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Alcohol Fueled Heavy Duty Vehicles Using Clean, High Efficiency Engines*

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Abstract

Ethanol and methanol derived from a variety of sources could make a substantial contribution to replacing oil-derived transportation fuels and reducing greenhouse gas emissions. Particularly important are next generation, low carbon biofuels derived from agricultural, forestry, municipal and industrial waste by biochemical or thermochemical processes including plasma gasification, as well as specially grown biomass such as switchgrass. Ethanol, methanol or mixtures of these fuels can be used in turbocharged direct injection spark ignition engines which are as or more efficient than diesel engines and also provide advantages of lower cost, lower emissions and higher power. The strong knock suppression resulting from direct alcohol injection enables engine operation with power densities of up to three times that which can be provided by diesel engines. A representative power density is 200 hp/liter. The introduction of these engines in heavy duty vehicles could be relatively rapid because of the need to replace present heavy duty diesel engines in order to meet more stringent air pollution regulations and relatively modest fueling infrastructure requirements. During the initial market introduction phase the fuel could be presently produced ethanol. In addition, these engines could be operated as flexible fuel engines to allow use of gasoline as the main fuel when alcohol fuel is not available or it is more economically attractive to use gasoline. The flexible fuel engines could use a secondary tank of independently controlled direct ethanol injection to prevent knock and allow operation primarily on gasoline without compromising performance. Depending on the application the amount of alcohol from this second tank would be around 15-20 % of gasoline use for prolonged high torque operation long haul trucks with gasoline alone in the main tank. It would be less than 3% for a typical non long haul truck operation drive cycle.

The high power density turbocharged operation enabled by the knock suppression from direct alcohol injection could allow super engine downsizing where, for example, a 5 liter spark ignition engine could be used to provide the same power as a diesel engine with an 11 liter displacement and possibly a 15 liter diesel engine. High power density, alcohol fueled heavy-duty vehicle engines could be used for both long and short haul trucks, buses and off road vehicles. Dedicated ethanol operation could be particularly attractive for farm vehicles. High power density alcohol fueled engines can also be attractive for light duty vehicles.

I. Introduction

There is an increasing need to decrease greenhouse gas (GHG) emissions and reduce oil dependence. Around 70 % of US oil consumption is for ground transportation and ground transportation accounts for about one third of US GHG emissions. Heavy duty vehicles account for about 25% of the ground transportation total. It is estimated that over the next decades CO_2 emissions from heavy duty vehicles will grow faster than light duty vehicles [SMP]. Most of these vehicles are presently powered by diesel engines.

In addition, air quality is adversely affected by emissions by diesel vehicles and more stringent regulations are being put in place. Both emissions from oxides of nitrogen (NOx) and particulate matter (PM) affect the environment and health in urban areas. Emissions from diesel vehicles have decreased substantially since being regulated by EPA in the US and by regulatory agencies elsewhere in the world. The proposed nitrogen oxide (NOx) and particulate matter (PM) emission requirements for Europe, for diesel powered light duty vehicles, are 0.08 NOx g/km and about 0.005 g/km PM. For gasoline powered light duty vehicles, the emissions requirements are 0.06 NOx g/km. The reduced emission from diesel engines is obtained partly through the use of expensive aftertreatment components, such as NOx abatement (SCR catalyst with urea injection) and filters for PM.

In comparison, gasoline powered vehicles have substantially lower emissions. PM is very small in most spark ignited engines. NOx can be controlled to very low levels through the use of the highly effective three-way catalyst, which decreases NOx emission by as much as 98%.

Ethanol, particularly from next generation technologies for production from agricultural, forestry, municipal and other waste as well as from specialty crops and trees offers a potential means for substantial replacement of replace oil derived fuels [NAS]. Ethanol has the attractive features of the high energy density of a liquid fuel and low air pollutant emissions when used in spark ignition engines with a three-way catalyst exhaust aftertreatment system. The use of ethanol in the US, which presently is corn based, is presently geared towards implementation as a gasoline replacement in light duty vehicles, with blends limited to 10% (E10). The large increases mandated for ethanol in the coming years would necessitate the increase of the alcohol allowable in gasoline blends, or the substantial expansion of a high ethanol blend such as E 85 with 79 % ethanol and 21 % gasoline by volume. E85 is presently available in about 2000 stations, mainly in the mid West [E85 stations].

It is challenging to utilize all the mandated ethanol [EISA] or the projected ethanol [Groode, Stark] in light duty fleet. Not only it is necessary to expand the vehicle fleet that can operate on alcohol-gasoline blends with larger ethanol concentration than presently allowed, it would be costly to implement a refueling infrastructure to satisfy the dispersed needs of the light duty fleet. It may be easier to implement expanded ethanol (and other alcohol fuel) use in engines for the heavy-duty fleet. The ethanol can be used as either the sole fuel or in combination with gasoline. The implementation of an alcohol based heavy-duty (HD) fleet may be attractive both because of the need to replace diesel engines to meet more stringent NOx and particulate emissions regulations and a smaller fuel infrastructure than that of light duty vehicles.

In contrast to the case of conventional port fuel injected spark ignition engines, the use the directly injected ethanol in spark ignition engines can also offer the same or improved engine efficiency compared to diesels [Cohn, Bromberg1]. It has been shown that, for heavy duty applications, spark-engine efficiency is comparable to diesel through downsizing and high compression ratio operation, with knock suppressed by direct injection (DI) of ethanol [Blumberg].

In this report, we investigate the opportunity of using ethanol-fueled engines as an alternative fuel in heavy duty vehicle applications and the possibility of obtaining efficiencies which are higher than diesel engine efficiency. Use of methanol is also assessed. These fuels (and mixed alcohol fuels which include mainly ethanol and methanol) can be made from a wide range of biomass feedstocks. Methanol (and, if desired, ethanol or mixed alcohols) can also be made from natural gas. They can also be made from coal.

II. Alcohol Fuels

In the US, there is a large expected growth of the use of ethanol, with about 35 billion gallons mandated by 2022, mandated by the 2007 Energy Independence and Security Act of 2007, up from 11 billion gallons for 2009 [EISA]. About half of the alcohol in 2022 can be manufactured from corn, the rest from various other means, including both bio chemical (sometimes referred to as “cellulosic ethanol”) and thermochemical processes. The thermochemical processes convert the feedstock to synthesis gas which can then be transformed into liquid fuels. These fuels include gasoline and Fischer Tropsch (FT) diesel as well as methanol, ethanol and mixed alcohols. Methanol and mixed alcohols can be produced with significantly greater efficiency and lower cost than gasoline or FT diesel [Stark].

Next generation biofuels can be produced from a variety of sources. It has been estimated that there is a US potential of around 1 billion tons/yr of biomass not including municipal and industrial waste. [Stark, Perlack] Assuming a production potential of 100 gallons of alcohol fuel/ton, this source could potentially provide 100 billion gallons of alcohol fuel per year. Because of the lower volumetric specific energy of the alcohol, this amount of alcohol would displace approximately 50 billions gallons of diesel. This biomass source includes agricultural and forestry wastes and specially planted crops and tress. Agricultural waste could provide 15 billions gallons annually, with processing that is continuously improving [see, for example, POET]. There is enough corn stover, for example, to generate about 9 billion gallons of ethanol /yr (assuming 90% conversion rate and 30% retrieval from the fields). There is also a large amount of forestry feed stock including forestry residues. The challenge for this source is its expensive collection, with difficult conversion at or near the source. Eventually either biochemical or thermochemical processing could be used with this feedstock. It may be advantageous to produce a mixture of alcohols using thermo-chemical conversion. A given fuel value produced in the form of methanol is easier o produce than the same fuel value in the form of ethanol. An additional potential feedstock is switchgrass and fast growing trees.

Municipal and industrial wastes can also be an important feed stock particularly in the near term since it is a negative cost feedstock (due to the fees received for their disposal). In addition fuel can be produced through use of established thermochemical conversion technology which includes plasma gasification [Pavlus]. Processing of all US municipal and industrial wastes could potentially provide up to 50 billion gallons of alcohol fuel per year, which could displace around 25 billion gallons of diesel.

The combined annual alcohol fuel production potential from municipal, industrial, agricultural and forestry together with specially planted crops and trees is thus around 150 billion gallons. This alcohol production could displace around 75 billion gallons of diesel fuel. Realization of one third of this potential could replace 25 billion gallons of diesel fuel, which is about 70 % of diesel fuel consumption for transportation.

Methanol and other alcohol fuels, including mixed alcohols and ethanol, can also be produced from natural gas from stranded natural gas, which cannot be economically transported to market.

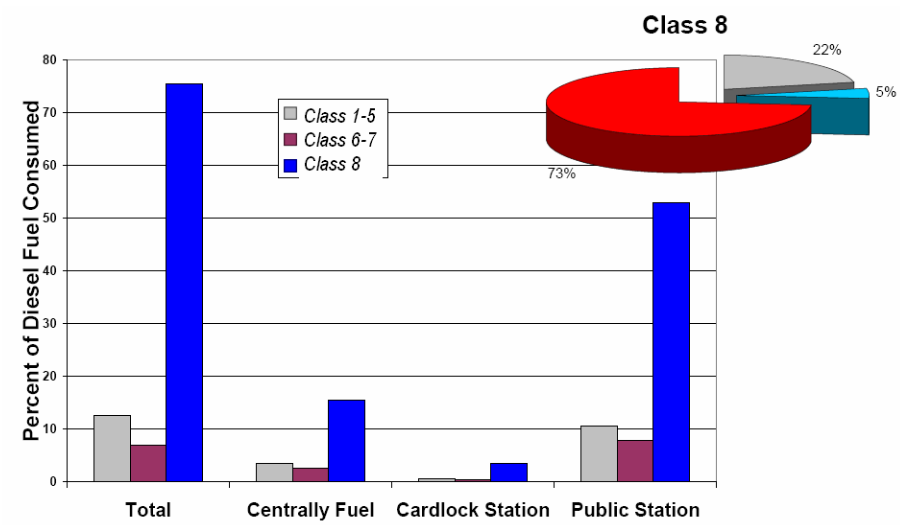

Figure 1. Percentage of diesel consumed by class and by refilling method

With respect to availability outside the US, Brazil has a full infrastructure for ethanol, both in the form of a blend with gasoline and hydrous ethanol (ethanol with a small water content). There are other parts of the world with distribution systems for alcohols, including Sweden and several parts of China. In the latter, both ethanol and coal-based methanol are available locally in several provinces. India and Thailand have also large production of ethanol, mainly from sugar, molasses and cassava.

Diesel fuel has a specific heating value (Lower heating value times density), of about 130,000 BTU/gallon. In comparison, E85 has about 85,000 BTU/gallon. Thus, if the efficiencies of the vehicles and fuel tank sizes are the same, the range of a vehicle operated on diesel will be about 50% more than that vehicle operated on E85. There are means, as described below to narrow this differential.

III. Heavy Duty Engine Applications

Diesel engines are used in applications that include local and long-haul applications. Buses, delivery and refuse trucks and long haul freight are applications of class 6-7 and class 8 diesel-powered engines.

The fuel consumption breakdown for different class vehicles in the US is shown in figure 1. [Jackson] Class 8 vehicles consume about 75% of diesel fuel in the US, and are mainly refueled in public stations (truck stops).

There are other markets where inroads in diesel replacement can have substantial effect, such as in centrally fuelled fleets, such as buses and trucks that operate on urban environments. The fraction of the diesel fuel consumed by these vehicles is about 15%.

The US diesel fuel consumption in 2005 by long haul trucks was about 25 billion gallons. For buses, the consumption was slightly over 1 billion gallons. Other trucks, with 6-liter engines and larger, consumed about 9 billion gallons. The total diesel fuel consumption for US heavy duty vehicles was in 2005 was around 35 billion gallons [EIA]

Emissions from diesel vehicles are of a different nature than spark ignited engines. While diesels have emission issues with nitrogen oxides (NOx) and particulate matter (PM), gasoline engines have issues with hydrocarbons and NOx. It is useful to compare the regulations for both types of vehicles. For heavy duty vehicles, Figure 2 shows the emission regulations in Europe and in the US. The PM emissions are comparable, but in the US the NOx are more stringent than those in Europe.

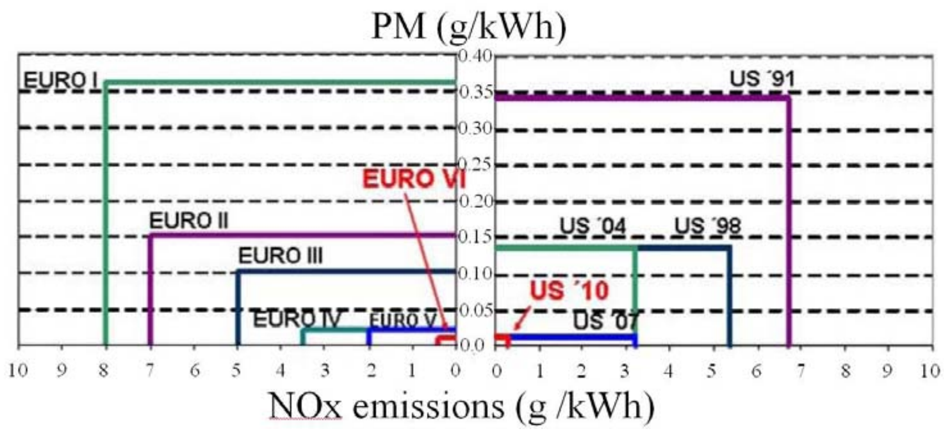

Figure 2. HD Emission control evolution in Europe (left) and US (right). (adapted from Rogers). Vertical axis corresponds to PM g/kWh; horizontal scale corresponds to NOx g/kWh.

On December 21, 2000 the EPA signed emission standards for model year 2007 and later heavy-duty highway engines. The California Air Resources Board (CARB) adopted virtually identical 2007 heavy-duty engine standards in October 2001. The rule includes two components: (1) emission standards, and (2) diesel fuel regulations. The first component of the regulation introduces new, very stringent emission standards for 2010 emissions from heavy duty vehicles, as follows:

* PM—0.01 g/bhp-hr
* NOx—0.20 g/bhp-hr
* NMHC (non-methane hydrocarbons)—0.14 g/bhp-hr

The PM emission standard took full effect in the 2007 heavy-duty engine model year. The NOx and NMHC standards is being phased in for diesel engines between 2007 and 2010. The phase-in would be on a percent-of-sales basis: 50% from 2007 to 2009 and 100% in 2010. Gasoline engines used in HD applications were subject to these standards based on a phase-in requiring 50% compliance in 2008 and 100% compliance in 2009. Very few diesel engines meeting the 0.20 g/bhp-hr NOx requirement will actually appear before 2010. [Dieselnet]

IV. Infrastructure issues

For heavy duty application, characteristics of the distribution system are such that centrally fueled vehicles consumed about 20% of the diesel fuel, cardlock station about 5% and public stations (truck stops) about 75%.

For the centrally fleet fueling, there are around 25,000 stations for heavy duty trucks. Although it should not necessarily be a problem to provide alcohol fuel distribution systems for centrally fueled fleets, the large number of stations implies a substantial cost for developing limited infrastructure. The fuel use in these refueling stations is about 25,000 gallons per month per station, and it is necessary to provide for tanker-delivery of fuel. About 16% of all on-road diesel is consumed by vehicles that are refilled at central fueling stations.

In contrast about 55% of the diesel fuel is consumed by heavy duty vehicles that refill at more than 5000 truck stops in the US. Although this is a large number, it is only about 2.5 times the number of present (2009) E85 stations in the US. The average diesel provided by these stations is about 200,000 gallons per month although it varies from a low about 10,000 gallons/month to about 1,000,000 gallons per month.

The present small distribution system for E85 is not particularly relevant to Heavy Duty applications, as the stations are usually out of the way from the main freight routes.

V High Power Density Spark Ignition Engines In Heavy Duty Trucks

Conventional spark ignited engines, running on gasoline, have been known to be substantially less efficient than diesels, especially at light loads. [Heywood] This is mainly due to the presence of knocking in spark ignited engines which limits the compression ratio and the amount of pressure boosting. Diesel engines are around 25 % more efficient than conventional gasoline engines.

It is possible to substantially increase the compression ratio and the manifold pressure without reaching knocking conditions through the use of knock avoiding technologies. HEDGE (High Efficiency Dilute Gasoline Engine) is one approach, using highly diluted operation with gasoline as the fuel [HEDGE]. The HEDGE approach results in substantially decreased exhaust temperature, decreasing the possibility of energy recovery in the exhaust. Also, there is limit on the peak pressure in the cylinder, due to the large amount of EGR required to prevent knock, as well as decreased efficiency due to the lower value of γ.

With the use of alcohol-based fuels it is possible to reduce much further the tendency of knocking in spark ignited engines. In some cases, it is possible to even eliminate it for all practical purposes. The use of direct injection of E85 to provide very strong knock suppression and enable diesel like efficiency in a port fueled gasoline engine has been demonstrated in engine tests [Stein, Agarwal]. The same benefit should of course be realized in an engine operating on E85 alone.

By eliminating the knock constraint, much higher compression ratios can be used. Similarly, turbocharging allows for substantial engine downsizing. The high knock-free pressure resulting from the use of direct injection in combination with stoichiometric operation with no EGR at high torque and use of high rpm operation makes possible a engine power density which is up to three times that of a diesel engine.
A representative number for the alcohol engine power density is 200 hp/liter.

In the next two sections two alcohol based fuels will be described. The first section covers ethanol blends, while the second one covers methanol blends. The third section covers efficiency estimates of these engines.

V.a Direct Injection Alcohol Engines

The high octane of ethanol, as well as the large evaporative cooling of ethanol, have been shown to strongly suppress the knock in spark ignited engines [Stein, Agarwal] High compression ratio and/or high turbocharging can be achieved when this fuel is used in conjunction with direct injection.

The model developed by Bromberg [Bromberg1] has been used in order to evaluate the limits of the concept. The model uses a simple description of the manifold/turbocharger, in conjunction with a chemical kinetics code. The mechanism used for the calculation is the PRF mechanism developed by Curran [Curran]. The model includes the Marinov model for ethanol combustion [Marinov], as well as mechanisms for n-heptane and iso-octane. The chemical kinetics code has been benchmarked with experiments in appropriate temperature-pressure regimes. A gasoline with the appropriate octane can be modeled using appropriate blends of n-heptane and iso-octane.

Two ethanol blends have being studied. The first one is E85, while the second one is hydrous ethanol. It is assumed that the E85 fuel composition is of 79% ethanol and 21% gasoline (by volume). This value corresponds to summer blends of E85; winter grades have larger gasoline concentration. We assume that the gasoline mixed with the ethanol is regular gasoline, with an octane number of 87. This gasoline is modeled as 87% iso-octane, 17% n-heptane, by volume.

The second ethanol blend is hydrous ethanol which is ethanol produced prior to the final dewatering step. The water concentration in the ethanol is ~ 5% by volume and is referred to as h5EtOH. This fuel has substantially higher evaporative cooling potential, as it is not diluted with gasoline, plus the water has substantially higher vaporization enthalpies than ethanol. Although it could in principle be readily obtained from ethanol producers, hydrous ethanol is not available at service stations in the US, but it is available in other countries. In particular, it is widely available in Brazil.

The analysis has been performed assuming that the manifold pressure was traded off with compression ratio. At a given manifold pressure, the compression ratio that results in auto-ignition of the unburned fuel was determined (borderline knock). Condition of just slightly lower compression ratio where used as the definition of knock-free. The process was repeated for a different manifold pressure.

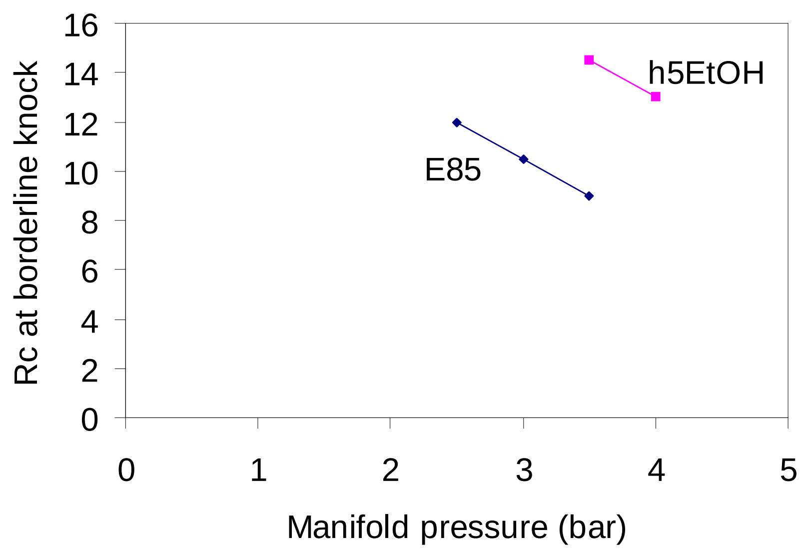

Figure 3 Compression ratio (Rc) at borderline knock as a function of the manifold pressure (absolute) for E85 and 5% hydrous ethanol.

The peak pressure in the cylinder of the knock-free compression ratio has also been monitored. The model assumed instantaneous evaporation of the DI fuel at IVC (Inlet Valve Closing), in order to maximize the cooling effect. It should be noted that because of the imprecise nature of the knock process and the models, the results are to be taken as trends, and experimental studies need to be performed to bench-mark the model. It is assumed that the engine speed is 3000 rpm for all the cases considered.

V.a Ethanol Operation

The results for ethanol blends are shown in Figure 3. The manifold pressure in Figure 3 is the absolute pressure. As expected, the knock-free compression ratio decreases with increasing manifold pressure. It is possible to increase the manifold pressure by about 1 bar through a decrease of 3-4 numbers in the compression ratio. An engine operating with E85 with a manifold pressure of 3.5 can be knock-free if the compression ratio is decreased to about 8-9. It also should be noted that the use of hydrous ethanol results in such high compression ratios and boosting that, for all practical conditions, the knock limit is removed.

Figure 4 shows the peak pressure at knock-free conditions. For the case of the E85, the maximum pressure indicated by the engine is about 140 bar. For the case of hydrous ethanol with 5% water, the peak pressure is about 200 bar. It is likely that peak pressure could limit the operation of an IC engine to lower values, in particular if the engine head is made from aluminum. It has been assumed that there is no EGR and no spark retard. Both of these techniques may be useful for decreasing the peak pressure.

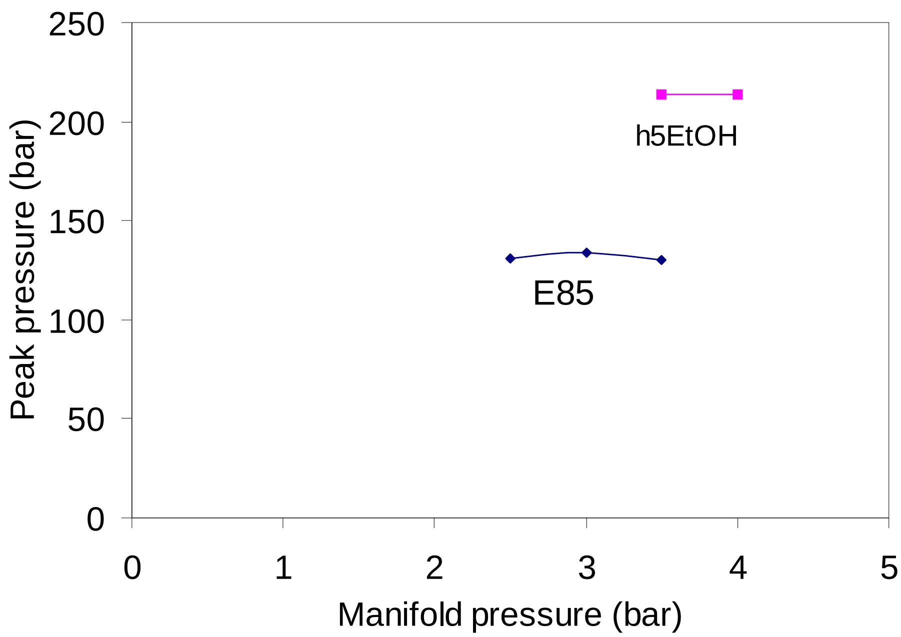

Figure 4. Peak in-cylinder pressure as a function of the manifold pressure (absolute) at borderline knock, for two different fuels (E85 and 5% hydrous ethanol).

It is interesting to note that model predicts that the peak pressure in the cylinder is unchanged when trading off compression ratio for boosting (manifold pressure). It will be interesting to determine experimentally whether this is the case. The conclusion is valid only for direct injection of the alcohol.

V.b Methanol Operation

In this section, the use of methanol blends is described. Methanol is a very good fuel for spark ignited engines, from a combustion point of view. A substantial program was carried out in the US in the late 80's and presently in China using this fuel.

The heat of vaporization per Joule of combustion energy is about 9 times larger for methanol than gasoline, and about twice that of ethanol. Thus, the methanol has much larger equivalent octane, especially when it is directly injected [Bromberg3].

Figure 5 shows the results for methanol/gasoline blends, in the form of M85 (assumed to be 15% by volume gasoline, the rest methanol). The assumptions are similar to those in the previous section for ethanol blends. The Curran/Marinov mechanism includes methanol chemistry. The maximum pressures are probably above what can be made in production engines, and thus it will be necessary to either operate with spark retard or limited EGR. Similarly, the compression ratios are probably as high as could be desired in a spark ignition engine [Heywood].

The cooling effect of either M85 or h5EtOH is so large that a substantial amount of the evaporative cooling will occur later in the compression stroke, reducing the cooling effect. It should be stressed that the calculations in this and the previous sections are to be used to identify trends rather than to take the actual values.

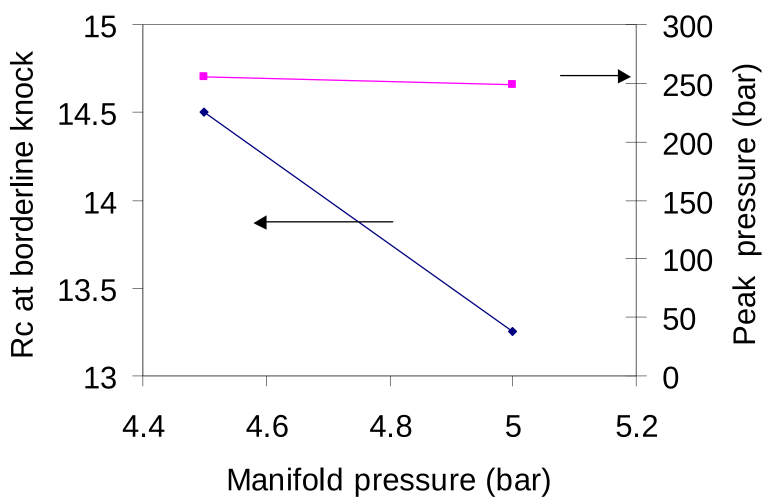

Figure 5. Compression ratio at borderline knock and peak cylinder pressure as a function of the manifold pressure; 3000 rpm, MBT (Maximum Brake Torque timing), for M85 directly injected.

V.c Estimates of performance of directly-injected alcohol engines

A computational study was carried out to determine the performance of a HD engine operating with alcohol-based fuels [Blumberg]. It was shown that for comparable peak pressures, a SI engine operating stoichiometrically, with no EGR, can have substantially more torque than a diesel engine, which requires heavy EGR and dilute operation for emission control (at the high torque points). Thus it is possible to downsize the directly injected engine, operating at constant torque.

The investigation by Blumberg showed that a diesel engine and a downsized engine have just about the same efficiencies, throughout the range. There is slightly higher efficiency at high load, slightly lower efficiency at the lower loads, as shown in Figure 6. Two engines have been considered. A conventional heavy duty diesel engine with 11 liter displacement (Volvo engine, MD11). The second engine is a gasoline engine, 7 liter, using the alcohol boosted concept, with a displacement of 7 liter. At the higher torque points, it is required to have most of the fuel directly injected, if the antiknock agent is E85. If, however, hydrous ethanol is used instead, the amount of antiknock agent can be decreased by about a factor of about 2 [Blumberg1].

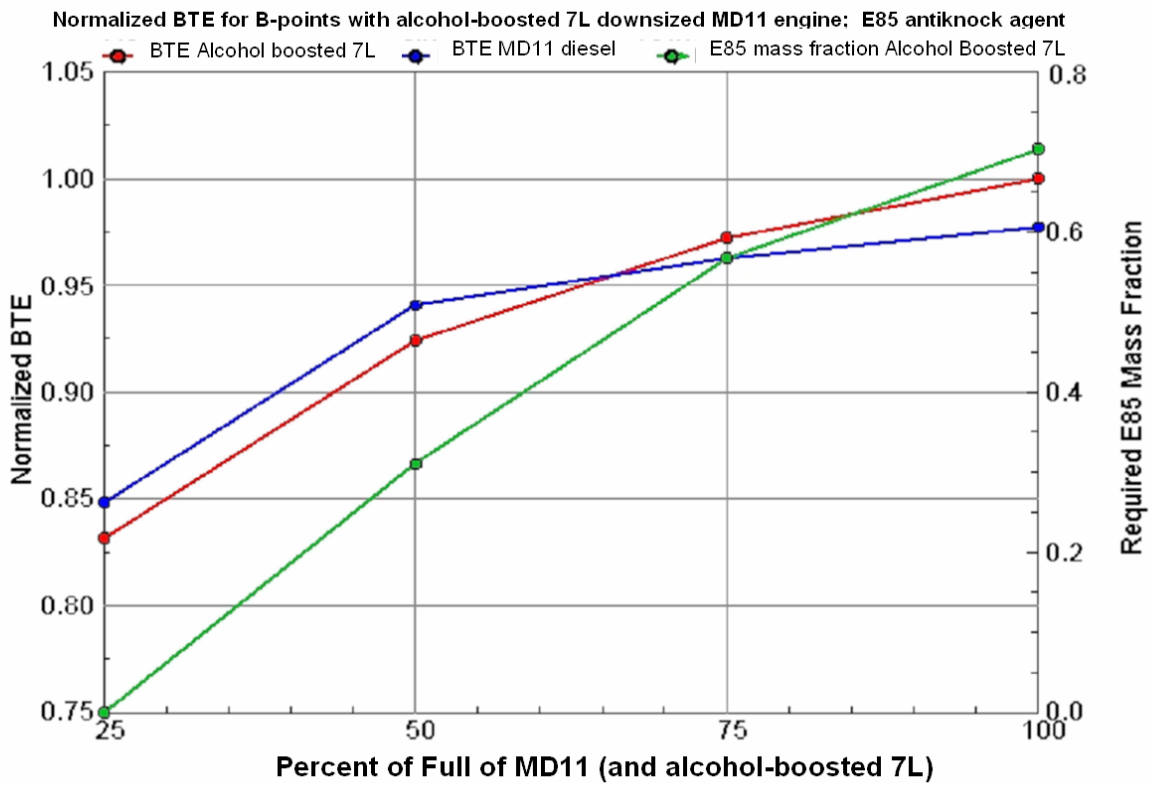

Figure 6. Normalized BTE (brake thermal efficiency) and E85 fuel fraction (by mass) as a function of percent of full load for B-speeds of the ESCAPE cycle for the DI alcohol injection 7L engine [Blumberg]

Even if operating at high torque, an SI engine could operate at higher engine speeds that a diesel engine. For constant power, it would be possible to further downsize the engine, although this would require operation at slightly higher engine speeds (for a given power). There will be a small decrease in efficiency by the need to increase the engine speeds.

Table 1 shows extrapolations from the previous work by Blumberg, with downsizing taken to an extreme. We extrapolate the engine performance assuming that the peak cylinder pressure of the downsized engine is the same as the peak pressure of the diesel engine, while engine speed is limited by SI engine speed. The conditions of the engine are shown for conditions that represent the B-loading of the ESC engine map, corresponding to about 1500 rpm for the engine under consideration. The four points of the B-loading of the ESC test are shown, corresponding to 25%, 50%, 75% and 100% of full torque at 1500 rpm. The diesel engine used as the baseline is 11 liters. Only the ratios of the efficiency of the SI engine to that of the baseline diesel engine at that particular point are presented, in order to protect confidential information from the manufacturer. Similarly, the peak pressures have been normalized to that of the diesel engine, at the B100 point.

The direct alcohol injection 11 liter and 7 liter engine columns are results that have been presented previously [Blumberg]. The entries represent ratio between the gasoline engine efficiency and baseline engine efficiency at the same speed and torque. As the torque is increased and relative friction is decreased in the gasoline engine, there is marked increased in efficiency, which at the high torques is very close to the diesel efficiency.

Table 1. Downsizing potential, at constant peak pressure and constant engine speed. The entries in the first 4 rows are ratios of the efficiency of the high power density spark ignition alcohol engine to the reference diesel engine. The next three rows give engine size, pressure and speed ratios.

Fraction of torque	Diesel baseline	Alcohol 11 liter	Alcohol 7 liter	Same peak pressure as Diesel Engine	Same peak speed as Spark Engine
	Efficiency ratio				
0.25	1	0.87	0.98	1.02	1.07
0.50	1	0.92	0.98	1.01	1.03
0.75	1	0.97	1.01	1.03	1.04
1	1	0.99	1.02	1.04	1.05
Engine size	11	11	7.00	5.43	3.62
Peak pressure, B100	1	0.45	0.775	1	1
Max Engine speed	1	1	1	1	1.5

The direct alcohol injection 11 liter engine, with same displacement as the baseline diesel engine, operates at about half the peak pressure as the diesel. This is because of the stoichiometric operation and lack of EGR which substantially increase peak pressure in the diesel, especially at high torque. The engine was downsized to 7 liter, but even then, the peak pressure was about 78% of the peak pressure in the baseline diesel, allowing for further downsizing. The engine performance is then extrapolated to further downsizing so that the peak pressures in the gasoline engine and the baseline diesel were the same. In this case, the efficiency of the gasoline engine is slightly higher than that of the diesel throughout the torque range for the B-speeds. At higher torques, relevant to heavy duty long haul application, the gasoline engine is about 4% more efficient than the baseline diesel engine.

It should be noted that the fuel penalty associated with aftertreatment in the case of the diesel engine is not included in these calculations, only the engine efficiency. When included, the gasoline engine will be a few percentage points higher than indicated in the table.

Finally, the case when the engine up-speeding is used to further decrease the engine size is shown in the last column of the table. The engine speed throughout the map is increased by a factor of 1.5, and the engine downsized by the same factor. The advantage of the gasoline engine, even at light loads, is suggested by the last two columns of Table 1.

On noticeable feature in Table 1 is the potential to substantially higher BMEP values of the knock-free, stoichiometric, undiluted gasoline engine, substantially higher than the baseline diesel. The potential exists for 40 bar BMEP. In this case a 3.6 liter engine could provide the same power as a 11 liter diesel engine. With similar downsizing, a 5 liter engine could provide the same power as a 15 liter diesel engine

VI. Increased Efficiency:

There are several options available to spark ignited engines for increased overall efficiency, even at constant engine efficiency. Two such options are bottoming cycle and fuel reformation.

Table 2
Useful enthalpies and associated powers for baseline diesel engine and high power density spark ignition (SI) alcohol engine for the A-100, B-25, B50, B75 and B-100 points in the ESC map

		A100	B25	B50	B75	B100
Fuel energy flow rate	kJ/s	660	221	398	584	742
Diesel						
Temperature, post turbine	(K)	654	580	598	625	665
Mass flow rates	kg/s	0.41	0.24	0.35	0.44	0.48
Recoverable energy	kJ/s	38	2	10	26	50
Fraction recoverable			0.01	0.02	0.04	0.07
SI						
Temperature, post turbine	(K)	849	787	869	880	884
Mass flow rate	kg/s	0.24	0.08	0.14	0.20	0.26
Recoverable energy	kJ/s	81	21	52	77	102
Fraction recoverable		0.12	0.09	0.13	0.13	0.14

VI.a Efficiency Enhancement by Exhaust Enthalpy Utilization

The enthalpy of the exhaust can be used in a bottoming energy recovery system downstream from the engine. The exhaust enthalpies have been calculated for both the cases of diesel combustion and for stoichiometric combustion of ethanol, at the B-speed points of the ESC cycle. Only the change in the enthalpy from the post-turbine temperature above 300 C (573 K) is assumed to be available for the post-engine energy recovery. Using the available enthalpy and the flow rates at the corresponding conditions, it is possible to calculate the energy rate (power) available for recovery. The data from the study by Blumberg [Blumberg] has been used in this study.

Table 2 shows the results for the maximum toque at the A speed of the ESC cycle (A100) and the for torque levels of the B-speeds of the ESC cycle (B25, B50, B75 and B100). For diesel, with higher flow rates but lower temperatures, the available energy rate for recovery is about half that of the SI engine. In the case of a spark ignition engine with direct injected alcohol, it is possible to recover about 15% of the heating value of the fuel. The fraction of recoverable energy is smaller in the case of A100 than B100 because of reduced post-turbine temperatures.

Means of recovering the energy are presently being investigated, as technologies for further increasing the efficiency of internal combustion engines. The bottoming cycle could be a Rankine cycle [Teng], a thermo-electric [Fairbanks], or an endothermic fuel reformation, where the reformate has higher heat capacity than the original fuel. The possibility of fuel reformation in the case of alcohol fuels is described next.

VI.b Efficiency Enhancement By Fuel Reformation

In addition to very high octane, methanol and ethanol are fuels that are easily catalytically reformed into hydrogen-rich gas through endothermic pyrolytic decomposition, at low temperatures. Fuel reformation is more attractive for heavy duty vehicles which are operated for a substantial amount of time at low torque than for vehicles that have prolonged high torque operation such as long haul trucks. For vehicles operating a substantial amount of time at low torque, use of ultra dilute operation at low loads can provide up to 12% increase in fuel efficiency [Tully]. The gain is due to decreased throttling, better value of γ because of the reduced residuals, and decreased thermal losses to the cylinder walls, due to lower in-cylinder gas temperature. In the case of high torque, the only efficiency improvement possible is due to reduced heat exchange to the cylinder walls, and thus the maximum improved efficiency is ~ 5% [Tully].

By reforming the methanol onboard the vehicle using exhaust heat, some of the energy in the exhaust is recovered, increasing the energy content of the fuel by about 10%, adequate for capturing the heat available if most of the fuel is reformed, as indicated in Table 2. There is a tradeoff, as reforming methanol results in hot hydrogen rich gas reduces the evaporative cooling of the fraction of the methanol that is not passed through the reformer. Synthesis gas (H_2 + CO) has relatively high octane [Topinka], but much smaller than the effective octane of directly injected methanol [Bromberg3]. In order to explore the tradeoff, the model described in reference [Bromberg1] has been modified to include H_2 and CO, in addition to directly injected methanol. It is assumed that the fuel available is mostly methanol (1% gasoline by volume). It is assumed that the reformed methanol is converted only to H_2 and CO. It is assumed that the reformate H_2 and CO is injected into the manifold downstream from the turbocharger, as the liquid methanol can be pressurized with lower energy consumption than the gaseous reformate. The reformate temperature is 300 C, corresponding to the limit of reformation assumed in obtaining Table 2. It is assumed that the post-turbo air and the reformate are cooled through the intercooler assumed in above mentioned model [Bromberg1]. The fraction of the methanol that is not reformed is directly injected into the cylinder.

The results of the calculation are shown in Figure 7. The compression ratio at borderline knock is shown as a function of the fraction of the methanol that is reformed, for a boosting of 4. As the fraction of methanol that is reformed increases and the fraction of the directly injected methanol decreases, the knock-free compression ratio decreases. Increasing the reformate fraction from 25% to 40% decreases the borderline compression ratio by 4 units.

It is interesting to note that in comparing Figures 5 and 7, the maximum borderline knock with M85 direct injection at ~ 4 bar is ~ 15.5 (by extrapolation), vs 14.5 for the case of 25% methanol reforming. The amount of methanol directly injected into the cylinder in the case of the 25% reformation (and thus, the evaporative cooling) is slightly smaller than the amount injected in the cylinder in the case of 100% M85 operation.

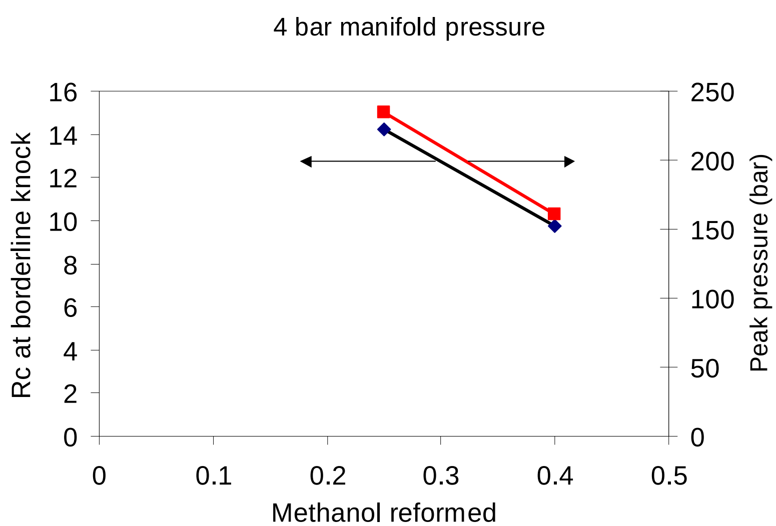

Figure 7. Compression ratio and peak pressure at borderline knock conditions, for an engine with a boost pressure of 3 bar (4 bar absolute), as a function of the fraction of the methanol that is reformed.

The calculations have been performed at stoichiometric conditions. Our goal is to illustrate the opportunity, rather than a detailed optimization of the system. In addition, the temperature of the exhaust would be decreased because of the lean operation. Under those conditions it is likely that the allowable heat will match the endothermic heat requirements, with about 50% of the methanol being reformed. It has been determined that, with constant torque, there is little difference when comparing knocking conditions at lean vs stoichiometric [Topinka]

A further improvement in efficiency could be obtained by performing the thermal decomposition of methanol at high pressure and expanding the high pressure, high temperature reformate through a turbine that drives a generator, or injection at high pressure in the cylinder, after inlet-valve closing.

For ethanol, the decomposition reaction generates CH_4, CO and H_2, according to the equation:

$$C_2H_5OH \rightarrow CH_4 + CO + H_2 \; (+50 \text{ kJ/mol}) \qquad (1)$$

Novel catalysts have been found that result in good conversions at temperatures as low at 300 C [Morgenstern]. This process has an endothermicity of about 5%, when all the fuel is reformed. The value is too small to be able to absorb the available energy shown in Table 2.

An alternative process would be to combine equation (1) at low temperatures with steam reforming of the ethanol at higher temperatures, equation (2). If the fuel is hydrous ethanol, then the stronger endothermicity of the process at the higher temperature. For steam reforming, the equation is:

$$C_2H_5OH + H_2O \rightarrow + 2CO + 4H_2 \; (+715 \text{ kJ/mol}) \qquad (2)$$

This reaction requires higher temperature, and it is not clear that it can be achieved at the exhaust temperatures of Table 2.

The hydrogen rich gas can be used to facilitate lean burn operation, particularly useful at light load conditions where the engine is throttled. Operating lean decreases the friction losses in the engine, decreases in-cylinder heat transfer and improves the ratio of specific heats, further improving efficiency at light loads. Because of the strong turbocharging and downsizing, the efficiency improvement from lean operation needs to be determined. Lean operation of the aggressively downsized engine minimizes the use of engine throttling.

The engine could operate stoichiometric or lean at high loads, depending on application and the availability of alcohol. The emissions of nitrogen oxides (NOx) can be reduced to very low levels during lean-burn by operation at low equivalence ratios (< 0.5). [Apostolescu], at the expense of increased hydrocarbons and CO. As opposed to other reformer work [Tully, Topinka], the reformate does not have a large amount of nitrogen, improving the effect of reformate addition in improving the flame speed and thus the misfire constraint.

VII. High Power Density Flexible Fuel Engines

High power density, direct alcohol injection engines can also be operated in an “alcohol boosted flexible fuel engines “. These engines would have a main fuel tank that contains gasoline. alcohol or a mixture of the two fuels. They would also have a secondary tank that would contain only alcohol fuel which would only be directly injected when and in an amount needed to prevent knock at high torque [Cohn, Bromberg]. For long haul truck operation where there is prolonged high torque operation, the required alcohol consumption from the second tank would be around 15- 20 % of gasoline consumption for gasoline use alone in the main tank for an engine power density that would give an efficiency comparable to a diesel engine. The ethanol consumption would be more for a higher power density engine.

For heavy vehicle applications where there is no prolonged high torque operation, the consumption of the alcohol fuel from the secondary tank over a typical drive cycle can be less than 3% of the consumption of the fuel from the primary tank and in some cases around 1% over a typical drive cycle. The alcohol from the secondary tank can be viewed more as an “octane boost fluid.” The fuel from the primary tank can be introduced into the engine by either port or direct injection. Direct injection of the fuel from the primary fuel can reduce the amount of octane boost fluid by around a factor of two relative to the case of port fuel injection.

Cold-start of the flexible fuel direct injection engine or a dedicated direct injection engine may benefit from a means to increase the conversion of alcohol into the gaseous state during initial start up. One possibility is to use a plasma fuel reformer [Bromberg4]. This device would rapidly convert the liquid alcohol into a hot hydrogen-rich gas. Alternatively, introduction of a light ether (such as DME) from a separate container can be used to prevent misfire.

VIII. Summary

Alcohol fuels, particularly from a wide range of next generation biomass feedstocks including agricultural, forestry, municipal and industrial waste and specially grown crops and trees can potentially offer a substantial substitute for oil derived fuel. Alcohol fuels can also be produced from natural gas. Use of ethanol, methanol or alcohol mixtures could be particularly attractive for relatively rapid introduction into heavy duty vehicles because of the need for cleaner engines than present diesel engines and a relatively modest fueling infrastructure requirement.

Small, high power density, spark ignition engines which are fueled with ethanol methanol or mixed alcohols can be used as a substitute of heavy duty diesel engines, with higher engine thermal efficiency and much reduced size and weight. The cost of the engine and exhaust system would be considerably less than that of a diesel engine. A key factor in the lower cost is the use of the well established, highly effective and relatively low cost three-way catalyst system for exhaust emissions control. In contrast diesel engine vehicles will require considerably more complex and expensive exhaust treatment systems, Moreover, even with these systems diesel engines will not be as robust as spark ignition engines in dealing with the possibility of even more stringent emissions regulations in the future. By removing the knock limit on spark ignition engine operation through direct alcohol injection a 3.6 liter turbocharged spark ignition engine could potentially be used to replace a diesel engine with a displacement as high as 11 liters

A direct injection alcohol engine downsized by around a factor of three could have an efficiency advantage over the diesel of about 4-5%. In addition, it has been shown that reforming about half the methanol or ethanol would result in capture of about an additional 5% from the exhaust, as a bottoming cycle. The lean operation could result in an additional 5% improvement in efficiency. There could also be a small increase in efficiency with up-speeding and further downsizing.

The calculations do not include the fuel penalty expense needed for the diesel engine aftertreatment, which when included will further increase the comparative advantage of the spark ignited engine by a few percentage points. The calculations suggest that for long haul trucks a 5.0 liter direct injection ethanol engine could be used to replace a diesel engine of at least 11 liter displacement and perhaps a 15 liter engine

For introduction of the alcohol fuel, smaller, above ground tanks could be distributed, at a fraction of the cost of a regular filling station with an underground tank. Because of the limited number of trucks during introduction of the technology, these above ground tanks would satisfy the demand at lower costs. [Methanol] Cheaper units, similar to those placed by the NY State Thruway Authority for refueling of methanol vehicles are even less expensive, around $5000 per tank [Dolan].

In addition to long haul freight trucks, high power density alcohol engines could also be used for other trucks, for buses and for various off-road applications. The potential off road applications include farm equipment, construction equipment and vehicles that would benefit from substantially more power than the largest diesel engines.

For both long haul and non long haul heavy-duty vehicles, the high power density engine vehicles would be lighter, cleaner and cheaper than comparable diesel engines. They would use three-way catalyst for NOx, CO and hydrocarbon control, with little or no soot production. They may also be cheaper to operate, depending on the relative cost of gasoline, diesel and alcohols. In terms of maintenance, they may require more frequent maintenance than the diesel engine that they replace. And the fuel tanks will need to be larger (or more frequent refills) because of the lower volumetric heat content of the fuel.

With an appropriate engine fuel management system for alcohol boosting from a secondary tank, direct injection alcohol engines could also be operated as high power density flexible fuel engines which could operate primarily on gasoline without losing performance. The high efficiency downsized alcohol engine approach described in this report could also be used in light duty vehicles.

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