

Optimal Use of Ethanol in Dual Fuel Applications: Effects of Engine Downsizing, Spark Retard, and Compression Ratio on Fuel Economy

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ABSTRACT

Turbocharging, increasing the compression ratio, and downsizing a spark-ignition engine are well known strategies for improving vehicle fuel economy. However, such strategies increase the likelihood of engine knock due to higher in-cylinder pressures and temperatures. A high octane fuel, such as E85, effectively suppresses knock but is not necessary in most parts of the engine operating map. To better utilize a high octane fuel, dual fuel injection has been suggested where high octane fuel is injected only when the engine is about to knock. However, the effects of downsizing, retarding spark timing, and increasing compression ratio on dual fuel applications are not well understood. To investigate these questions, GT-power simulations along with engine experiments and engine-in-vehicle simulations for a passenger vehicle and a medium-duty truck were conducted. First, engine performance maps for various boosts, compression ratios, and spark retards were created, with lines of constant fuel RON superposed. Then, parametric studies were conducted to analyze the effects of cylinder volume/boost level, spark retard, and compression ratio on the vehicle fuel consumption, ethanol usage, and the average engine efficiency. Downsizing a naturally-aspirated engine by 50% resulted in about a 30% increase in fuel economy. Ethanol consumption varied from 5 to 40% (by volume) of the total fuel used, depending on the details. Moderate amounts of spark retard reduced ethanol consumption by half while not deteriorating fuel economy by much. Increasing compression ratio above 11.5 had a marginal return in fuel economy while demanding a significantly larger amount of ethanol.

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INTRODUCTION

According to the United States' Department of Transportation, about 28% of total U.S. energy is consumed in the transportation sector, and among the transportation sector, about 62% of energy is consumed by light-duty vehicles [1, 2]. The U.S. Department of Transportation and the Environmental Protection Agency (EPA) have set stringent fuel economy and greenhouse gas (GHG) emission target for light duty vehicles up to 2025 [3]. As the regulations on fuel economy and greenhouse gas (GHG) emissions get stricter, increasing engine efficiency has become an important task for engine manufacturers. In particular, spark ignition (SI) engines comprise over 90% of the light-duty vehicles sold in the United States, so increasing SI engine efficiency is crucial for meeting the fuel economy goal [4]. New vehicles with spark ignition engines adopt many different strategies to increase efficiency such as increasing compression ratio, displaced volume downsizing, increasing boost by turbocharging, etc.

Considering the ideal gas constant-volume combustion cycle, the gross indicated efficiency of the engine is related to the compression ratio by the equation below.

$$\eta_{f,ig} = 1 - \frac{1}{r_c^{\gamma-1}} \quad (1)$$

where $\eta_{f,ig}$, r_c , γ represent the indicated gross fuel conversion efficiency, compression ratio, and the specific heat ratio respectively [5]. Based on this model explaining the fundamental relationship between compression ratio and efficiency, many researchers have done engine experiments or simulations to understand the effect of increasing compression ratio. Modern spark engines' compression ratios range from 8.1:1 to 13:1 depending on the types of technologies applied to the engine [6, 7]. Recent studies conducted at MIT by Smith, Heywood, and Cheng carefully analyzed data sets generated by other researchers along with their own results and determined the incremental efficiency gain with increasing compression ratio: on average, efficiency increased by about 5% by increasing a compression ratio from 10:1 to 14:1 with a diminishing return beyond that [8].

Downsizing along with turbocharging also increases efficiency by changing the engines' operating regions to where pumping, friction and heat transfer losses are relatively low [5, 8, 9]. Smith and Cheng have utilized a cycle simulation model to estimate the effect of downsizing on loss mechanisms. With reduction of the cylinder volume by 100cc, about 25%, 10%, and 4% decrease in relative pumping, friction, and heat transfer losses are estimated in part-load operation [10]. Engine efficiency increases as the load increases, but there is a diminishing return, and this is more apparent for a turbocharged engine. According to the present authors' previous

work, from part load to half of the maximum torque, engine brake efficiency increases to about 33% while efficiency only increases to 36% from that point to the maximum torque [11]. These features allow efficient operation of a downsized engine where the overall engine operation region shifts up to a more efficient area.

However, both downsizing and higher compression ratio increase the likelihood of engine knock due to higher in-cylinder pressures and temperatures. Knock is an atypical combustion process where premixed charge in a local region of the end gas autoignites. Strong pressure waves are caused by a sharp increase in local pressure and temperature generated by a spontaneous release of chemical energy [5].

Knock can be avoided, whether by changing operating conditions of the engine to lower the in-cylinder pressures and temperatures, or by using a higher octane fuel such as ethanol or methanol. Spark retard is an easy and widely used method to avoid knock, but efficiency drops since the spark timing is no longer at Maximum Brake Torque (MBT) timing; the efficiency drop can be 10% with ~10 Crank Angle Degree (CAD) retard from MBT timing [11, 12]. High octane fuels let the engine operate at MBT timing without spark retard at most parts of the operating range, but they are not necessary in the low-load part of the map where knock does not normally occur.

E10 and E85 are well known and commercialized ethanol-gasoline fuels. However, high octane blends (such as E85) have a lower volumetric energy content than a conventional gasoline, which results in lower fuel economy if used in flexible fuel vehicles (FFVs) [13]. Also, there is an on-going discussion about the energy output-to-input ratio of corn ethanol production and land use changes caused by corn-based ethanol production [14, 15, 16]. While there are some promising aspects on the use of ethanol at large scale (improvements of corn ethanol production efficiency and production of cellulosic ethanol), the use of ethanol has to be optimized for now in the context of limited supply.

To resolve this issue, a dual fuel approach has been suggested where sufficiently high-octane fuel in the minimum amount is separately injected when it is required to suppress knock. It is well known that E85 suppresses knock well due to its inherently high octane number and its evaporative cooling effects when used with direct injection (DI) [17, 18]. Stein, House, and Leone and others from Ford Motor Company have found that such dual fuel use makes the engine more efficient by enabling increased CR, engine downsizing, and downspeeding [13].

Displaced volume change with turbo-downsizing, spark retard strategies, compression ratio increase, and vehicle types are interesting variables for analyzing performance of dual fuel applications. However, the effects of those variables on engine efficiency, drive cycle fuel economy, and ethanol consumption in dual fuel applications are not well quantified. Also, it is not easy to do a complete parametric studies as changing engine geometries (compression ratio, displacement volume) and loading them on a vehicle for dyno tests takes tremendous amounts of time and other

resources. Still, it is important to estimate the ethanol consumption required for knock avoidance in many different scenarios before actually producing engines for dual fuel applications.

Ethanol consumption in dual fuel applications is a function of many variables: engine speed and load, driving aggressiveness, acceleration, spark retard, boost level, compression ratio, etc. However, the actual quantification of the ethanol requirement in a dual fuel application for different engine scenarios has not yet been done. In this paper, engines with various downsizing/boost levels, spark retard strategies, and compression ratios and vehicle type are considered. Then those engines are loaded on the engine-in-vehicle simulation tool Autonomie to estimate the average engine efficiency, fuel economy, and ethanol consumption for standard U.S. driving cycles. By changing 5 variables (4 downsized engine volumes, 3 compression ratios, 3 spark retard scenarios, 3 driving cycles, and 2 vehicle types), total of 216 cases were compared.

METHODOLOGY

To analyze the effect of downsizing, retarding spark timings, and increasing compression ratio on dual fuel applications, performance maps were generated. Both engine experiments and GT-Power simulations were utilized to generate performance maps of different engine scenarios and to superimpose knock limits on the maps.

The generated maps include efficiency contours and knock limits: for a given engine speed and load (or torque), efficiency and the amount of ethanol required to suppress knock are known. Then, generated performance maps were simulated in the engine-in-vehicle simulation tool called Autonomie. By setting up the vehicle models and using various driving cycles, it was possible to calculate fuel-economy, average efficiency, and ethanol fraction required to complete a specific cycle.

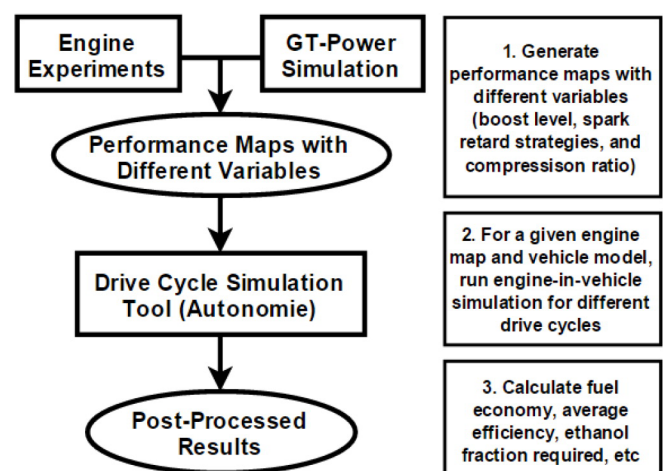


Figure 1. Schematic diagram explaining how different tools (engine experiments, GT-Power simulation, engine-in-vehicle simulation) were utilized to complete the intended parametric studies.

Engine Experiments and GT-Power Simulation

The engine used for experiments was a General Motors 2.0L turbocharged engine. [Table 1](#) shows the specifications of the engine.

Table 1. Engine specifications.

Displaced Volume	1998 cc
Cylinders	In-line 4 Cylinders
Stroke	86 mm
Bore	86 mm
Compression Ratio	9.2 : 1
Maximum Torque	353 Nm (2500 rpm ~ 5000 rpm)
Maximum Power	194 kW (at 5300 rpm)
Injection & Ignition	Direct Injected Spark Ignition
Valve Timing	Dual Independent Cam Phasing

The engine experiments were first conducted to generate the brake efficiency contours on the engine's performance map and superimpose knock limits on the efficiency contours. About 30 steady state operating points were selected to cover the frequently driven region on the engine map. For a set of chosen operating points a complete set of spark sweeps (-5 CAD to +15 CAD retard from MBT timing) were conducted, and brake efficiencies of the engine were measured at MBT timing. Then, fuel blends were used to establish knock limits of various ethanol blends (RON 91 E0, RON 96 E0, E10, E20, E25, and E85) over the range of speeds and loads at a compression ratio of 9.2:1. For a given fuel, 24 spark sweeps were done on average to generate knock limits.

In order to simulate the performance of the engine beyond the limits of our engine experimental setup, GT-Power simulations were conducted. After benchmarking the model to the experimental results, it was possible to generate performance maps of the engine for different boost levels, compression ratios, and spark retard strategies. By comparing the brake fuel conversion efficiency estimated from GT-Power to that from engine experiments, it was possible to confirm the fidelity of the engine model [11].

Utilizing the GT-Power model, various performance maps were generated by changing engine geometries and boost conditions. [Table 2](#) shows the variables changed in the GT-Power simulation. Four different boost scenarios (from no boost (naturally-aspirated) to maximum boost), three different spark retard strategies, and three different compression ratios gave 36 performance maps. The lowest engine displaced volume case represents an aggressive downsizing, which involves pragmatic concerns due to the transient response of a turbocharger, potentially limiting vehicle performance. Since such pragmatic constraints may well evolve in the future, the case is still included to represent an extreme downsizing limit.

To keep the engine performance in a vehicle the same, turbocharger's boost levels were chosen so that the corresponding engine displacement volumes would result in the same maximum torque output (230 Nm, which is the maximum torque of Camry's base

engine model). 0-60 mph acceleration times were checked for different engines, and they were comparable (around 9 seconds), assuming fast turbo response. Although there are other practical factors involved in maintaining vehicles' drivability other than peak torque, the vehicle performance metric was kept simple for our parametric study. All the maps were run in the engine-in-vehicle simulation for three different U.S. standard driving cycles, resulting in 108 driving cases for this vehicle model.

Table 2. Variables changed for parametric studies and their ranges.

Displaced Volume (Turbo Boost Level)	2.4L ~ 0.8L (NA to 1.8 bar extra boost), 4 cases
Spark Retard Strategies	0~10 CAD retards from MBT timing
Compression Ratio	9.2, 11.5, 13.5 to 1

Passenger Vehicle Simulation

The engine-in-vehicle simulation tool Autonomie was utilized to connect the operating points on the engine map for the different driving cycles. The vehicle model chosen was Toyota's Camry, as it was the most popular passenger vehicle at the United States in 2014. [19] Later in this paper the implications of a medium duty truck will be presented. Different engine models were loaded on the vehicle model to study the effects of various parameters on a vehicle performance and ethanol consumption. [Table 3](#) shows Toyota Camry's important details: vehicle weight, fuel economy, and engine specifications. As previously mentioned, maximum torque was kept the same for all the performance maps so the vehicle has a similar acceleration capacity. Even though the maximum power varied slightly, all the performance maps tested showed closely comparable performance in terms of 0-60 mph acceleration times. Also, they were all able to follow the required speed profile of the driving cycles tested (none showed particular over-speed or under-speed relative to the given speed profile of the driving cycle.)

Table 3. Vehicle model and engine specifications [20].

Vehicle Model	2014 Toyota Camry LE
Vehicle Curb Weight	1470 kg
Fuel Economy (CTY/HWY)	25 / 35 mpg
Engine Displacement Volume and Configuration	2.4 L, in-line 4 Cylinder
Maximum Torque	230.5 Nm (at 4100 rpm)
Maximum Power	132.7 kW (at 6000 rpm)
Valve Timing	Dual Independent Cam Phasing
Injection & Ignition	Direct Injected Spark-Ignition

Three of the EPA's standard light-duty vehicle chassis dynamometer driving schedules were tested; Urban Dynamometer Driving Schedule (UDDS), Highway Fuel Economy Test Cycle (HWFET), and US06 Supplemental Federal Test Procedure (US06). UDDS represents city driving conditions, HWFET represents highway driving conditions, and US06 represents an aggressive driving style. Detailed features of the three cycles are shown below in [Table 4](#).

Table 4. Driving cycles tested [21].

Driving Cycles	UDDS	HWFET	US06
Duration (s)	1369	765	596
Distance (miles)	7.45	10.26	8.01
Average Speed (mph)	19.59	48.3	48.37
Maximum Speed (mph)	56.7	59.9	80.3
Maximum Acceleration (m/s ²)	1.48	1.43	3.78

It is still controversial whether the UDDS, HWFET, and US06 cycles give realistic estimates for the real-world fuel consumption [22, 23]. The three cycles are chosen because these are utilized the most, and their characteristics are well known.

DISPLACED VOLUME EFFECTS

The engine displaced volume is decreased while increasing a boost level to maintain constant maximum torque, resulting in similar engine performance to that of the original Camry engine. Figure 2 and 3 show the results of the engine-in-vehicle simulation. Both maps were used to run the vehicle on the UDDS cycle, but the boost levels are different; Fig. 2 shows a naturally aspirated engine map (2.5 L) while Fig. 3 shows a downsized engine map (1.1 L) with 1 bar extra boost (2 bar absolute intake manifold air pressure) generated from the turbocharger. Compression ratio was 11.5:1 for both cases. Each black point means that the engine was run at that point for 0.1 second.

To compare how the operating points shifted, the average operating point was determined. It can be seen that the average operating point in Brake Mean Effective Pressure (BMEP) is much higher in the turbo-downsized engine though the torque outputs are the same. This makes the engine operate in a more efficient area on the engine map.

The average operating point lies on the 26% brake engine efficiency for NA engine, but it lies on the 35% for the turbo-downsized engine. However, this also makes the engine more prone to knock.

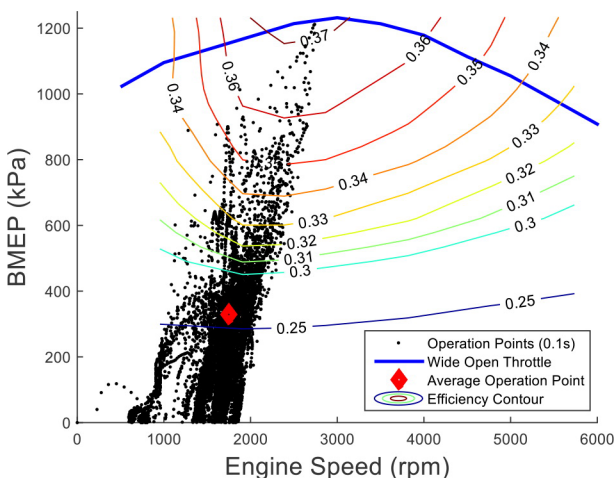


Figure 2. Operating points on the engine's performance map (2.5L NA engine with compression ratio of 11.5:1). Blue line is a wide open throttle limit, black points are operating points simulated from Autonomie over the UDDS cycle run with a passenger vehicle, a red diamond is the average operating point, and efficiency contours are in varying colors. Spark timings were kept at MBT timing.

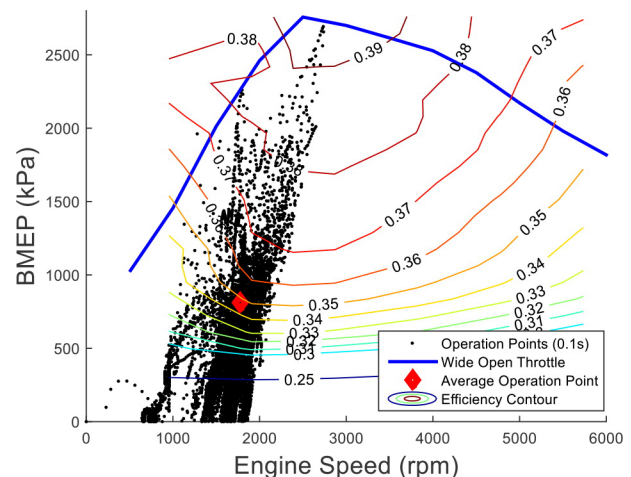


Figure 3. Operating points on the engine's performance map (1.1L turbo-downsized engine with compression ratio of 11.5:1). Blue line is wide open throttle limit, black points are operating points simulated from Autonomie over the UDDS cycle run with a passenger vehicle, red diamond is the average operating point, and efficiency contours are in varying colors. Spark timings were kept at MBT timing.

Figure 4 shows the effect of downsizing on the average brake efficiency for the UDDS cycle as the engine is operating on a more efficient region on the engine map. For each operating point, the engine brake efficiency is calculated from the equation below:

$$\text{Efficiency} = \frac{\text{Engine Power Out (W)}}{\text{Fuel Consumption Rate (kg/s)} * \text{LHV (J/kg)}} \quad (2)$$

The average engine brake efficiency was calculated for the three driving cycle. For all three cycles, downsizing the engine greatly increases the average engine brake efficiency. Downsizing the engine by 50% provides about a 30% increase in the average brake efficiency for the UDDS cycle. The US06 cycle shows a 15% increase with 50% downsizing, because in this cycle the engine is already operating in a higher efficiency region without downsizing. As a vehicle is driven in more aggressive cycle, the benefits of downsizing decrease further, and ethanol consumption increases substantially.

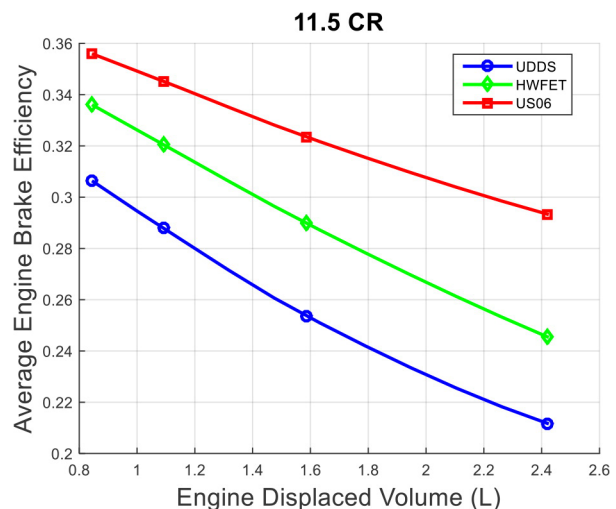


Figure 4. The average brake efficiency vs. engine displaced volume for three standard U.S. driving cycles run with a passenger vehicle. Compression ratio was at 11.5:1, and the spark timings were kept at MBT timing.

As the average engine brake efficiency for the cycle increases, fuel economy also improves. However, because of the decreased LHV of ethanol compared to gasoline, we compare the fuel economy on the basis of gasoline-equivalent fuel consumption. Figure 5 shows that 50% downsizing increases the combined fuel economy from 25 mpg to 30 mpg, which is about a 20% increase. Engine brake efficiency increase and the fuel economy increase are not exactly the same, since transmission efficiency and gear ratios affect the fuel economy. EPA has a new method for calculating the combined fuel economy that utilizes the average of 5 different cycles to include high speeds, quicker accelerations, use of auxiliaries, cold temperature driving, etc. This new method allows utilization of US06 cycle results, which is proven to have significant impacts in a real world driving. However, to simplify the variables for parametric studies, fuel economy was combined using values from the UDDS and HWFET cycles using EPA's earlier equation, below (before 2008). The cycle fuel economy values were then multiplied by the adjustment factor of 0.8 as EPA did to obtain the official vehicle label. This adjustment factor of 0.8 is used also for the other driving cycles, as discussed later in the paper.

$$MPG_{combined} = \frac{1}{\frac{0.55}{MPG_{UDDS}} + \frac{0.45}{MPG_{HWFET}}} \quad (3)$$

Downsizing results in higher efficiency and fuel economy as shown in Fig. 4 and 5, but this benefit comes with a cost of increased ethanol consumption. Figure 6 shows the effect of downsizing on the ethanol requirements, with the engine operating at MBT spark timing. UDDS and HWFET require very little ethanol when the engine is naturally aspirated. However, 50% downsizing increases the ethanol fraction to 8~10%. This is worse for US06 as the ethanol fraction increases by almost 7 times with 50% downsizing. However, ethanol fraction can be reduced with a proper use of spark retard. This will be discussed in the next section.

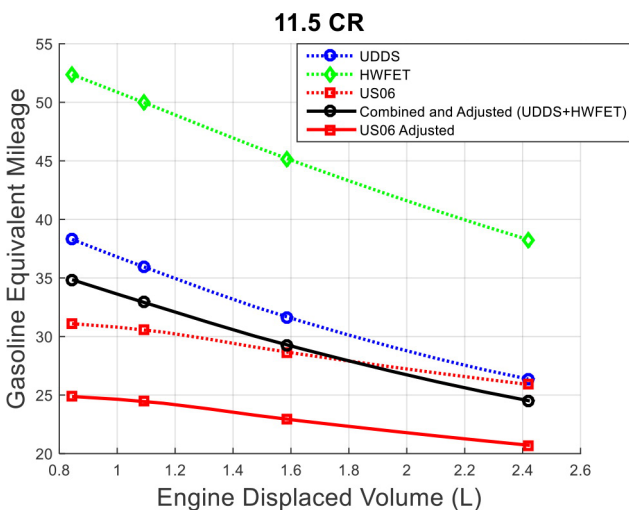


Figure 5. Gasoline equivalent fuel economy vs. engine displaced volume for three standard U.S. driving cycles run with a passenger vehicle. Dotted lines are the results from the engine-in-vehicle simulation, and solid lines are combined (UDDS and HWFET) and adjusted values. The fuel economy for the US06 cycle was just adjusted. Compression ratio was at 11.5:1, and the spark timings were kept at MBT timing.

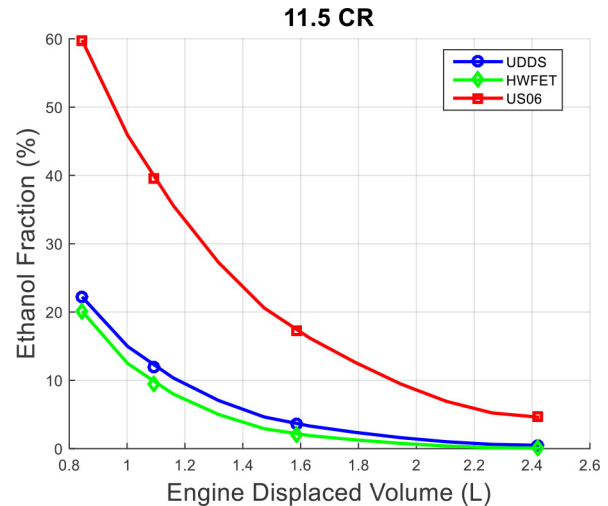


Figure 6. Ethanol fraction in percentage by volume vs. engine displaced volume for three standard U.S. driving cycles run with a passenger vehicle. Compression ratio was at 11.5:1, and the spark timings were kept at MBT timing.

SPARK RETARD EFFECTS

By retarding the spark timing from MBT timing, the knock tendency decreases due to lower in-cylinder pressures and temperatures, at the expense of reduced engine brake efficiencies. Therefore, spark retard can reduce the amount of ethanol required to suppress knock. Figure 7 shows the effect of spark retard on ethanol consumption. UDDS and US06 cycles were considered with 3 different retard scenarios. The first case is when the engine is run always at MBT timings with no spark retard. In the second case, the engine is allowed to retard the spark timing when needed up to 5 CAD from MBT timing, at the engine load where the knocking without ethanol starts. After retarding by up to 5 CAD, ethanol is then used to suppress knock, while maintaining 5 CAD spark retard. Similarly in the last retard case, the engine is allowed to retard by up to 10 CAD from the engine load where knocking starts.

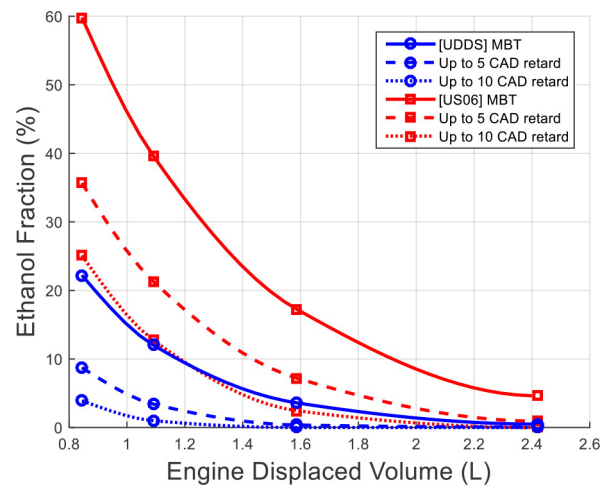


Figure 7. Ethanol fraction in percentage vs. engine displaced volume for UDDS (blue) and US06 (red) cycles run with a passenger vehicle. Dashed-lines represent the case with up to 5 CAD retard allowed, and dotted-lines represent the case with up to 10 CAD retard allowed. Compression ratio was at 11.5:1.