ECONOMICAL, HIGH-EFFICIENCY ENGINE TECHNOLOGIES FOR ALCOHOL FUELS

Matthew Brusstar
U. S. Environmental Protection Agency
National Vehicle and Fuel Emissions Laboratory
Ann Arbor, Michigan 48105-2425
Tel.: (734) 214-4791
FAX: (734) 214-4573
E-mail: brusstar.matt@epa.gov

Marco Bakenhus
FEV Engine Technology, Inc.
Auburn Hills, Michigan 48326-1766
Tel.: (248) 373-6000
FAX: (248) 373-8084
E-mail: bakenhus@fev-et.com

ABSTRACT

Alcohols fuels, principally methanol and ethanol, have the potential to displace a substantial portion of the domestic petroleum consumption in the U. S., used either neat or in blends with petroleum fuels. In order to develop effective policies that encourage economical and environmentally-sustainable use of such fuels, engine technology options must be made available that can achieve these ends. One promising option, being developed by the U.S. EPA's National Vehicle and Fuel Emissions Laboratory, uses low-cost port-fuel-injection, spark-ignition technology with neat alcohol fuels to reach peak brake thermal efficiency levels of over 40%, comparable to state-of-the-art diesel engines [1]. This research has more recently been extended to a full range of blends with gasoline, demonstrating significant efficiency gains using fuel containing as little as 30-50% alcohol by volume. The engine research program described in this work examines the efficiency benefits of higher compression ratio and reduced intake air throttling, enabled by the high octane rating and high dilution tolerance of alcohol fuels. The research centers on a turbocharged, 1.9-liter, 4-cylinder base diesel engine with 19.5:1 compression ratio, modified for port fuel injection and spark ignition. The engine was operated with stoichiometric fueling, to allow the use of conventional three-way catalysts, and to maximize power density. The full range of loads and speeds was characterized, with fuel blends ranging from 10% to 100% alcohol by volume. In addition, an equivalent single-cylinder research engine was run in parallel with the multi-cylinder engine to optimize the combustion system, and to explore cold starting strategies. The results demonstrate a path toward a cost-effective alternative to conventional gasoline and diesel engines, and, moreover, show an economical and environmentally-sustainable means of utilizing methanol or ethanol fuels.

INTRODUCTION

Alternative transportation fuels, especially methanol and ethanol, have been researched extensively for more than two decades, due to their potential economic, national security and environmental benefits. Domestic production of significant quantities of alternative fuels would result in a better balance of trade and a raised gross domestic product, while reducing reliance on imports from politically unstable regions of the world [2, 3]. Moreover, substituting neat alcohol fuels in place of petroleum-based fuels in the transportation sector would also give lower emissions of greenhouse gases, evaporative hydrocarbons and criteria pollutants, and would promote the use of environmentally-sustainable feedstocks [4]. These benefits notwithstanding, neat alcohol fuels have seen only limited success over the last ten years, due to their high cost relative to gasoline on a unit energy basis, as well as many supply and infrastructure challenges.

Presently, only about one percent of the alcohol fuel produced in the U.S. goes toward "neat" alcohol fuels (i.e., those containing nominally 85% alcohol or greater), with the larger fraction going into gasohol [5]. Over the past five years, nearly all of the neat alcohol fuel produced has been in the form of E85, which consists of between 70% and 85% ethanol blended with gasoline. Despite the widespread availability of flexible fuel vehicles (FFVs) that are capable of using these fuels, the growth in E85 production has not kept pace with the overall growth of the ethanol market, for various reasons. For one, retailers and terminal operators have been largely unable to obtain a favorable return on investment with E85 due to low production volumes, high transportation costs, special handling requirements, and tax incentives that favor gasohol blends. Consequently, the sale of E85 is restricted to relatively few outlets, mostly in the Midwest region. Also, the availability of sustainable natural resources ultimately constrains the upper limit on ethanol production (a factor that may, in turn, favor alternative fuels like methanol, which can be more easily produced from readily-available sources such as coal or stranded natural gas reserves). More importantly, however, has been that the market price of E85 has remained closely tied to gasoline prices [6], which puts it at roughly a 25% disadvantage in terms of energy content relative to gasoline (assuming 74% average ethanol content in E85). Using conventional FFV technology, very little of this apparent disincentive can be recovered, making E85 less cost-effective for both fleets and rational consumers.

An important step toward increasing alcohol fuel demand, then, may lie in providing economical engine technology options that utilize such fuels more efficiently, to compensate for the lower fuel energy density. The FFVs produced today, however, use fairly typical gasoline engines, which, because they must retain dual-fuel capability, are not able to take full advantage of the favorable combustion characteristics of alcohols. Engines optimized for alcohol fuel use, on the other hand, may yield efficiencies that exceed that of state-of-the-art diesel engines—or, about one third higher than that of FFV engines. In earlier engine research at EPA with neat methanol and ethanol [1], for example, over 40% brake thermal efficiency was achieved over a relatively broad range of loads and speeds, with peak levels reaching over 42%. Similar work has also been performed with E85 [7], yielding up to 20% fuel economy improvement over baseline gasoline engines. The particular challenge explored in the present work, however, is to determine whether the fuel economy benefits with neat fuels can be realized with fuel blends containing significantly less alcohol, perhaps 30% or lower. Using lower blend fractions would help distribute supply and infrastructure costs over larger product volumes, thus being more cost-effective for fuel suppliers while also effectively improving availability to consumers. In short, were an engine technology made available that could use these more cost-effective alcohol-gasoline blends efficiently, it may represent a long-term market-sustainable option, which might one day foster more widespread alcohol fuel usage.

Relatively little fuel economy and emissions data has been published for engines operating with fuel blends ranging between 10% and 85% ethanol [8, 9]. Ordinarily, neither dedicated fuel vehicles nor FFVs operate in this range for a significant amount of time, since these "intermediate" fuel blends are not produced commercially in the U.S. Consequently, there has been little work to optimize the engine efficiency over this range, improving it to the level where it would offset the additional fuel cost. For example, while nearly a 25% increase in fuel economy is needed to operate economically with E85, only a modest increase of around 8% would be needed with E30. The present work examines the benefits of higher-compression ratio engines with alcohol-gasoline blends, focusing primarily on the range of 10%-50% alcohol.

Neat alcohol fuels have been shown in numerous works to offer some significant benefits over gasoline. Their high octane number gives the ability to operate at higher compression ratio without preignition [9]; its greater latent heat of vaporization gives a higher charge density [10]; and its higher laminar flame speed allows it to be run with leaner, or more dilute, air/fuel mixtures [11, 12]. In addition, alcohol fuels generally yield lower criteria pollutant emissions than gasoline [8, 13], lower evaporative emissions due to somewhat lower vapor pressures [14], and, when renewable feedstocks are used, lower life-cycle greenhouse gas emissions [15]. In the present work, a high-compression-ratio PFI, SI engine is operated using a combination of turbocharging and relatively high levels of exhaust gas recirculation (EGR) dilution to explore the potential brake thermal efficiency gains possible with alcohol-gasoline blends.

Since most of the earlier work at EPA focused mainly on neat alcohol fuels, these programs faced a persistent challenge with hydrocarbon emissions during cold starting [16], as is typically seen with dedicated alcohol fuel engines using a high compression ratio. Such challenges may be mitigated somewhat through secondary air injection [10] or with more volatile fuel additives [17] such as gasoline. The present work addresses cold starting by focusing on alcohol-gasoline blends in the range of 10% to 50% alcohol content, in which startup emissions can be addressed effectively with conventional oxidation catalysts. This work also explores the degree to which these fuel blends retain superior knock resistance at high compression ratio and higher tolerance for dilution with EGR, when used in the unique dedicated fuel engine described below. The efficiency gains demonstrated here for somewhat moderate levels of alcohol content may indicate a path toward a more economical and sustainable alternative for PFI SI engines.

EXPERIMENTAL SETUP

The ongoing research described below is part of EPA's Clean Automotive Technology Program, whose goal is to demonstrate the feasibility of cleaner, more efficient engine technologies, and to transfer these technologies to the private sector. The main focus of this work is on alcohol-gasoline blends, since these are more likely to be cost-effective as a transportation fuel than neat alcohol fuels.

ENGINE AND TEST DESCRIPTION

The engine designed for this work was derived from the 1.9L Volkswagen TDI direct injection diesel engine, modified suitably to accommodate port fuel injectors and spark plugs. The compression ratio for the present study was 19.5, which was optimized for use with neat alcohol fuels and carried through as a baseline for analyzing the relative performance with both methanol and ethanol fuel blends. To accommodate the high compression ratio, the swirl ratio on the inlet ports and the design of the combustion chamber had to be carefully selected to reduce the tendency for engine knock. The design characteristics are described below in Table 1.

A variable geometry turbocharger was employed to maintain the engine's specific power, despite relatively high levels of charge dilution with EGR. EGR, meanwhile, was metered from the turbine exhaust to the compressor inlet using a variable backpressure device in the exhaust, at the expense of a relatively small amount of pumping work (5 kPa or less). The EGR temperature was controlled with a conventional liquid-to-gas cooler, while the fresh air and EGR charge entering the intake manifold (after the compressor) was cooled with a moderately-oversized intercooler. Together, these heat exchangers were able to maintain intake manifold temperatures in the vicinity of 40^{o}C, even at higher speeds and loads.

The port fuel injectors were selected based on earlier work with neat alcohol fuels, which were rated nominally at 30 lb/hr flow 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 high-energy ignition coil with a single-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.

Table 1: Test engine specifications.

Engine Type	4 cylinder, 4-stroke
Combustion Type	Port fuel injection, spark ignition
Displacement	1.9L
Valvetrain	OHC, 2-valve per cyl
Bore	79.5 mm
Stroke	95.6 mm
Compression Ratio	19.5:1
Swirl Ratio	2.0
Injectors	30 lb/hr, 12-hole
Fuel Rail Pressure	4 bar
Spark Plugs	Recessed gap, single electrode
Turbocharger type	Variable geometry

The engine was run with anhydrous chemical-grade methanol and ethanol (with no denaturant), blended with 87 octane ((R+M)/2 method) gasoline. Batch chemical analyses were performed on each blend to verify the heating value, Reid Vapor Pressure (RVP) and density. A summary of measured fuel properties is given in the Appendix.

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 basic principles: (1) High compression ratio, for high efficiency and an expanded range of dilute operation; (2) Turbocharging with high levels of EGR, for and extended load range and low NOx emissions; (3) Stoichiometric fueling, for highest power density and to allow use of conventional three-way catalyst technology.

Methanol and ethanol possess high octane numbers, and are often used as octane improvers in reformulated gasoline blends. Published RON values for methanol and ethanol are between 105-109, compared to between 91-99 for gasoline [18, 19]. As a result, neat alcohols and high-volume-fraction alcohol blends may be run at a significantly higher compression ratio, thereby yielding higher engine thermal efficiency. Earlier works with multi-cylinder SI ethanol engines [20], for example, showed 6% improvement in BSFC when raising the compression ratio only modestly, from 9.2 to 12.0, while still achieving minimum best torque (MBT) spark timing with E22. A compression ratio of 19.5:1 was chosen for this work based on earlier experience with neat alcohol fuels (both ethanol and methanol), which showed this to be the best compromise between full spark authority without knock at high load and dilute combustion range at light load. 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 alcohol fuels.

The engine load control strategy was designed to take advantage of the high flame speeds of methanol and ethanol to run unthrottled, or with less throttling, and therefore more efficiently, over a relatively wide range of loads. The laminar flame speeds of M100, E100 and gasoline are given below in Figure 1 [11, 12], illustrating the dependence on equivalence ratio in lean operation. Alcohol-fueled engines that instead use high levels of EGR to modulate load [1, 7] have demonstrated efficiency gains of greater than 10% over throttled engines, while at the same time giving considerably lower NOx emissions. Engine out NOx levels of well below 1.0 g/kW-hr and peak efficiency around 42% can be achieved in this manner for DI, lean stratified-charge methanol engines [22] and similar improvements in PFI lean burn methanol engines [23]. In the present engine, EGR and boost levels are maintained to achieve best efficiency, and to enable MBT (or near MBT) ignition timing at high loads. Manifold absolute pressure (MAP) was varied between 1.0-2.0 bar absolute, although some throttling was needed to map the engine from about 5 bar BMEP down to idle load. The maximum EGR dilution level ranged from as much as 45-50% EGR for neat alcohols, down to about 30% EGR for 10% alcohol blends. The maximum EGR level was constrained by the acceptable degree of variability in the cycle-by-cycle engine torque, limited for this study at +/-3%.

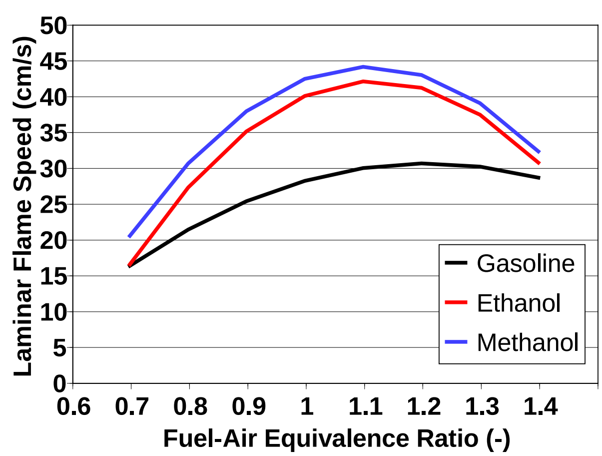

Figure 1. Laminar flame speed of M100, E100 and gasoline as a function of fuel-air equivalence ratio (1 atm, 300K).

The exhaust gas oxygen level was controlled to give stoichiometric fueling, enabling use of conventional exhaust oxygen feedback sensors and a three-way catalyst. Earlier experience at EPA using specially-formulated three-way catalysts with alcohol fuels demonstrated the ability to achieve the level of Federal Tier II-Bin 5 regulations for criteria pollutant emissions, but with relatively high precious metal loading [24]. Similar aftertreatment was used for the present study, but the results shown below focus primarily on the steady-state brake thermal efficiency.

RESULTS AND DISCUSSION

The initial phase of engine testing focused mainly on neat fuels, beginning with M100 and E100. Figure 2 below shows the brake thermal efficiency versus speed and load with M100, over the full range of stable, unthrottled operation.

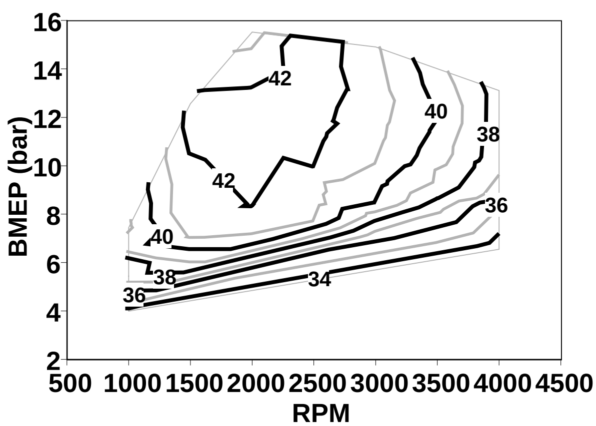

Figure 2. Brake thermal efficiency as a function of engine load and speed for M100.

The lower load boundary of the operating envelope follows the limit of dilute operation, while the upper boundary represents the limit to the range of full spark authority. The upper load limit was able to be extended to beyond 20 bar BMEP with M100, at the cost of some limitation on the spark advance, resulting in a moderate drop in engine efficiency. Nevertheless, Figure 2 demonstrates the high brake thermal efficiency levels possible with M100 over a broad range of engine speeds and loads. The brake efficiency exceeded 40% over a power range of 12 to 75 kW, with a significant part of the map reaching above 42% efficiency.

Figure 3, below, shows the brake thermal efficiency with E100, over the full range of unthrottled operation. In a manner similar to that shown with M100, the brake thermal efficiency remained high over a significant part of the operating map. M100, however, with a marginally superior burn rate and octane rating, gave somewhat better efficiency over a broader load range compared to E100. Relative to typical gasoline engines, on the other hand, whose peak efficiency levels reach the mid-30% range, the results for both M100 and E100 are remarkable, and can be attributed mainly to the ability of alcohols to operate with a considerably higher compression ratio and with substantial levels of EGR. Even when blended with gasoline, these favorable properties of alcohol fuels contribute to the enhanced efficiency levels shown in the subsequent figures below.

Figure 3. Brake thermal efficiency as a function of engine load and speed for E100 (no denaturant).

Figure 4, below, shows the brake thermal efficiency for methanol blends ranging from M50 to M100, at an engine speed of 2000 rpm, with no throttling. As the percentage of gasoline increases, the inlet charge cooling effect and the octane rating both decrease, giving less spark authority at higher loads. At the same time, the engine becomes less tolerant of high levels of EGR at lighter loads, as the higher gasoline fraction suppresses the fuel burn rates. As a result, the engine operates less efficiently and with a significantly reduced load range.

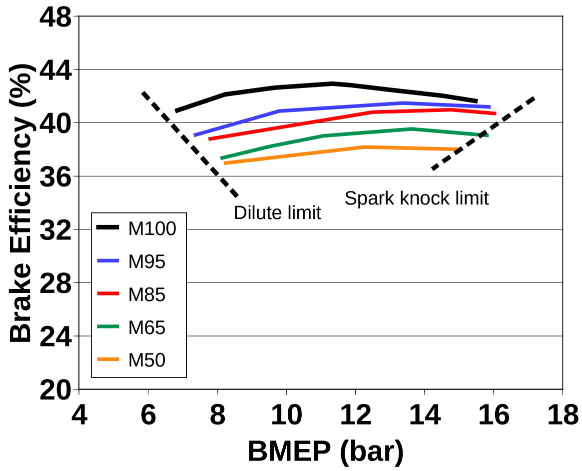

Figure 4. Brake thermal efficiency as a function of engine load for various methanol-gasoline blends, at 2000 rpm.

Figure 5, below, shows the effect of increasing gasoline fraction on ethanol blends, in the range from E10 to E100, at an engine speed of 2000 rpm. Some throttling was necessary with E10 and E30 to extend the engine operation to a broader range of engine loads, since spark knock precluded operation at higher loads. The trends exhibited are similar to those in Figure 4 for methanol blends, in that the range of stable, unthrottled operation became somewhat narrow with increasing gasoline fraction, especially for E30 and E10. Both the peak efficiency and light load efficiency for E30 are nonetheless significantly higher than that for a typical gasoline engine demonstrating the benefits of octane improvement and reduced engine throttling with blends containing as little as 30% ethanol.

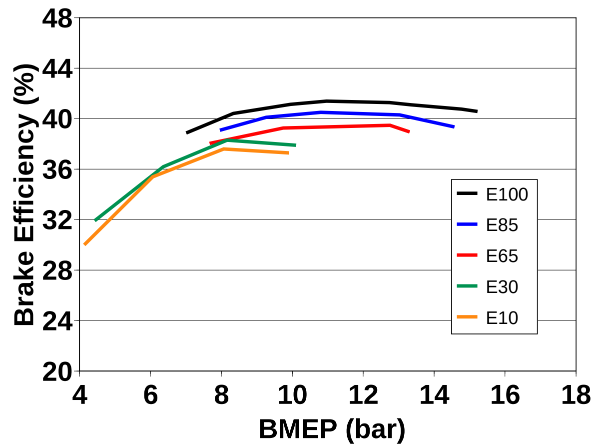

Figure 5. Brake thermal efficiency as a function of engine load for various ethanol-gasoline blends, at 2000 rpm.

The fuel efficiency benefits with E30, shown in Figure 5, point to a potentially attractive fuel from a performance and economic standpoint, for reasons discussed earlier. The full engine efficiency map with E30, given below in Figure 6, demonstrates in greater detail the extent of the higher-efficiency operating range, with peak levels well exceeding that of even the most efficient production gasoline engines [25].

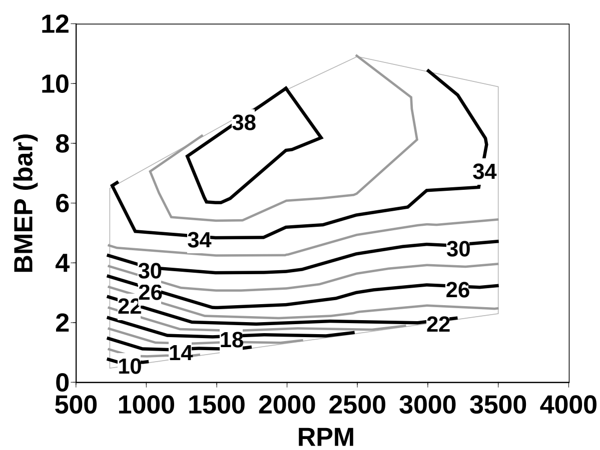

Figure 6. Brake thermal efficiency as a function of engine load and speed for E30 (30% Ethanol-70% gasoline).

Combined with an optimized conventional drivetrain, the efficiency gain shown in Figure 6 for E30 should yield an estimated

10% to 12% gain in fuel economy, and thus more than compensate for the approximately 8% loss in fuel energy density compared to gasoline.

Although the low volatility of neat alcohol fuels gives lower evaporative emissions and some efficiency benefits, at the same time it hinders cold starting. For alcohol blends with significant gasoline content, perhaps 30% or greater, cold starting is not likely to be an unusually difficult challenge, due to the relatively high RVP of the fuel. However, considerable work has been done toward improving cold starting with neat alcohol blends [10, 16, 26]. Below, in Figure 7, the cold starting behavior of M100 in a single-cylinder research engine was explored for an ambient temperature of 20 degrees Celsius. A range of cranking speeds was investigated, but a higher speed was determined to be necessary for successful startup, with no misfires or partial burns. The curves shown in Figure 7 represent the misfire boundaries for a given set of start conditions, above which successful startup was achieved. The effect of increased engine cranking speed is shown to be quite significant, since it raises the startup compression ratio and end compression temperature, while also improving the fuel-air mixing. An even stronger effect is seen with increased inlet temperature, due partly to enhanced fuel evaporation, indicating that substantial improvement in cold starting could perhaps also be made with a charge air heating strategy.

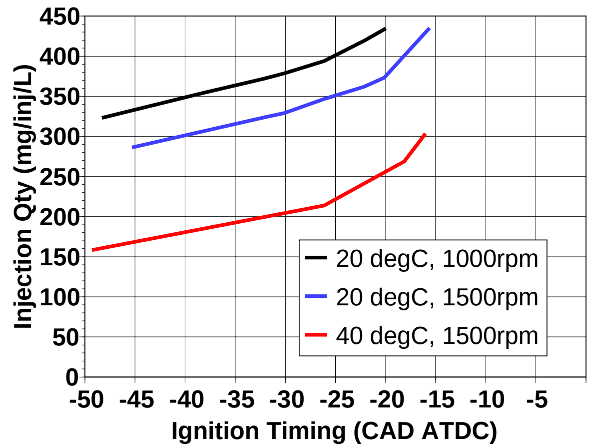

Figure 7. Injection quantity versus ignition timing required for ambient starting of the engine operating with M100.

Further engine research will be needed to ascertain the optimal balance between cold startability and engine performance, with full account taken of the considerable aftertreatment issues. However, the engine technology presented in this work appears to offer promise toward more economical and sustainable use of alcohol fuels.

CONCLUSION

Substantial improvements in brake thermal efficiency have been demonstrated with neat alcohol fuels in a cost-effective port-fuel injected, spark-ignited engine configuration. Recently, this work has been extended to include alcohol-gasoline blends, showing significant benefit with fuels containing as little as 30% alcohol. From the results presented above, it is concluded that:

1. Over 40% brake thermal efficiency can be obtained in a high compression ratio, PFI SI engine using neat methanol and ethanol fuels, resulting from the favorable combustion properties of alcohols.
2. Decreasing the fuel alcohol content generally gives lower brake thermal efficiency and somewhat decreased load range.

3. High efficiency was demonstrated with fuel blends down to 30% alcohol content. Such fuels may present a more economical and efficient means of utilizing alcohol fuels, and provide a path toward their more widespread, long-term use.

REFERENCES

1. M. Brusstar, et al., “High Efficiency and Low Emissions from a Port-Injected Engine with Neat Alcohol Fuels”, SAE Paper 2002-01-2743, 2002.
2. C. L. Gray, Jr., J. A. Alson, “Moving America to Methanol”, Univ. of Michigan Press, 1985.
3. M. K. Evans, “The Economic Impact of the Demand for Ethanol”, Midwestern Governor’s Conference, Feb. 1997.
4. M. A. Raynolds, et al., “A Case Study for Life Cycle Assessment (LCA) as an Energy Decision Making Tool: The Production of Fuel Ethanol from Various Feedstocks”, SAE Paper 982205, 1998.
5. “1995-2004: Estimated Consumption of Vehicle Fuels”, Department of Energy, Energy Information Administration, September 2004.
6. “The Alternative Fuel Price Report”, DOE-EERE Clean Cities, March 28, 2005.
7. D. P. Gardiner, et al., “Improving the Fuel Efficiency of Light-Duty Ethanol Vehicles-An Engine Dynamometer Study of Dedicated Engine Strategies”, SAE Paper 1999-01-3568, 1999.
8. K. J. Kelly, et al., “Federal Test Procedure Emissions Test Results from Ethanol Variable-Fuel Vehicle Chevrolet Luminas, SAE Paper 961092, 1996.
9. M. S. Radwan, “Performance and Knock Limits of Ethanol-Gasoline Blends in Spark-Ignited Engines”, SAE Paper 850213, 1985.
10. L. G. Dodge, et al., “Development of an Ethanol-Fueled Ultra-Low Emissions Vehicle”, SAE Paper 981358, 1998.
11. M. Metghalchi, J. C. Keck, “Burning Velocities of Mixtures of Air with Methanol, Isooctane, and Indolene at High Pressure and Temperature”, Combustion and Flame, vol. 48, pp. 191-210, 1982.
12. N. M. Marinov, “A Detailed Chemical Kinetic Model for High Temperature Ethanol Oxidation”, Int. J. Chemical Kinetics, Vol. 31, No. 3, pp. 183-220, 1999.
13. J. E. Sinor, B. K. Bailey, “Current and Potential Future Performance of Ethanol Fuels”, SAE Paper 930376, 1993.
14. R. L. Furey, “Volatility Characteristics of Gasoline/Alcohol and Gasoline/Ether Fuel Blends”, SAE Paper 852116, 1985.
15. M. Wang, C. Saricks, D. Santini, “Effects of Fuel Ethanol Use on Fuel-Cycle Energy and Greenhouse Gas Emissions”, U. S. DOE, ANL-CTR, 1999.
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. N. R. Silva, J. R. Sodre, “Using Additive to Improve Cold Start in Ethanol-Fuelled Vehicles”, 2000-01-1217, 2000.
18. R. L. Bechtold, *Alternative Fuels Guidebook*, SAE International, 1997.
19. J. B. Heywood, *Internal Combustion Engine Fundamentals*, McGraw-Hill, 1988.
20. A. C. Castro, “Flexible Ethanol Otto Engine Management System”, SAE Paper 942400, 1994.
21. C. DePetris, et al., “High Efficiency Stoichiometric Spark Ignition Engines”, SAE Paper 941933, 1994.
22. Y. Sato, et al., “Combustion and NOx Emission Characteristics in a DI Methanol Engine Using Supercharging with EGR”, SAE Paper 971647, 1997.
23. G. M. Pannone, R. T. Johnson, “Methanol as a Fuel for a Lean Turbocharged Spark Ignition Engine”, SAE Paper 8904535, 1989.
24. G. K. Piotrowski, J. D. Murrell, “Catalysts for Methanol Vehicles”, SAE Paper 872052, 1987.
25. D. Hermance, S. Sasaki, “Hybrid Electric Vehicles Take to the Streets”, IEEE Spectrum, November, 1998.
26. N. Iwai, et al., “A Study of Cold Startability and Mixture Formation of High-Percentage Methanol Blends”, SAE Paper 880044, 1988.

DEFINITIONS, ACRONYMS, ABBREVIATIONS

BMEP	Brake Mean Effective Pressure
CAD	Crank Angle Degrees
EGR	Exhaust Gas Recirculation
Exx	xx% Ethanol blend
FFV	Flexible Fuel Vehicle
Mxx	xx% Methanol blend
PFI	Port Fuel Injection
RON	Research Octane Number
RPM	Revolutions Per Minute
RVP	Reid Vapor Pressure
SI	Spark-Ignition
(A)TDC	(After) Top Dead Center

APPENDIX

Table A1: Measured fuel properties of gasoline, methanol and ethanol

Prop.	**Gasoline***	**Methanol**	**Ethanol**
RVP (psi)	9.72	4.60	2.30
S. G. @ 60 F	0.759	0.792	0.794
Heating Value (kJ/g)	42.79	20.004	26.75

*-Gasoline: (R+M)/2 = 87 octane

Table A2 Reid Vapor Pressure (RVP) of gasoline-alcohol blends (psi)

Alcohol Fraction	Gasoline Fraction	Methanol Blend	Ethanol Blend
100	**0**	**4.60**	**2.30**
90	**10**	**7.20**	**4.30**
70	**30**	**10.00**	**7.00**
50	**50**	**11.40**	**8.70**
30	**70**	**12.05**	**9.50**
10	**90**	**10.00**	**12.40**
0	**100**	**9.00**	**9.00**