diesel engine, given in Figure 2.

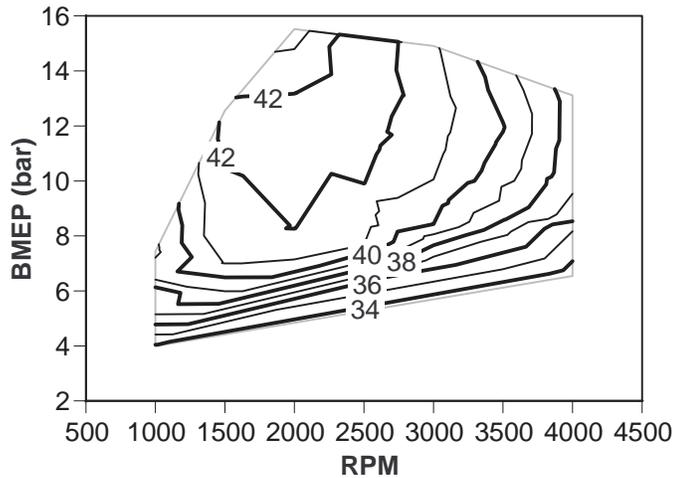


Figure 1. Methanol: BTE (%) as a function of BMEP, RPM.

The methanol engine exhibits peak efficiency of nearly 43%, and maintains over 40% efficiency over a much wider range of speeds and loads as compared to the diesel engine shown in Figure 2. This region of high efficiency, at levels normally associated with the diesel, extends from 6.5 to more than 15 bar BMEP, from 1200 to 3500 rpm. Despite high levels of EGR dilution at light load, combustion variability did not dramatically affect BTE at the BMEP levels shown. In addition, the engine was nearly able to achieve MBT without heavy knock at high loads, due to relatively high dilution with cooled EGR and higher manifold pressure (i.e., higher charge air mass). As a result, the COV of IMEP for the engine operating normally with methanol was less than 3% over the entire range of speeds and loads.

Unlike Figure 2, Figure 1 does not show measured BTE for low BMEP, since the focus of the present work was to explore the boundaries of the control strategy outlined earlier in this work. Extending the efficiency map to lower BMEPs with the current engine requires a throttling

device, though one that is less restrictive than that for conventional PFI gasoline engines.

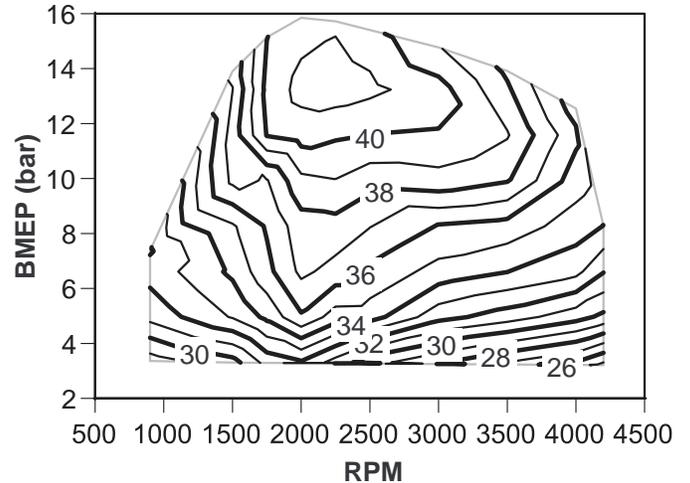


Figure 2. Baseline stock 1.9L VW TDI Diesel: BTE (%) as a function of BMEP, RPM.

The baseline diesel efficiencies given in Figure 2 were obtained at EPA using a Volkswagen stock TDI engine control unit and stock hardware. The figure shows slightly lower peak efficiency than the PFI methanol engine, and a more rapid drop-off in efficiency with decreasing load. The two major factors that possibly account for this difference are: 1) the parasitic losses of the high-pressure diesel fuel system, and 2) the considerable differences existing in the combustion and heat transfer processes, illustrated clearly by the cylinder pressure versus crank angle comparison given in Figure 3. The figure shows typical pressure traces for the engine operating with diesel and methanol fuel, at 11.5 bar BMEP, 2000 rpm, 1.5 bar intake manifold pressure (absolute) and 19.5:1 compression ratio.

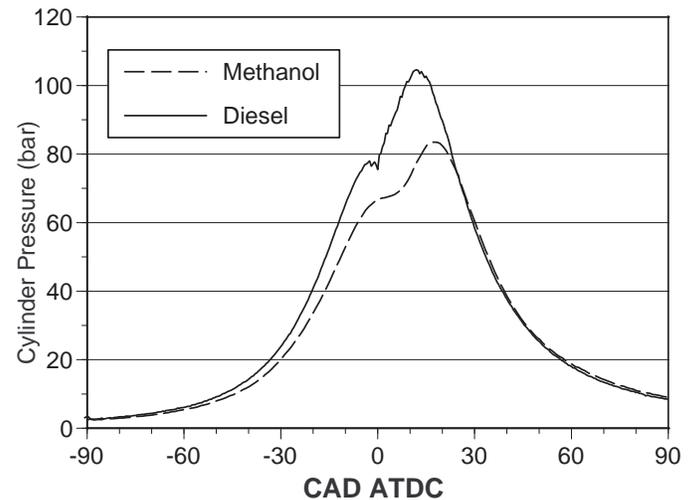


Figure 3. Comparison of cylinder pressure versus crank angle for diesel and methanol engines; 11.5 bar BMEP, 2000 rpm, 1.5 bar intake manifold pressure, 19.5:1 compression ratio.

The figure shows that the compression work with methanol is reduced considerably, due to the intense charge cooling resulting from methanol vaporization. Also, the methanol engine exhibits a slower rate of combustion heat release, leading to comparatively lower heat losses.

The measured BTE with ethanol fuel is shown below in Figure 4. Both the peak efficiency and the load and speed range with higher efficiency are comparable to that of the diesel in Figure 2. However, the engine was not able to achieve levels of BTE as high as methanol. This was mainly due to knock sensitivity at high load and high speed with ethanol, which prevented the engine from achieving MBT.

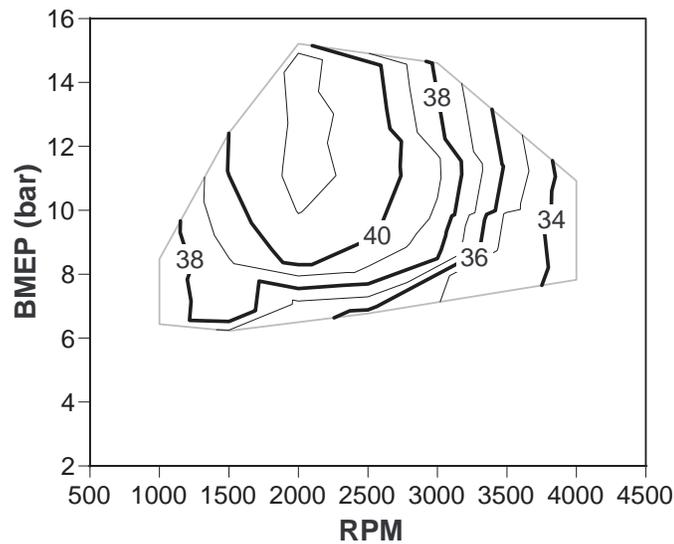


Figure 4. Ethanol: BTE (%) as a function of BMEP, RPM.

Moreover, the engine experienced greater levels of combustion variability at light load and at higher speeds, as witnessed by the high COV of IMEP shown in Figure 5, which, in turn, reduced the BTE under these conditions. Some of the BTE differential demonstrated here, however, may be recovered by optimizing the compression ratio and calibration for ethanol fuel.

BRAKE-SPECIFIC EMISSIONS FOR METHANOL

The figures below show NO_x and HC emissions for the engine operating with methanol. Similar results are expected for ethanol [25], but are not included.

Brake-specific NO_x emissions as a function of BMEP and RPM are shown in Figure 6. The high EGR dilution, combined with the slower heat release of methanol yields low levels of NO_x, at 0.1-0.2 g/kW-hr over much of the operating map. The NO_x emissions increase at lower speed, partly due to the inability of the turbocharger to maintain the intake manifold pressure at a level sufficient to permit higher rates of EGR.

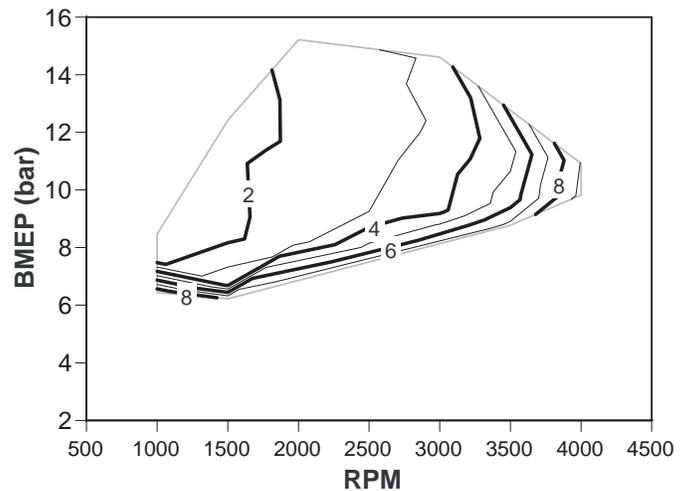


Figure 5. Ethanol: COV of IMEP (%) as a function of BMEP, RPM.

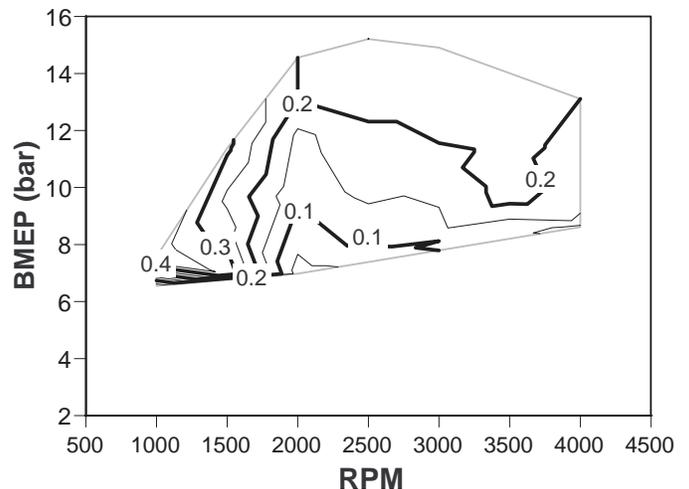


Figure 6. Brake-specific NO_x emissions (g/kW-hr) as a function of speed and load for methanol.

Figure 7 below shows brake-specific HC emissions as a function of speed and load. HC emissions are controlled to less than 0.2 g/kW-hr over most of the map, indicating the effectiveness of the aftertreatment system.

Brake specific CO measurements are not specifically shown in this work, since they were consistently very low, at less than 0.2 g/kW-hr over the entire map. PM and aldehyde emissions were not measured, though earlier work at EPA with DI methanol engines [16] demonstrated the ability to control these to very low levels with a conventional oxidation catalyst.

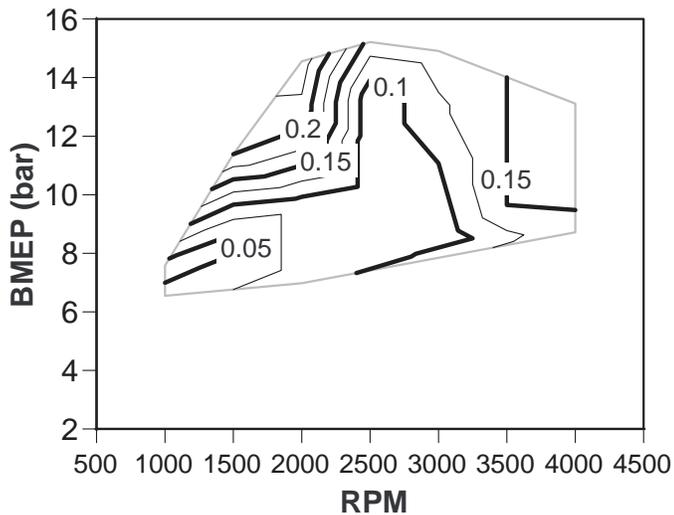


Figure 7. Brake-specific HC emissions (g/kW-hr) as a function of load for methanol.

COLD STARTING IN A SINGLE-CYLINDER ENGINE WITH METHANOL

The ongoing research at EPA includes work with single-cylinder engines that simulate closely the characteristics of the multi-cylinder engine. The single-cylinder results presented below were for a PFI SI configuration with 19.5:1 compression ratio and identical cam timings, displacement, bore/stroke ratio, and intake manifold geometry as the multi-cylinder engine described earlier in this work. It was run naturally aspirated and lean, to simulate early stages of the open-loop startup strategy used in the multi-cylinder engine.

Cold starting with the single cylinder was examined at ambient temperatures from 20°C down to 0°C [13]. The initial fueling and ignition timing sequences were varied to determine optimal combinations to ignite the charge and sustain combustion during the first ten firing cycles. The engine was ramped quickly up to speeds ranging between 1000 rpm to 2000 rpm, simulating conditions commonly seen during startup on the EPA hydraulic hybrid chassis. This higher cranking speed results in a higher compression temperature, and therefore improved low-temperature ignition [26]. Fueling with neat methanol was initiated such that the end of the injection event occurred just prior to intake valve closure. Startability, quantified by the measured IMEP during the first ten firing cycles, was very good at 20°C. At 0°C, the measured IMEP and in-cylinder wall temperatures indicated that significant quenching had occurred during the first few firing cycles, yet the engine was able to sustain combustion and achieve load. A more detailed exposition of this topic is planned for a later work.

In summary, the results above with methanol- and ethanol-fueled engines exhibit equal or greater efficiency than the comparable diesel engine, and low emissions of NO_x, CO and HC. Moreover, preliminary work with cold starting in a single cylinder engine exhibits good combustion down at 0°C. These studies are part of the

Clean Automotive Technology Program at EPA to demonstrate feasibility of clean technologies, and to develop attractive alternatives to conventional-fueled engines.

CONCLUSION

The present work describes a PFI, SI, turbocharged, high compression ratio engine operating with relatively high EGR dilution rates, operating on neat alcohol fuels. From the steady state results presented above, it is concluded that:

1. The present engine, optimized for alcohol fuels, exceeds the performance of current conventional-fueled engines, and has potential as a lower-cost alternative to the diesel.
2. Brake thermal efficiency levels better than a comparable turbocharged diesel are demonstrated. The engine operating with methanol fuel showed peak BTE of nearly 43%, and a broader high-efficiency operating range than the baseline diesel.
3. Emissions of NO_x, CO and HC using a conventional aftertreatment system were shown to be extremely low with methanol, enabling attainment of emissions at the levels required to achieve Federal Tier II LDV standards.
4. Brake thermal efficiency with ethanol fuel is also favorable compared to that of the baseline diesel engine.
5. The present engine offers the potential for a lower-cost renewable fuel alternative to the diesel, by virtue of its less-complex PFI fuel system.

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REFERENCES

1. M. N. Nabi, et al., "Ultra Low Emission and High Performance Diesel Combustion with Highly Oxygenated Fuel", SAE Paper 2000-01-0231, 2000.
2. N. Miyamoto, et al., "Smokeless, Low NO_x, High Thermal Efficiency, and Low Noise Diesel Combustion with Oxygenated Agents as Main Fuel, SAE Paper 980506, 1998.
3. R. Baranescu, et al., "Prototype Development of a Methanol Engine for Heavy-Duty Application-Performance and Emissions", SAE Paper 891653, 1989.
4. B. Dhaliwal, et al., "Emissions Effects of Alternative Fuels in Light-Duty and Heavy-Duty Vehicles", SAE Paper 2000-01-0692, 2000.
5. N. D. Brinkman, "Effect of Compression Ratio on Exhaust Emissions and Performance of a Methanol-

- Fueled Single-Cylinder Engine”, SAE Paper 770791, 1977.
6. P. Mohanan, M. K. Gajendra Babu, “A Simulation Model for a Methanol-Fueled Turbocharged Multi-Cylinder Automotive Spark Ignition Engine”, SAE Paper 912417, 1991.
 7. T. Ryan, S. Lestz, “The Laminar Burning Velocity of Isooctane, N-Heptane, Methanol, Methane and Propane at Elevated Temperatures and Pressures in the Presence of a Diluent”, SAE Paper 800103, 1980.
 8. V. Battista, et al., “Review of the Cold Starting Performance of Methanol and High Methanol Blends in Spark Ignition Engines: Neat Methanol”, SAE Paper 902154, 1990.
 9. K. Hikino, T. Suzuki, “Development of Methanol Engine with Autoignition for Low NO_x Emission and Better Fuel Economy”, SAE Paper 891842, 1989.
 10. L. G. Dodge, et al., “Development of an Ethanol-Fueled Ultra-Low Emissions Vehicle”, SAE 981358, 1998.
 11. K. Iwachidou, M. Kawagoe, “Transient Unburned Methanol and Formaldehyde Emission Characteristics in Cold Operation of a SI Engine Powered by High-Methanol-Content Fuels”, VIII Int. Symp. On Alcohol Fuels, pp. 443-448, Nov. 13-16, 1988.
 12. N. Iwai, et al., “A Study on Cold Startability and Mixture Formation of High-Percentage Methanol Blends”, SAE Paper 880044, 1988.
 13. D. M. Swain, D. Yerace, *Cold-Start Hydrocarbon Reduction Strategy for Hybrid Vehicles*, M. S. Thesis, University of Michigan, 2002.
 14. W. Clemmens, “Performance of Sequential Port Fuel Injection on a High Compression Ratio Neat Methanol Engine”, SAE Paper 872070, 1987.
 15. C. D. de Boer, et al., “The Optimum Methanol Engine with Electronic Control for Fuel Efficiency and Low Emissions”, VIII Int. Symp. On Alcohol Fuels, pp. 425-430, Nov. 13-16, 1988.
 16. R. I. Bruetsch, K. H. Hellman, “Evaluation of a Passenger Car Equipped with a Direct Injection Neat Methanol Engine”, SAE Paper 920196, 1992.
 17. R. M. Siewert, E. G. Groff, “Unassisted Cold Starts to -29 °C and Steady-State Tests of a Direct-Injection Stratified-Charge (DISC) Engine Operated on Neat Alcohols”, SAE Paper 872066, 1987.
 18. W. H. Haight III, P. C. Malte, “Methanol Preignition Temperature Behavior”, SAE Paper 912414, 1991.
 19. R. L. Bechtold, *Alternative Fuels Guidebook*, SAE International, 1997.
 20. J. B. Heywood, *Internal Combustion Engine Fundamentals*, McGraw-Hill, 1988.
 21. C. DePetris, et al., “High Efficiency Stoichiometric Spark Ignition Engines”, SAE Paper 941933, 1994.
 22. G. R. Neame, et al., “Improving the Fuel Economy of Stoichiometrically Fuelled S.I. Engines by Means of EGR and Enhanced Ignition—A Comparison of Gasoline, Methanol and Natural Gas”, SAE Paper 952376, 1995.
 23. Y. Sato, et al., “Combustion and NO_x Emission Characteristics in a DI Methanol Engine Using Supercharging with EGR”, SAE Paper 971647, 1997.
 24. G. M. Pannone, R. T. Johnson, “Methanol as a Fuel for a Lean Turbocharged Spark Ignition Engine”, SAE Paper 8904535, 1989.
 25. B. Bartunek, et al., “Influence of the Methanol Fuel Composition on Performance and Exhaust Emissions of Diesel-Derived Alcohol Engines”, SAE 881197, 1988.
 26. D. P. Gardiner, M. F. Bardon, “The Effect of Cranking Speed on Cold Starting Performance with Methanol Fuels”, VII Int. Symp. On Alcohol Fuels, pp. 191-195, 1986.