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# Optimal Use of E85 in a Turbocharged Direct Injection Engine

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## ABSTRACT

Ford Motor Company is introducing "EcoBoost" gasoline turbocharged direct injection (GTDI) engine technology in the 2010 Lincoln MKS. A logical enhancement of EcoBoost technology is the use of E85 for knock mitigation. The subject of this paper is the optimal use of E85 by using two fuel systems in the same EcoBoost engine: port fuel injection (PFI) of gasoline and direct injection (DI) of E85.

Gasoline PFI is used for starting and light-medium load operation, while E85 DI is used only as required during high load operation to avoid knock. Direct injection of E85 (a commercially available blend of ~85% ethanol and ~15% gasoline) is extremely effective in suppressing knock, due to ethanol's high inherent octane and its high heat of vaporization, which results in substantial cooling of the charge. As a result, the compression ratio (CR) can be increased and higher boost levels can be used. The increased full load BMEP allows downsizing of the engine at equivalent or enhanced vehicle performance.

By enabling higher CR and engine downsizing, the use of E85 DI + gasoline PFI makes the engine more efficient in its use of gasoline, thereby leveraging the effect of the available ethanol in reducing the consumption of gasoline. This leveraging has a profound influence on ethanol's net energy balance and $CO_2$ reduction potential. The vehicle owner will realize high fuel economy because gasoline, with its high heating value per volume, is primarily used for most driving modes in a downsized, high CR engine.

In this paper, the concept of E85 DI + gasoline PFI is assessed using a Ford Motor Company 3.5L turbocharged direct injection "EcoBoost" engine. A PFI system was added to the engine and CR was increased to 12:1. The amount of E85 required to avoid knock was quantified as a function of BMEP at various engine speeds on an engine dynamometer. A full load torque curve subject to the peak pressure and turbine inlet temperature constraints of the engine was also acquired. A vehicle simulation program was then used to quantify the amount of E85 required for various drive cycles, and to determine vehicle fuel consumption.

## BACKGROUND

The US Energy Independence and Security Act of 2007 mandates the use of 36 billion gallons of biofuels by 2022. This law allows for 15 billion gallons of corn-based ethanol and 16 billion gallons of cellulosic biofuel, which is likely to be primarily comprised of cellulosic ethanol. Although 31 billion gallons of ethanol would be a significant fraction of the total gasoline used in the US (about 142 billion gallons in 2007), it represents only about 14% on an energy content basis. Thus, even if the ethanol mandate of the energy bill is fulfilled, the nation's automobiles and light trucks will still rely heavily on gasoline.

There is also much debate about the net energy value of corn-based ethanol. An authoritative and thorough analysis conducted by Shapouri, Duffield, McAloon, and Wang [1], based on input data from 2001, estimates an energy output-to-input ratio of 1.67 for corn ethanol produced in the US. (That is, an amount of ethanol with 1.67 MJ of energy content requires 1 MJ of energy for production.) Continued improvement in corn farming productivity and in the efficiency of ethanol producing facilities is expected to result in higher energy ratios [2]. However, other researchers have estimated energy ratios which are less favorable, as the results are sensitive to the input assumptions and boundary conditions [3,4,5].

Further, there has been a large amount of controversy about the effect of corn-based ethanol production on land use changes [6,7], and on the availability and price of the world's food supply. A recent study on the future availability of corn for ethanol production which accounts for the trend in improving corn farming productivity [8] indicates that this criticism of corn ethanol may be unfounded. Nevertheless, balancing ethanol production with perceived effects on land use and food availability constitutes a potential argument for limiting the supply of corn ethanol.

Although production of cellulosic ethanol holds the promise of dramatically increased energy ratios [3] and reduced impact on land use changes and the food supply, it has yet to reach a large commercial scale.

With this highly political and controversial backdrop, it is apparent that the supply of ethanol is constrained, and its favorable properties should be utilized in an optimal manner to reduce petroleum consumption and $CO_2$ emissions.

## INTRODUCTION

It is well known that ethanol has a high octane but low heating value compared to gasoline. Table 1 shows fuel properties of pure ethanol (E100) and gasoline [9].

A conventional flexible fuel vehicle (FFV) is capable of running on either gasoline or E85 or a mixture of the two fuels. Some FFV calibrations adjust the spark timing with E85 to take advantage of the higher octane of ethanol. For example, for the Ford 5.4L-3V in the 2009 F-150, peak torque is increased from 365 ft lbs with 91 RON gasoline to 390 ft lbs with E85, and peak power is increased from 310 hp to 320 hp [10].

Table 1: Properties of ethanol and gasoline

| Property | Fuel | |
|---|---|---|
| | Ethanol (E100) | Gasoline |
| Octane (RON) | 107 | 91-98 |
| Heat of vaporization (kJ/kg) | 840 | ~350 |
| Stoichiometric A/F | 9.0 | ~14.6 |
| Heat of vaporization of stoichiometric fuel quantity (per mass of air) | 3.9 x base | base |
| Lower heating value (MJ/kg) | 26.9 | ~44 |
| Heating value of stoichiometric fuel quantity (per mass of air) | 0.99 x base | base |
| Density (kg/L) | 0.785 | ~0.75 |
| Heating value – volumetric basis (MJ/L) | 0.64 x base | base |

Recently, turbocharging has been applied to the basic technology of direct injection gasoline engines, giving rise to the gasoline turbocharged direct injection (GTDI) engine which enables downsizing of the engine to improve fuel economy for a particular vehicle application. For example, the 2010 Lincoln MKS will use a 3.5L V6 “EcoBoost” GTDI engine instead of a larger displacement V8. The increasing application of these turbocharged direct injection engines has provided a further opportunity to increase engine performance with E85 by adjusting boost levels and spark timing to take advantage of ethanol’s high octane. For example, the peak power of the Saab 9-5 BioPower 2.0t is increased by 30 horsepower with E85 compared to gasoline [11].

However, even in these engines, which take advantage of ethanol’s high octane to some extent, FFVs operating on E85 suffer from a dramatic reduction in range and volumetric (mpg) fuel economy due to ethanol’s low heating value. Starting at cold temperatures is also problematic due to ethanol’s boiling point of 78.5°C, high heat of vaporization, and low saturated vapor pressure at low temperatures [12,13]. To mitigate this issue, the percentage of gasoline components is typically increased to approximately 30% in winter blends of “E85” which are distributed in cold climates. Direct injection also shows promise for improving starting characteristics at cold temperatures on ethanol, but requires high injection pressure during cranking to be effective [14,15].

The use of E85 DI + gasoline PFI in principle overcomes these disadvantages of FFVs. The purpose of the work described in this paper is to assess E85 DI + gasoline PFI and quantify its benefits.

## CONCEPT DESCRIPTION

First proposed by Cohn, Bromberg, and Heywood of MIT [16], the basic premise of E85 DI + gasoline PFI is that ethanol (or another lower alcohol such as methanol) suppresses knock due to the large evaporative cooling effect it has on the air-fuel mixture when it is injected directly into the cylinder, supplemented by ethanol’s high inherent octane number [17,18,19,20]. It is widely accepted that direct injection engines benefit from an improved knock limit as a consequence of cooling of the fresh charge due to the heat of vaporization of the fuel [21]. As shown in Table 1, the heat of vaporization of ethanol is approximately four times higher than that of gasoline for a stoichiometric mixture.

The concept combining gasoline PFI with E85 DI in the same engine is illustrated in Figure 1. The concept requires separation of E85 and gasoline, which is accomplished most readily with a separate storage tank for each of the two fuels. Gasoline PFI is used for starting and low to medium load operation. The amount of directly injected E85 is increased as a function of load, but only in the amount required to prevent knock. In this way, an E85 DI + gasoline PFI engine utilizes the full octane benefit of E85 while minimizing the negative effects of low heating value and poor cold start capability.

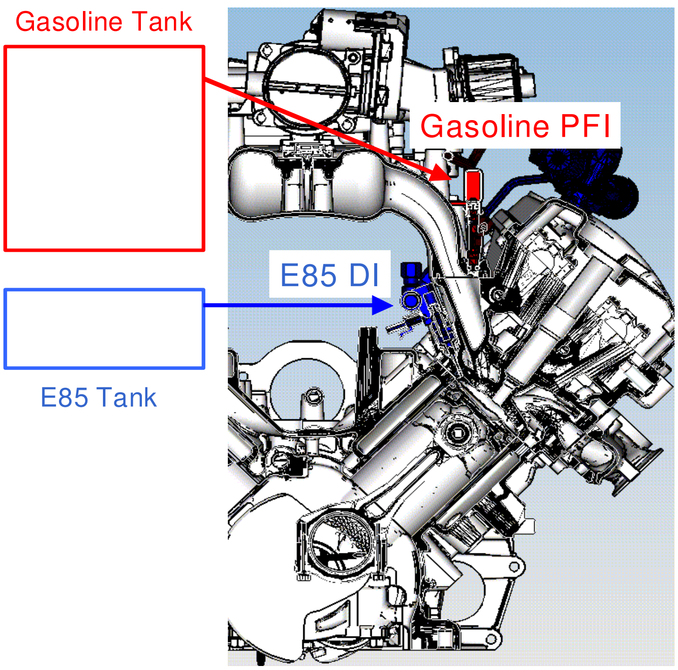


Figure 1: Cross section illustrating E85 DI + gasoline PFI

Because knock is suppressed, the compression ratio can be increased, and in a turbocharged or supercharged engine higher boost pressures can be used. The resulting higher BMEP levels allow downsizing of the engine displacement at equivalent vehicle performance. Operation at high loads with the spark timing at or close to MBT results in lower turbine inlet temperatures and consequently enrichment is reduced or eliminated. These factors result in broad regions of low BSFC in the speed/load map. In the vehicle, this enables the use of lower numerical gear or axle ratios, so that the engine operates at a lower engine speed and higher BMEP for a particular vehicle condition. This is referred to as downspeeding. Both downsizing and downspeeding move the operating regime of the engine to a more efficient part of the speed/load map.

The vehicle owner will realize high fuel economy because gasoline, with its greater heating value per volume, is the fuel that is primarily used for most driving modes. Furthermore, by enabling higher CR, downsizing, and downspeeding, E85 DI + gasoline PFI makes the engine more efficient in its use of gasoline. Improved engine efficiency leverages the effect of the limited supply of E85, compared to simply displacing gasoline as in an FFV. As will be shown in a later section of this paper, this leveraging can be very substantial, and has the effect of dramatically improving the net energy balance of ethanol, and therefore its beneficial impact on reducing petroleum consumption.

## CONCEPT ASSESSMENT

ENGINE DYNAMOMETER DATA - As a preliminary step to quantify the knock benefit of directly injected E85 on a modern turbocharged engine, an early prototype 3.5L EcoBoost engine at 9.8:1 CR was used to compare the knock-limited combustion phasing of E85 to that of 98 RON gasoline. A BMEP sweep was performed at 2500 rpm with the spark timing set to borderline knock or MBT, or as limited by the peak pressure constraint of the engine. As shown in Figure 2, with 98 RON fuel the spark timing must be progressively retarded as BMEP is increased above about 8 bar. With E85, the spark can be held at MBT until the peak pressure limit of the engine structure is reached. As expected, maintaining combustion phasing at MBT results in much lower turbine inlet temperatures compared to the retarded combustion phasing with 98 RON fuel. This can have a significant effect on the amount of fuel enrichment required and/or the cost of materials for the turbine.

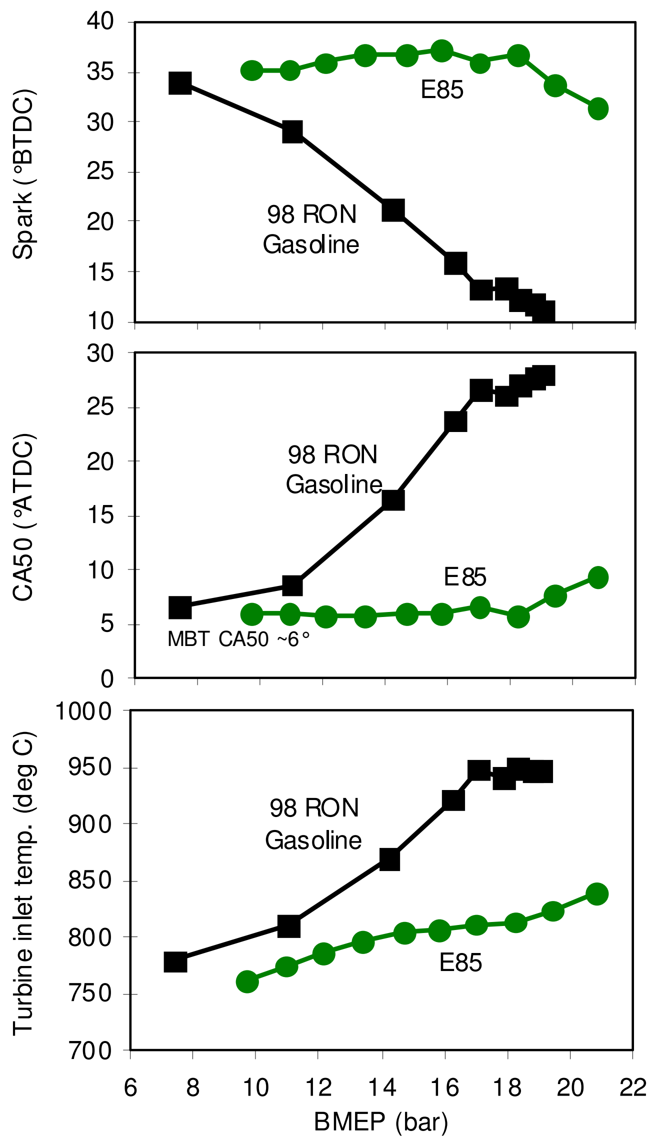


Figure 2: E85 vs. gasoline BMEP sweeps at 2500 rpm

Following this initial data which substantiated the improvement in knock with direct injection of E85, an early prototype 3.5L EcoBoost engine was outfitted with PFI, and CR was increased to 12:1 by using modified pistons. The PFI system was supplied with 91 RON gasoline, and the DI system was supplied with E85. Properties of the fuels used are shown in Table 2.

Table 2: Selected properties of E85 and 91 RON fuels

| Property | Fuel | |
|---|---|---|
| | E85 | Gasoline |
| % Ethanol | 80.8 | 0 |
| Net heating value (MJ/kg) | 28.84 | 43.36 |
| Specific gravity @ 60°F | 0.78 | 0.74 |
| H:C Ratio | 2.77 | 1.88 |
| O:C Ratio | 0.375 | 0 |
| Stoichiometric A/F ratio | 10.0 | 14.62 |
| RON | N/A* | 91 |
| MON | N/A* | 83 |
| Vapor Pressure (kPa) | 38-59 | 48-56 |

*Fuel property not available.

BMEP sweeps were performed at various engine speeds to determine the amount of directly injected E85 required to avoid knock. Results from this testing at 2500 rpm are shown in Figure 3. In this paper, the amount of E85 is expressed as a percentage of the total mass of fuel flow, since fuel flow was measured on a mass basis in the engine dynamometer test cell.

For this early prototype EcoBoost engine, the peak pressure (mean + 3 sigma variation) was constrained to 100 bar due to engine structural limitations. As shown in Figure 3, the peak pressure limit was reached at 15 bar BMEP, requiring spark retard to control peak pressure at higher BMEP.

The spark retard results in later combustion phasing and higher turbine inlet temperature. To protect the prototype hardware, turbine inlet temperature was limited to 900°C, which is relatively conservative compared to the state-of-the-art for spark ignition engine turbochargers. The 900°C limit required enrichment with additional gasoline above 20 bar BMEP.

As shown in the bottom graph of Figure 3, the amount of E85 required to avoid knock increased with load up to 15 bar BMEP. Spark timing was limited by the peak pressure above 15 bar BMEP, and hence the required amount of E85 started to decrease with load.

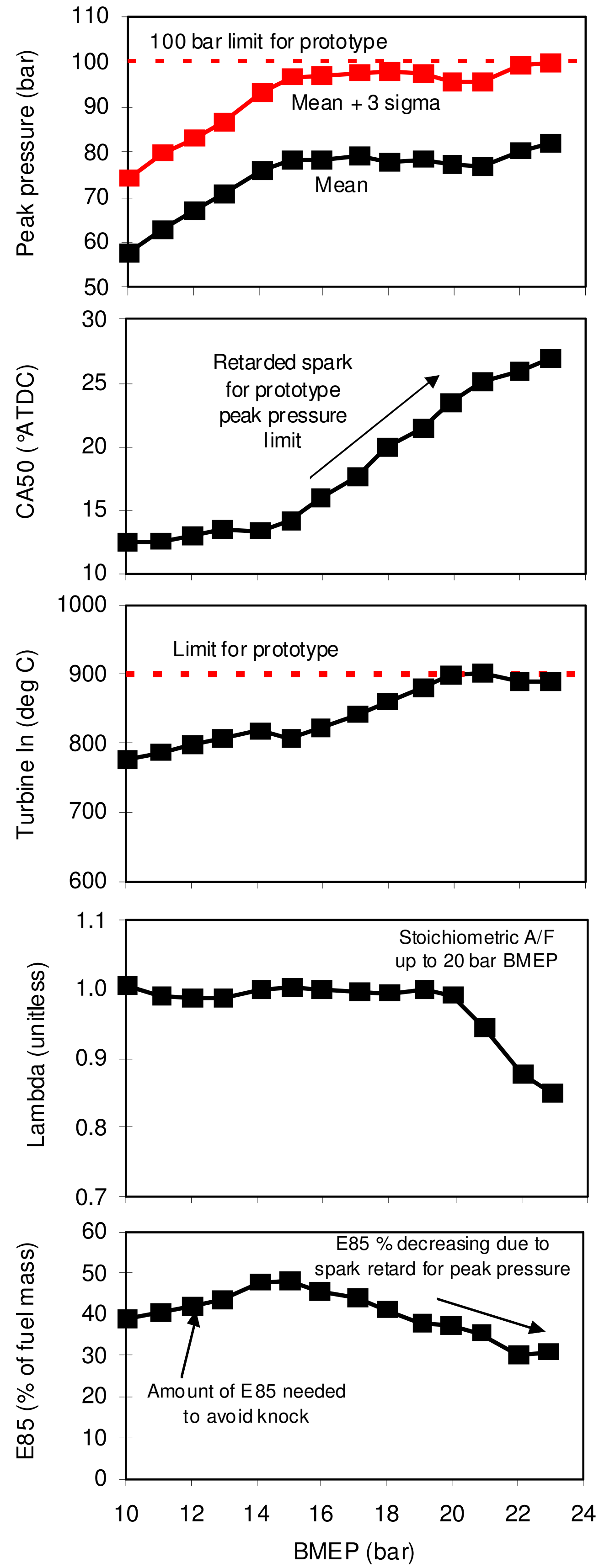


Figure 3: BMEP sweep at 2500 rpm

EFFECT OF MODERATE SPARK RETARD ON E85 CONSUMPTION - Spark retard from MBT significantly reduces the amount of E85 required to avoid knock, as shown in Figure 4. Note that small amounts of spark retard have a very limited effect on thermal efficiency but a large effect on the peak unburned gas temperatures which govern knock kinetics. As spark is retarded and the amount of E85 is reduced, the amount of gasoline must be increased, and the brake thermal efficiency of the engine degrades due to non-optimal combustion phasing. However, because of E85's low heating value compared to gasoline, the combined mass quantity of fuel initially decreases as the spark is retarded. This results in a minimum combined BSFC at moderate spark retard.

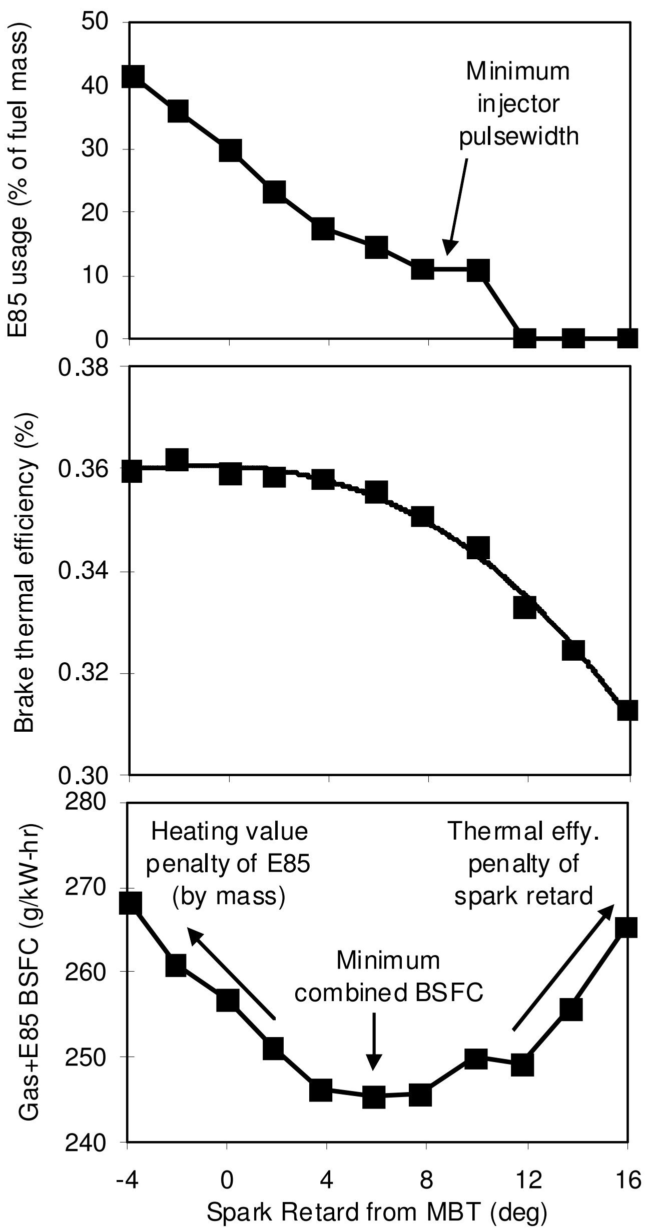


Figure 4: Spark sweep at 2000 rpm, 8 bar BMEP

The difference between MBT and the spark retard for minimum combined BSFC illustrates an opportunity for optimizing an E85 DI + gasoline PFI engine. The control strategy could be optimized for best thermal efficiency (by heating value) or minimum BSFC (by mass) or maximum MPG (by volume). In an ideal case, optimization for the vehicle owner would account for the relative cost and availability of gasoline and E85, which could vary with time and geographic location.

E85 CONSUMPTION - The amount of E85 required for various drive cycles was estimated for a hypothetical 5.0L E85 DI + gasoline PFI engine in a Ford F-series pickup truck. The 3.5L data described earlier was assumed to be representative of the percentage of E85 required to avoid knock in the 5.0L engine as a function of BMEP at 12:1 CR. The data for all engine speeds was fitted to a single line vs. engine BMEP, as shown in Figure 5. The data indicated that no E85 is required below 6 bar BMEP. Above 6 bar, E85 requirement to avoid knock increased with BMEP, to about 40% of total fuel mass at 10 bar BMEP, and about 65% at 18 bar BMEP (at MBT spark). Note that it was necessary to extrapolate the E85 requirement at MBT spark above 15 bar BMEP, due to the prototype engine peak pressure constraint.

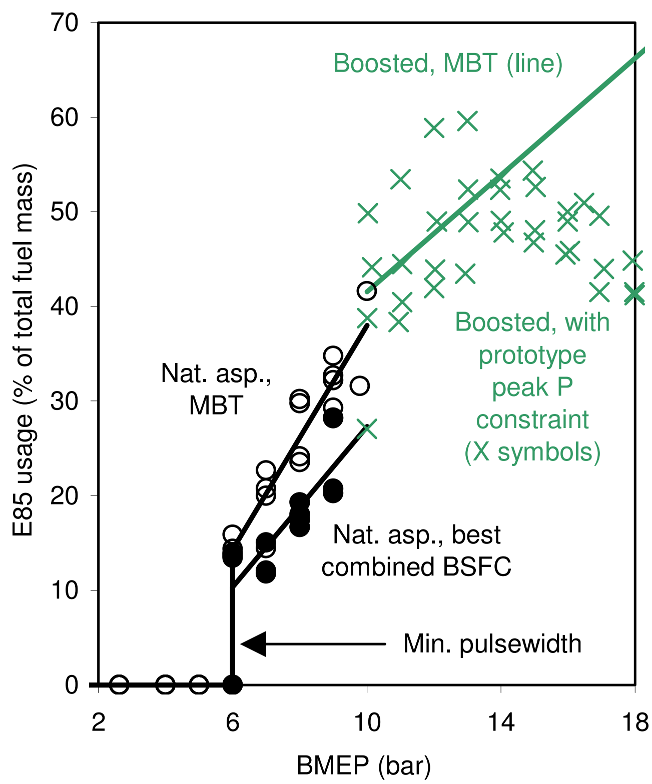


Figure 5: Percent E85 (by mass) required to avoid knock vs. BMEP at 12:1 CR for various engine speeds

This data was then used as input to Ford's Corporate Vehicle Simulation Program (CVSP) to estimate the amount of E85 required for various drive cycles. The results of this drive cycle modeling are shown in Table 3.

Table 3: Modeled E85 consumption as a percent of total fuel mass for a 5.0L E85 DI + gasoline PFI engine in a pickup truck

| | at ETW | at GCWR |
|---|---|---|
| EPA City | 1% | 19% |
| EPA Highway | 1% | 30% |
| US06 | 16% | - |
| Davis Dam | - | 48% |

An unloaded vehicle on a mild drive cycle like the EPA Metro/Highway (M/H) required very small amounts of E85 because the engine rarely operated above 6 bar BMEP. More aggressive drive cycles required more E85 to maintain MBT, for example, US06 required about 16% of total fuel mass to be E85. In extreme conditions, like towing a fully loaded trailer up Davis Dam (~6% grade for over 10 miles), about half the fuel must be E85 to maintain MBT.

However, it is important to note that less E85 can be used by moderately retarding the spark timing (as described above), with a small effect on efficiency. The transmission shift strategy can also be modified to avoid operation at low speeds and high BMEP, which will decrease E85 consumption at the expense of overall thermal efficiency.

VEHICLE RANGE - E85 DI + gasoline PFI requires two fuel tanks *and* vehicle owner acceptance of dual fueling. Thus, the maximum range of the vehicle before re-fueling with E85 could be very important, depending on local E85 cost, availability, and refueling convenience. For example, a single co-fueling nozzle which dispenses gasoline and E85 simultaneously would be more convenient than a separate E85 pump. Similarly, an expanded E85 infrastructure would relieve concerns related to availability.

Range depends on E85 consumption (as discussed earlier), and on E85 tank size. Range was estimated for a Ford F-series pickup truck with a 10 gallon E85 tank and a 26 gallon gasoline tank, compared to a baseline truck with a 26 gallon gasoline tank (2009 F-Series trucks are available with either a 26 or 36 gallon gasoline tank). Range was estimated for a hypothetical 5.0L E85 DI + gasoline PFI vehicle compared to a baseline 5.4L gasoline vehicle, as shown in Figure 6.

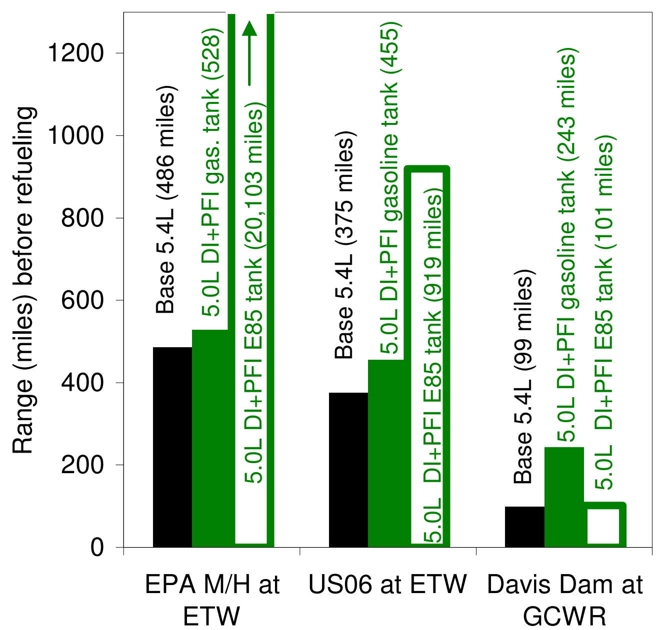


Figure 6: Vehicle range for various drive cycles

The E85 DI + gasoline PFI vehicle had higher compression ratio, lower engine displacement, and equal gasoline tank size, so it had a longer range before gasoline refueling was required.

An unloaded vehicle on a mild drive cycle like EPA M/H uses so little E85 that refueling with E85 might only be required about once a year (20,000 miles). An aggressive drive cycle like US06 might require refueling with E85 at every other gasoline fill-up. In extreme conditions like towing a fully loaded trailer up Davis Dam, the E85 tank must be refueled more often than the gasoline tank, but E85 range is about the same as the baseline vehicle with the 26 gallon gasoline tank.

It is important to note that these range estimates are based on MBT spark, and (as described earlier) less E85 can be used by modifying the spark timing and/or transmission shift strategy calibrations. Of course, range also depends on the fuel tank sizes and on the degrees of engine downsizing and downspeeding.

## FULL LOAD PERFORMANCE

Full load performance for E85 DI + gasoline PFI was assessed on a 3.5L EcoBoost engine at 9.8:1 CR. For this testing, the mean + 3 sigma peak cylinder pressure constraint was increased to 125 bar. This peak pressure limit was encountered at 2000 rpm and above. The turbine inlet temperature constraint for this testing was increased to 950°C, to be more representative of the state-of-the-art. To provide a conservative safety margin for knock, the percentage of DI E85 was increased above the minimum required to avoid knock. No issues were noted with pre-ignition during this testing. The results are shown in Figures 7 and 8.

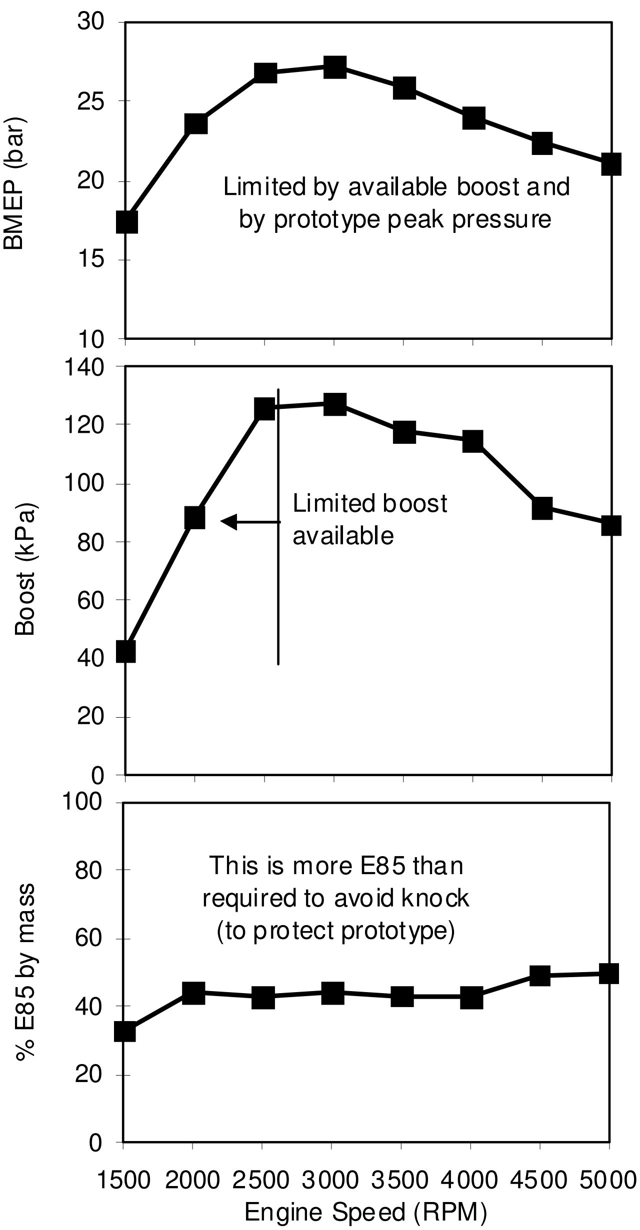


Figure 7: Full load performance for 3.5L E85 DI + gasoline PFI engine at 9.8:1 CR

As shown in Figure 7, a maximum BMEP of 27 bar was achieved at 2500 - 3000 rpm. At 2500 rpm and below, torque was limited by the available boost provided by the prototype turbochargers.

A cylinder pressure P-V diagram for the data at 2000 rpm is shown in Figure 8. This diagram illustrates the positive pumping work obtained when the wastegate is closed, as is typical of turbocharged engines at low speed.

From these results it is apparent that an optimized E85 DI + gasoline PFI engine with improved structure to enable high peak cylinder pressures will be capable of high BMEP levels, which will enable further engine downsizing and downspeeding while maintaining equivalent or enhanced vehicle performance.

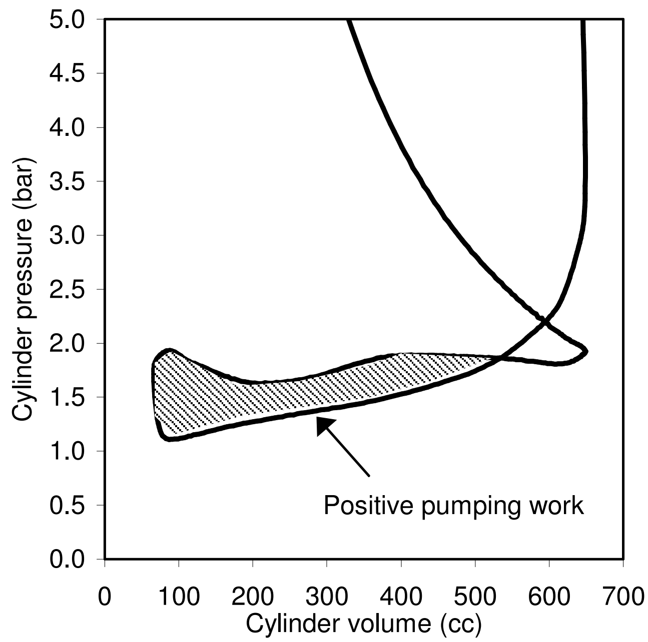


Figure 8: Pressure-volume diagram at 2000 rpm, 24 bar BMEP

## LEVERAGING EFFECT

As stated previously, E85 DI + gasoline PFI leverages the effect of the available ethanol in reducing gasoline consumption, because it makes the engine more efficient in its use of gasoline due to higher CR, downsizing, and downspeeding.

To illustrate the leveraging effect of only the increased CR component, it was assumed that a Ford F-series pickup truck was driven 1000 miles on the EPA M/H cycle. The amount of fuel consumed was calculated for a 5.0L GTDI FFV engine at 9.8:1 CR and for a 5.0L E85 DI + gasoline PFI engine at 12:1 CR. For the 5.0L GTDI engine with 20 mpg fuel economy, 50 gallons of gasoline are used. For the 5.0L E85 DI + gasoline PFI engine, 0.5 gallon of E85 and 47.5 gallons of gasoline are used, as shown in Figure 9. (From the previous section of this paper on E85 consumption, it was noted that E85 consumption on the EPA M/H cycle was 1% of the total fuel mass.)

Hence, while the E85 DI + gasoline PFI engine uses 0.5 gallons of E85, gasoline consumption is reduced by 2.5 gallons relative to the baseline 5.0L GTDI engine. This is equivalent to 0.5 gallons of E85 replacing 2.5 gallons of gasoline, which is a leveraging of 5:1 (2.5/0.5), as illustrated in Figure 9.

This is in stark contrast to the 5.0L GTDI FFV where E85 is used for the entire 1000 miles, and 72.5 gallons of E85 are used. In this case, 72.5 gallons of E85 replace 50 gallons of gasoline. The displacement of gasoline in this case is only 0.7:1 (50/72.5) because of E85's low volumetric heating value compared to gasoline.

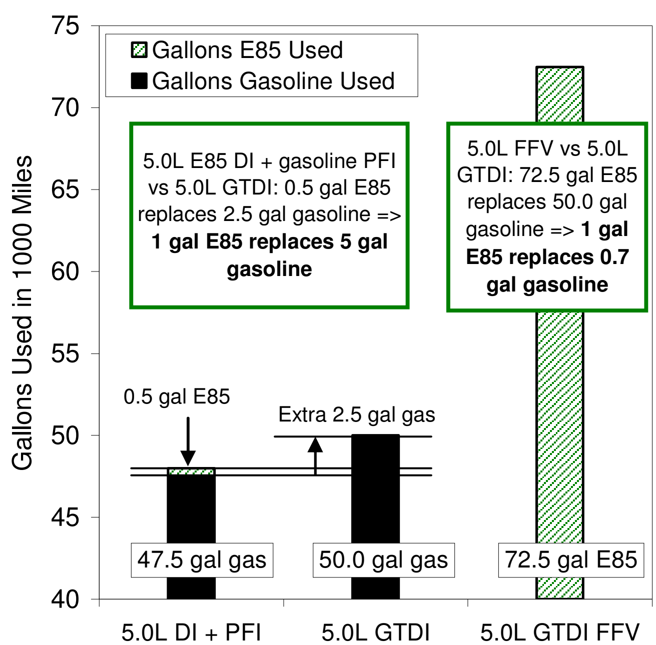


Figure 9: Leveraging of E85 in reducing gasoline consumption for 5.0L engine in F-series pickup

This leveraging effect will be significantly reduced for more aggressive drive cycles, where the amount of E85 required to avoid knock is greater due to operation at higher BMEP. The leveraging also depends on the amount of engine downsizing, compression ratio, and level of spark retard. Unfortunately, the existing data was constrained by prototype peak pressure at higher BMEP and thus is insufficient for detailed analysis on more aggressive drive cycles. Nevertheless, because the overall engine efficiency is improved, the leveraging will still be significantly better than for a conventional FFV.

As mentioned previously, the leveraging occurs because E85's effectiveness in suppressing knock allows the engine to operate with increased thermal efficiency. The leveraging can provide a substantial increase in the effective net energy value of ethanol. In this case, the output energy value is not based on the energy content of ethanol, but rather on the energy content of the gasoline which it has replaced in the vehicle.

For the example of an F-series pickup on the EPA M/H cycle, 0.5 gallons of ethanol replaced 2.5 gallons of gasoline. Using the fuel properties of Table 2 for E85 and gasoline and assuming an energy output-to-input value for corn-based ethanol of 1.67 [1], an effective energy output-to-input ratio of approximately 14:1 is obtained when the leveraging effect is included (see Appendix). Assuming an energy output-to-input ratio of 6 for cellulosic ethanol [3] results in an effective energy ratio of approximately 50:1.

These effective net energy values will be much less favorable for more aggressive drive cycles, but still significantly better than 1.67 and 6. Note that these estimates of leveraging and net energy value only account for the CR benefit of E85 DI + gasoline PFI, and do not include the effects of engine downsizing and downspeeding.

Effective energy ratios of 14:1 for corn-based ethanol and 50:1 for cellulosic ethanol equate to savings of 14 MJ and 50 MJ of gasoline energy for every 1 MJ of energy used to produce the ethanol. It is apparent from these energy ratios that high volume adoption of E85 DI + gasoline PFI would have a large-scale impact on the petroleum consumption, $CO_2$ emissions, and energy security of the nation. This impact would be significantly greater than that obtained by simply blending the available ethanol in gasoline at low percentages (such as E10 or E20), or by using it as E85 in conventional FFVs. However, although adoption of E85 DI + gasoline PFI in high volume would be facilitated by the widespread availability of E85, it would still require the acceptance by vehicle owners of refueling two separate fuel tanks.

## TECHNICAL CHALLENGES

There are a number of technical challenges associated with an E85 DI + gasoline PFI engine. These include high peak cylinder pressures, combustion noise, and direct injector cooling.

PEAK PRESSURE REQUIREMENT - An estimate of the required peak pressure capability of the engine structure was made based on an extrapolation of the BMEP sweep at 12:1 CR at 2500 rpm. Extrapolation of this data to 27 bar BMEP indicates that a peak pressure capability of 150 bar is required, as shown in Figure 10. This extrapolation also indicates that stoichiometric operation should be feasible up to 27 bar BMEP with 950°C turbine inlet temperature and 150 bar peak pressure limits. A peak pressure of 150 bar is at the lower end of the range of modern diesel engines, and hence an engine structure similar to a diesel will be required for an optimized E85 DI + gasoline PFI engine.

COMBUSTION NOISE - Combustion noise is expected to be a major concern for an engine with high CR, high boost levels, and spark timing close to MBT. To some extent, this concern is mitigated by the rigidity of an engine structure which is capable of high peak pressures. It may also be necessary to use moderate spark retard to reduce the rate of pressure rise, and/or to take other NVH mitigation actions.

INJECTOR COOLING - Direct injectors rely upon fuel flow for a major fraction of the cooling that keeps the injector temperature within design limits. In the E85 DI + gasoline PFI engine, the direct injectors are not used at light loads because the engine is not knock-limited on gasoline. Therefore the direct injectors are not cooled by fuel flow and the injector temperature is a concern at these conditions. To assess the severity of this concern, an engine with both PFI and DI was run on an engine dynamometer at speed-load conditions where fuel may not be flowing through the DI system in an E85 DI + gasoline PFI engine, and injector tip temperatures were measured.

As shown in Figure 11, tip temperatures were approximately 105-110°C whenever fuel was flowing through the DI system. Reducing the fuel flow through the DI system to 60% of the total fuel flow only increased tip temperature slightly. When no fuel was flowing through the DI system, tip temperature increased with speed and load up to a maximum of approximately 175°C, which is comparable to maximum injector tip temperatures of gasoline DI engines [22,23]. This data indicates that injector tip temperature is an important design factor for E85 DI + gasoline PFI engines, but it does not appear to preclude feasibility of E85 DI + gasoline PFI. Note that fuel could be injected with the DI injectors specifically to control injector temperature and/or to avoid injector deposits or coking, even when DI E85 is not required to avoid knock.

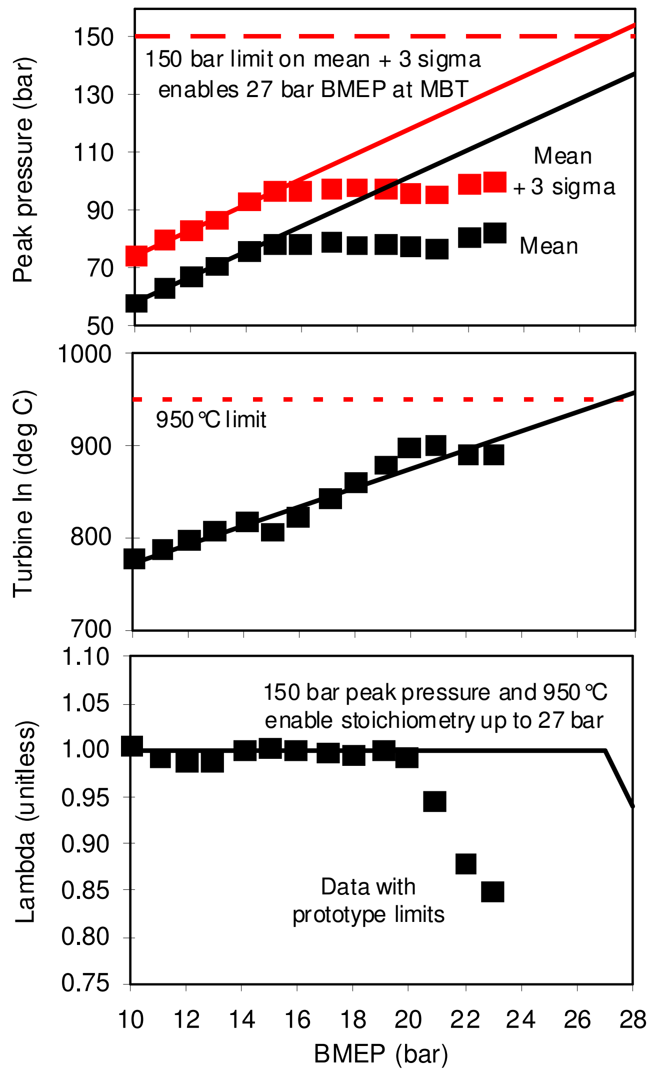


Figure 10: Extrapolation to 27 bar BMEP at MBT spark timing at 2500 rpm

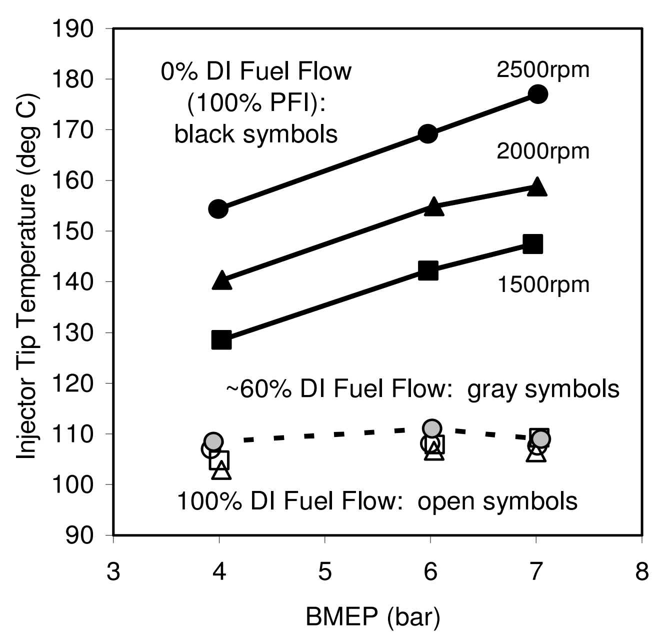


Figure 11: Injector tip temperature vs. rpm and BMEP

## ADDITIONAL CONSIDERATIONS

As well as the aforementioned technical and vehicle owner acceptance challenges, E85 DI + gasoline PFI is expected to provide a reduction in the severity of a number of implementation challenges inherent to a conventional GTDI FFV. These considerations are briefly listed here only for the sake of completeness, and the extent of the benefit of E85 DI + gasoline PFI on mitigating these challenges has not been studied or quantified.

VALVE SEAT RECESSION - Valve seat recession is a common concern with the use of E85 in FFVs, and can require the use of upgraded valve seat materials, especially with mechanical lash valvetrains. With an E85 DI + gasoline PFI engine, gasoline is used at light to medium loads, and a combination of gasoline and E85 is used only at higher loads. This would be expected to reduce valve seat recession.

BORE WASH - With a conventional GTDI FFV, bore wash is a major concern due to the use of E85 for cold starting, and due to the high volume flow rate of E85 at high loads. In an E85 DI + gasoline PFI engine, the issue of bore wash with E85 is reduced because gasoline PFI is used for cold start, and because less E85 is used at high load conditions.

INTAKE VALVE AND PORT DEPOSITS - With a direct injection FFV, minimizing intake valve and port deposits can be a challenge, and requires careful attention to the interaction of PCV and exhaust residual. With an E85 DI + gasoline PFI engine, intake valve and port deposits should be prevented because of the cleansing effect of gasoline PFI.

DYNAMIC RANGE OF DIRECT INJECTION SYSTEM - With a conventional GTDI FFV, an injection system with high dynamic range is required to cover idle through peak power. Because only gasoline PFI is used at light loads in an E85 DI + gasoline PFI engine, the required DI pump and injector dynamic range is reduced. In addition, at peak power, the E85 DI + gasoline PFI engine uses a combination of both PFI and DI to achieve the necessary fuel flow, which also reduces the DI pump and injector requirements.

COLD START EMISSIONS/DRIVEABILITY - In a conventional FFV, cold start is a major challenge with E85, and excessive enrichment is required for starting and driveability which results in high HC emissions [12]. An E85 DI + gasoline PFI engine uses gasoline PFI for cold starting.

## COMPARISON TO MODERN DIESEL

In some regards, an E85 DI + gasoline PFI engine is similar to a modern diesel engine. Both engines use turbocharging, direct injection, and an engine structure capable of high peak pressures. Both engines necessitate complex controls and calibration. The diesel engine has a glow plug system; the E85 DI + gasoline PFI engine has a spark ignition system.

A second tank is required for both engines: for the diesel, a urea tank is needed for selective catalytic reduction (SCR) of NOx to achieve 2010 emission standards; for E85 DI + gasoline PFI, a second fuel tank is needed for E85.

There are, however, some major differences between the two types of engines. If the vehicle owner does not fill the E85 tank, an E85 DI + gasoline PFI engine will be much more knock-limited without direct injection of E85. By reducing boost levels and retarding spark, the engine can operate indefinitely with degraded performance using only gasoline. In comparison, a diesel SCR vehicle owner who does not refill the urea tank will experience a range of "inducements" including limited vehicle speed and eventually failure to start [24].

The DI + PFI fuel system of the E85 DI + gasoline PFI engine is less expensive than modern high pressure diesel injection systems. The E85 DI + gasoline PFI engine runs at stoichiometric air-fuel and uses a relatively inexpensive conventional three-way catalyst (TWC) system. The diesel engine uses a more complex and expensive aftertreatment system incorporating a diesel particulate filter (DPF) and urea SCR.

Because of these factors, the E85 DI + gasoline PFI engine will cost significantly less than a diesel engine, and will be able to achieve more stringent emission standards due to the extremely high conversion efficiency of a stoichiometric TWC aftertreatment system. The E85 DI + gasoline PFI engine also uses a renewable fuel in a leveraged manner to significantly reduce petroleum consumption and total net $CO_2$ emissions.

## CONCLUSION

1. By enabling increased CR, engine downsizing, and downspeeding, E85 DI + gasoline PFI makes the engine more efficient in its use of gasoline, thereby leveraging the constrained supply of ethanol in an optimal manner to reduce petroleum consumption and $CO_2$ emissions.

For a hypothetical 5.0L E85 DI + gasoline PFI engine in a Ford F-series pickup, the leveraging due to 12:1 CR is approximately 5:1 on the EPA M/H drive cycle. That is, 5 gallons of gasoline are replaced by 1 gallon of E85. This leveraging effect will be significantly reduced for more aggressive drive cycles.

2. E85 usage for a 5.0L E85 DI + gasoline PFI engine at 12:1 CR in a Ford F-series pickup is projected to be approximately 1% of the fuel for the EPA M/H cycle and 16% for the US06 aggressive driving cycle. With a 10 gallon E85 tank, this rate of E85 consumption would result in refueling intervals of approximately 20,000 miles on the M/H cycle and 900 miles on US06.

3. Moderate spark retard at loads where the engine is knock-limited on gasoline could significantly reduce E85 consumption with only a small effect on thermal efficiency.

4. A 3.5L EcoBoost GTDI engine modified for E85 DI + gasoline PFI operation and constrained by a peak pressure limit of 125 bar demonstrated 27 bar BMEP at 2500 - 3000 rpm.

5. Achieving the full potential of an E85 DI + gasoline PFI engine requires an engine structure capable of at least 150 bar mean + 3 sigma peak pressure (comparable to a modern diesel).

6. An E85 DI + gasoline PFI engine can be viewed as an alternative to a modern diesel. Both engines require a second tank: an E85 tank for E85 DI + gasoline PFI, and a urea tank for the diesel urea/SCR system in 2010. The E85 DI + gasoline PFI engine with a conventional TWC will have much lower aftertreatment cost than a diesel with DPF and SCR. In addition, the DI + PFI fuel system of the E85 DI + gasoline PFI engine is less expensive than the high pressure diesel fuel injection system.

7. An E85 DI + gasoline PFI engine is expected to have implementation advantages compared to a FFV GTDI engine operating on E85. These include reduced dynamic range requirement for the DI pump and injectors, improved starting and emissions under cold temperatures, and potentially improved durability aspects (valve seat wear, bore wash, intake port/valve deposits).

8. Volume production of an E85 DI + gasoline PFI engine will require vehicle owner acceptance of refueling two tanks, and convenient availability of E85.

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## DEFINITIONS, ACRONYMS, ABBREVIATIONS

A/F: Air/fuel ratio

BMEP: Brake Mean Effective Pressure

BSFC: Brake Specific Fuel Consumption

CA50: Crank angle for 50% mass fraction burned

cc: cubic centimeters

CR: Compression Ratio

DI: Direct Injection of fuel into cylinder, not intake port

DPF: Diesel Particulate Filter

EcoBoost: Ford's term for the combination of direct injection and turbocharging

EPA: U.S. Environmental Protection Agency or its fuel economy and emissions test cycle, consisting of City and Highway portions, or combined Metro/Highway (M/H)

ETW: Equivalent Test Weight used by EPA

E10: Fuel with approximately 10% ethanol and 90% gasoline

E20: Fuel with approximately 20% ethanol and 80% gasoline

E85: Fuel with approximately 85% ethanol and 15% gasoline

ft lbs: foot pounds of torque

FFV: Flexible Fuel Vehicle

g: grams

gal: gallon

GTDI: Gasoline Turbocharged Direct Injection

GCWR: Gross Combined Weight Rating, or maximum allowable weight of fully loaded vehicle and trailer

hp: horsepower

kg: kilograms

kPa: kilopascals

kW: kilowatts

L: liters

Lambda: air-fuel ratio / stoichiometric air-fuel ratio

MBT: Minimum spark advance for best torque

Mean: Average

MJ: megajoules

MON: Motor Octane Number of fuel

NOx: Nitrogen oxides (NO and NO2) emissions

PCV: Positive Crankcase Ventilation

PFI: Port Fuel Injection

RON: Research Octane Number of fuel

RPM: Engine speed (Revolutions Per Minute)

SCR: Selective Catalytic Reduction

Sigma: Standard deviation (3 sigma is three standard deviations)

TWC: Three-way catalyst

US06: Aggressive driving cycle used by EPA

## APPENDIX

Effective Net Energy Value of Leveraged Ethanol - Using the fuel properties of Table 2 for E85 and gasoline, and assuming an energy output-to-input value of corn ethanol of 1.67 [1], an effective energy output-to-input ratio can be calculated when the leveraging effect is included.

energy ratio = energy content of ethanol / energy input to produce ethanol = 1.67

So, energy input to produce ethanol = energy content of ethanol / 1.67

For the example of 2.5 gallons of gasoline replaced by 0.5 gallons of E85 (where E85 is approximately 80% ethanol from Table 2):

2.5 gal gasoline replaced by 0.5 * (.8 gal E100 + .2 gal gasoline) = (.4 gal E100 + .1 gal gasoline) => .4 gal E100 replaces 2.4 gal gasoline

Energy input to produce .4 gal E100 = energy content of 0.4 gal E100 / 1.67

Substituting the energy content of 2.4 gallons of gasoline for the energy content of .4 gal E100 to obtain an effective energy ratio:

effective energy ratio = energy content of 2.4 gal gasoline/(energy content of 0.4 gal E100/1.67)

effective energy ratio = (2.4 gal gasoline * 43.36 MJ/kg * $0.74 \times 10^3$ kg/m$^3$) / (.4 gal E100 * 28.84 MJ/kg * $0.78 \times 10^3$ kg/m$^3$ / 1.67) = 14.2

Thus, the effective energy ratio is 14.2 for a leveraging of 5:1. In this case, an effective energy ratio of 14.2 means that 14.2 MJ of gasoline energy are saved for every 1 MJ of energy used to produce ethanol.