

Modeling the Cold Start of the Ford 3.5L V6 EcoBoost Engine

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ABSTRACT

Optimization of the engine cold start is critical for gasoline direct injection (GDI) engines to meet increasingly stringent emission regulations, since the emissions during the first 20 seconds of the cold start constitute more than 80% of the hydrocarbon (HC) emissions for the entire EPA FTP75 drive cycle. However, Direct Injection Spark Ignition (DISI) engine cold start optimization is very challenging due to the rapidly changing engine speed, cold thermal environment and low cranking fuel pressure. One approach to reduce HC emissions for DISI engines is to adopt retarded spark so that engines generate high heat fluxes for faster catalyst light-off during the cold idle. This approach typically degrades the engine combustion stability and presents additional challenges to the engine cold start.

This paper describes a CFD modeling based approach to address these challenges for the Ford 3.5L V6 EcoBoost engine cold start. A Ford in-house developed CFD code MESIM (Multi-dimensional Engine Simulation) was applied to study the effect of injection parameters and piston designs on the fuel preparation process under various engine speed and engine temperature conditions during the engine crank, run up and cold idle operations. It was found through the modeling studies that the formation of a robust fuel-air mixture distribution around the spark plug was the key to improving combustion stability and reducing the emissions. The factors that could directly affect the fuel-air mixture distribution are the injection timing, fuel pressure, and piston bowl design. Modeling results provided the physical insight of the fuel preparation mechanism of various injection strategies, and helped to optimize the split-injection strategy to improve the fuel-air mixture distribution around the spark plug. The engine test data confirmed that the optimized split injection strategies reduce HC

emission by about 30% with lower fuel consumption and substantially improved combustion stability during engine cold-start.

INTRODUCTION

Turbo-charged Gasoline Direct Injection (GDI) Spark Ignition engines have been studied extensively in recent years and are becoming a leading engine technology concept for improving fuel economy, reducing emissions while maintaining the engine performance [1,2,3,4,5]. Ford developed a family of EcoBoost engines based on Turbo-charged GDI engine technology for volume production as the near term corporate fuel economy strategy. Typically, the main features of a turbo-charged GDI engine include: homogeneous direct injection, turbo-charging with charge-air-cooling, and variable valve timing. Among these EcoBoost engines, the 3.5L V6 engine (D35 EcoBoost) is developed based on the 3.5L V6 Duratec engine architecture. The D35 EcoBoost engine achieves the same level of performance as the Ford 5.4L 3V Port Fuel Injection (PFI) engine with more than 10% fuel economy (FE) benefit due to turbo-charging, downsizing and gasoline direct injection technologies. Port, chamber and sprays are optimized during the combustion system development for these engines [6]. The combustion system also produces very low smoke emissions over the entire engine operation map, and has PZEV emissionability as a functional objective [6].

One of the technical challenges encountered during the combustion system development for the EcoBoost engine is the GDI engine cold start development. Since the engine cold-start emissions constitute more than 80% of the HC emissions for the entire EPA FTP75 drive cycle, optimizing the cold start operation to reduce emissions is critical to lower cycle cumulative emissions.

For Direct Injection Spark Ignition (DISI) engines, however, cold start optimization is particularly challenging. For a conventional homogeneous GDI engine, the primary fuel pump is mechanically driven by the cam. Without compromising the start time, the conventional start fuel pressure is around 5 bar. Compared to the Port Fuel Injection (PFI) engines, however, GDI engines prepare the fuel air mixture by injecting the fuel spray directly into the cylinder. The low cranking fuel pressure produces relatively poor fuel spray atomization. Under the low cranking speed, cold thermal environment, and poor fuel atomization, the GDI engine could produce undesired fuel preparation such as heavy wall impingement, low fuel vaporization rate, and a lean fuel-air mixture cloud around the spark plug. Heavy surface impingement also creates wall films on the liner and piston top surface, which are a major source of engine-out unburned hydrocarbon and smoke emissions. As a result, high HC emission and smoke emission can be potential issues, and a high enrichment factor is also required due to lost fuel and HC emissions.

To address the emission issue, a cold-start strategy called "High-Pressure Stratified Start" was proposed [7,8,9]. Using this strategy, fuel is injected with increased fuel pressure (>20 bar) during the compression stroke instead of the conventional start with low-pressure fuel injection during the intake stroke. Consequently, the fuel air mixture preparation is improved and wall wetting is reduced. This compression injection strategy was shown to reduce the unburned hydrocarbon emissions and enhance the start reliability compared with the conventional homogenous start.

When applied in the D35 EcoBoost engine, the compression injection strategy appears to be able to create a certain degree of engine cold start ability that meets the emission target. However, a potential emission issue is still observed and is illustrated by Figure 1. The chart shows the measured cumulative feed gas hydrocarbon (FGHC) emission by fast FID (Flame Ionization Detector) for the first 20 seconds of D35 EcoBoost engine start using the single compression injection strategy. The HC emission target region is also specified in the chart. The chart shows that even though HC emissions of the D35 EcoBoost engine fall within the current emission target region, cumulative HC emissions of the first 20 seconds have potential for significant improvement that will be needed for more stringent future emission targets. In particular, the emissions increase rapidly during the crank and run-up in first few seconds. In the meantime, misfires are observed under certain operating parameters. The fuel air mixture in the combustion chamber is relatively lean even with a large enrichment factor. Clearly, the robustness of combustion and the level of emissions during the crank and speed run-up still need more investigations. Substantial improvements are required in order to meet the more stringent emission targets that may be applicable for future applications of this engine.

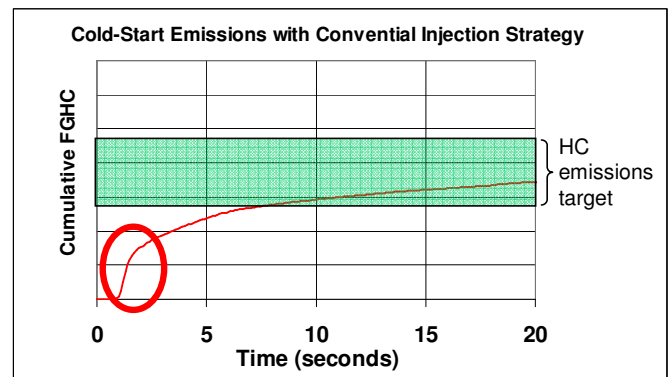


Figure 1. The cumulative FGHC emission during the first 20 seconds cold start of a D35 EcoBoost engine when using a single compression injection strategy

Another challenge encountered in the GDI cold start development is the marginal combustion stability observed in the cold idle condition (5-15s of the cold start). Under this condition, high heat flux is generated in the exhaust for the rapid light-off of the catalytic converter and the condition is therefore also referred to as the catalyst heating mode. Here, ignition delay via spark retard is typically used as the strategy to generate high heat flux for fast catalyst light-off. In principal, a GDI engine has an inherent advantage over a PFI engine, because delayed ignition is limited with homogeneous combustion in PFI engine due to deteriorated combustion stability. In contrast, for a GDI combustion system, a bowl-in-piston design is typically created to generate a stratified mixture with a rich fuel-air mixture around the spark plug. Ideally, the stratified combustion in a GDI system can enable much more ignition delay, higher exhaust heat flux and therefore accelerated catalyst light-off. The high heat flux is especially necessary in turbo-charged GDI engines, due to the heat loss through turbine prior to the catalyst. A homogeneous-split injection strategy was proposed for the catalyst heating mode [7,8,9]. With this strategy, the first injection takes place during the intake, while the second injection takes place during the compression. It was shown that for a delayed ignition time, this strategy can generate stratified local enrichment around the spark plug, and therefore improve the combustion stability and reduce the emissions compared with the single injection strategy. However, during the initial D35 EcoBoost engine combustion system development, marginal stability was observed when using a similar split injection strategy at the cold idle condition. Engine test measurements showed that the combustion stability represented by std NMEP at this condition is substantially worse than that has been measured using a conventional homogeneous charge mode with a single injection strategy. Clearly, the benefit of generating a stratified rich mixture cloud around the spark plug using the split injection strategy is not realized.

To resolve these technical challenges during the GDI engine cold start, an experimental trial-and-error approach for design improvement would be very difficult. Because the engine needs to be cooled down to a desired low-temperature condition between two

consecutive cold-start tests, this lengthy testing process makes it very difficult to obtain an optimized cold-start strategy since testing includes a large number of parameters. As a result, some of the current DISI engine products in the market often suffer from compromised functions, e.g., penalized emissions and long start times during engine cold starts.

In the development of the combustion system for the EcoBoost engines, an analytical CFD model driven upfront design and optimization process is applied to guide the design and optimizations. The same approach is adopted to resolve the technical challenges in the engine cold-start development. Specifically, Ford proprietary in-house CFD modeling tool MESIM is applied to understand the fuel-air mixture preparation mechanism during the engine cold start, and guide the development of the solution by means of optimizing injection strategies and piston top designs. The description of the combustion system port and chamber designs is given in Ref. [6]. The details of the experiments are in separate publications. This paper presents the analytical aspects of the investigation and resolutions to the two key technical challenges during a GDI engine cold start that are discussed above, namely, the HC emission reduction during engine crank and speed run-up, and the combustion stability improvement during the cold idle. The paper is organized as follows: First, the models and modeling methodology are described. Next, detailed analyses of fuel-air mixture preparation mechanisms using various injection strategies at three engine speeds during the engine crank and speed run-up are discussed. Then, the findings and solutions are proposed based on these analyses, and engine data confirmation is presented. This is followed by the model investigations and resolutions of the effect of injection timing and piston designs on the combustion stability at cold idle condition. Finally, the findings of the model investigations of the GDI cold-start are summarized. Through these model studies, it is found that the formation of a robust fuel-air mixture cloud distribution around the spark plug is the key to produce lower cycle to cycle variation and better combustion stability. By optimizing the split injection strategy at different engine operating conditions, a robust mixture cloud can be generated around the spark plug with reduced surface wetting during the entire engine crank and speed run-up transient period. As results, a good combustion stability and smooth speed run-up can be achieved with significantly reduced emissions. Similarly, for the cold idle operation, by optimizing the first injection timing of a split injection and the piston top design together, excellent combustion stability can be achieved with these EcoBoost engines.

MODELS AND METHODOLOGY

As discussed in [10,11,12,13], an upfront design and optimization process has been developed at Ford Motor Company and has been applied to develop DISI engine combustion systems. At the core of the process is the in-house developed MESIM (Multi-dimensional Engine

Simulation) CFD code. MESIM is developed based on KIVA-3V [14, 15] with an enhanced dynamic mesh moving algorithm [16]. It has improved physical sub-models including the spray atomization and vaporization models, spray wall interaction models, and wall film dynamic models [17,18,19]. A multi-hole injector spray injection and atomization model is also developed and validated with measurements from both optical bench and optical engine experiments [20].

In past CFD simulations of GDI engines, fuel is often modeled as single component (typically iso-octane). Gasoline fuel is multicomponent in nature and behaves very differently from the single component fuel representation under various operating conditions [21, 22,23]. It is found that the multi-component fuel model is critical to the accuracy of the model prediction of the fuel air preparation process, particularly under cold start conditions. A multi-component fuel model has been developed and incorporated in the MESIM code. To apply the model in engine simulations, a multi-component fuel recipe that represents the vaporization characteristic of gasoline is required. In addition, the recipe needs to be limited to a small number of fuel species for computation efficiency. This is achieved by using a numerical model that has been developed in Ford to simulate the ASTM D86 fuel distillation experimental procedure [24]. For this study, a three-component fuel recipe is used and its composition is shown in Table 1. The model predicted distillation of the three-component fuel is compared with experimental measurements of that from a gasoline fuel. As shown in Figure 2, the agreement is good in the entire range of volume fractions distilled.

Table 1. Three-component fuel recipe

Fuel Name	Formula	Mass fraction
n-pentane	c5h12	0.22
iso-octane	ic8h18	0.58
n-decane	c10h22	0.20

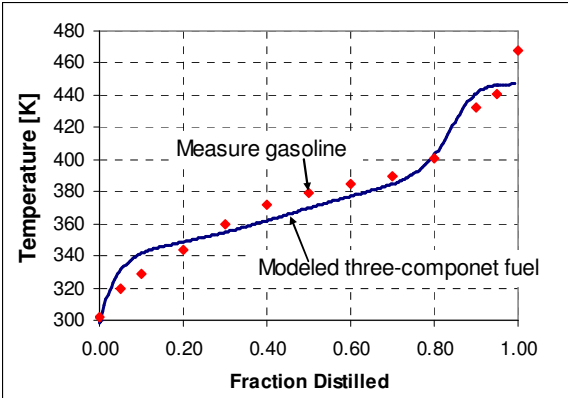


Figure 2. Comparison of distillation curves between the model simulated result of the three-component fuel and the measurement of a gasoline fuel

D35 EcoBoost is a turbo-charged four valve GDI engine with variable cam timing. The bore and stroke of the engine are 92.5mm and 86.7mm. The corresponding displacement is 583.3 cm³, and the compression ratio is 10:1. The combustion system features a 6-hole side mounted injector. More details of the port, chamber design and injector sprays are described in [6]. A shallow bowl-in-piston design is also developed for the cold start operation, and will be shown in the later part of this paper. The simulation domain of the combustion system including the runner, intake and exhaust ports, combustion chamber and piston top is shown in Figure 3.



Figure 3. The simulation domain of a D35 EcoBoost engine

Initial and boundary conditions of CFD simulations are set up based on engine test data and measured temperatures. Simulations start at 60° BTDC during the exhaust stroke, continue through the intake stroke and the compression stroke and end at 40° ATDC in the expansion stroke. Simulations shown in this paper are for a 20°C cold start.

The engine cold start begins with the crank and rapid engine speed run-up, followed by relatively steady state cold idle. Figure 4 shows a chart of engine RPM and intake manifold pressure vs. injection event for a 6-cylinder GDI engine. As illustrated by the chart, the engine speed increases from 200 RPM to about 1200 RRPM within the first six injection events, while the intake manifold pressure decreases from around 1 bar to below 0.9 bar. In order to investigate the fuel-air mixture preparation mechanism and capture the effect of the varying dynamics during this transient period, CFD simulations are conducted at several representative engine conditions. Three engine speeds 200, 500 and 1200 RPM are selected to represent conditions of low, mid and high speed range during the first six injection events, while 1200 RPM/2bar NMEP is used for the simulations at the cold idle condition.

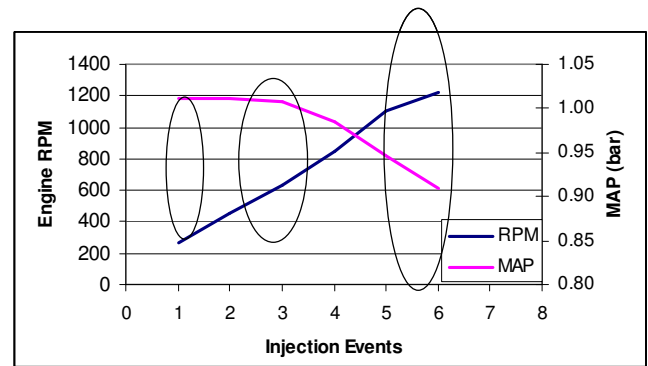
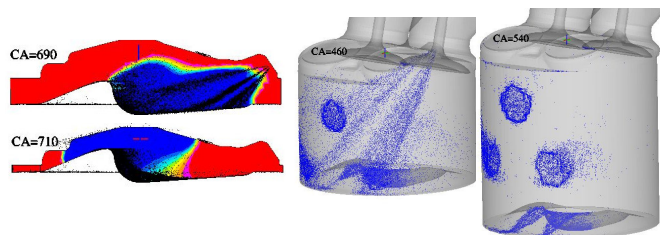


Figure 4. Engine speed and intake manifold pressure vs. injection events in a 6-cylinder GDI engine

Besides the engine operating conditions, the design factors that could influence the fuel-air mixture preparation during the cold start include fuel injection strategy, fuel pressure, combustion system hardware designs (port, chamber, piston and injector spray), cam strategy, etc. Among these factors, injection strategy and piston top design are found to have the most significant effects on the mixture formation around the spark plug, and therefore are most critical to the stratified combustion cold start concept. The focus of this study is to understand these effects and develop the resolutions to the technical challenges through optimization of the injection strategies and piston top designs.

The injection strategies that are investigated in this work are described in this section for later reference. Figure 5 shows two single injection strategies: stratified-charge (SC) start and homogeneous-charge (HC) start. For SC start, the injection takes place during the compression stroke with elevated in-cylinder pressure and temperature which improves the fuel vaporization rate. Ideally, this strategy places a rich mixture cloud around the spark plug and creates a local enrichment for stable combustion as shown by the A/F contour plots in Figure 5. However, if the mixture cloud is misplaced (not around the spark plug), this strategy tends to lead to unstable combustion or misfires and high HC emissions. For the single injection HC start, the fuel is injected during the intake stroke to produce a homogeneous fuel-air mixture in the cylinder. However, it produces heavy surface wetting under cold conditions (as high as 60% total injected fuel, as shown by modeling results in later sections) due to impingement of liquid fuel on the liner and piston surfaces. The heavy surface wetting reduces the overall fuel vaporization and therefore requires more injected fuel (higher enrichment factor) for stable combustion.



Stratified-charge (SC) Homogeneous-charge (HC)

Figure 5. Two single injection strategies

In addition to the single injection strategies, three split injection strategies are evaluated in this work. Both intake-compression (I-C) split injection and compression-compression (C-C) split injection can be used for the stratified-charge (SC) start. The mechanism of the SC split injection is to use the first injection to provide some background vapor fuel around the spark plug (preferable homogeneous) and the second injection (late during the compression stroke) produces the local enrichment around the spark plug. This mechanism is illustrated by A/F contours in Figure 6. The stratified split injection strategy should improve the combustion stability with proper local enrichment and reduce the injected fuel mass due to less surface wetting and stratified charge combustion.

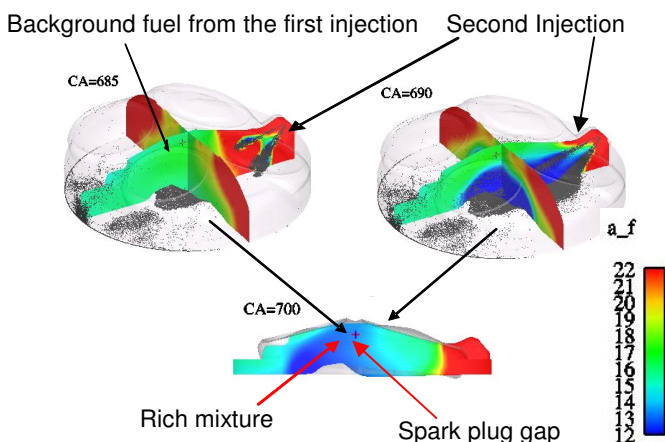


Figure 6. Stratified-charge split injection strategies

The third split injection option is the intake-intake (I-I) split injection for a homogeneous-charge (HC) start. I-I split injection reduces the spray momentum and the surface wetting (Figure 7) compared with single injection. However, I-I split injection still produces higher surface wetting than the two SC split injection strategies. Similar to C-C split injection, I-I split injection is viable only at the low and mid speed range due to limitation of the injector response time between two pulses.

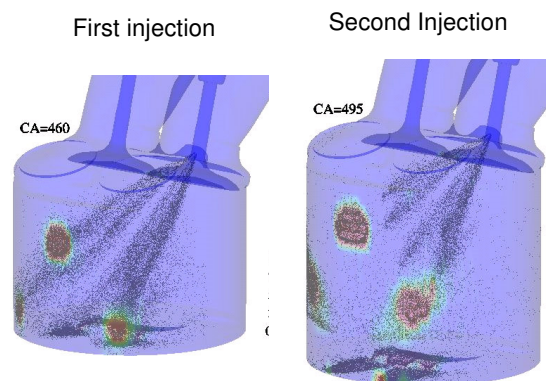


Figure 7. Homogeneous-charge intake-intake (I-I) split injection strategy

MODELING OF ENGINE CRANK AND RUN-UP

As mentioned in the introduction, when using a single injection strategy in the cold start of a D35 EcoBoost engine, relatively high HC emissions are measured and misfires are also observed under certain operating parameters. Specifically, when using single injection stratified-charge (SC) start, it is observed that the optimal injection time EOI (end of injection) is around $EOI=40^\circ$ BTDC at crank speed, and advances with the increasing engine speed. Event 5 and 6 require higher enrichment factors to avoid misfire. The slope of measured IMEP vs. enrichment factor indicates that even with a large enrichment factors, the fuel-air mixture in the combustion chamber is still lean. Higher fuel pressure produces lower HC emission than low fuel pressure. When using a single homogeneous-charge (HC) start strategy, engine data show that a high enrichment factor and a high fuel pressure are required to avoid engine misfire. The data suggests either a very lean fuel-air mixture or poor mixing in the cylinder.

Assessment at 200 RPM

To understand the engine test observations, the effects of injection timing and fuel pressure are first examined at a crank speed of 200 RPM using the SC single injection strategy. This is the first firing event. CFD simulations are performed for an injection timing sweep with $EOI=30^\circ$ to 75° BTDC at two fuel pressures (25 and 40 bar). Figure 8 shows the evolution of the air fuel ratio (A/F) contours and velocity vectors in the range of $CA=675^\circ$ to $CA=710^\circ$ for the case with an injection time $EOI=40^\circ$. These plots are created in the central cross-section plane through the combustion chamber. The results illustrate the mechanism of fuel-air mixture preparation using SC single injection under this condition. As shown in this figure, the fuel spray is injected in the late compression stroke targeting the piston bowl. Due to the low engine speed and the shallow piston bowl design, the top jet of the fuel spray overshoots the piston bowl and distributes fuel vapor to the exhaust side of the cylinder. In addition, the gas flow momentum induced by the fuel injection also carries the majority of rich mixture to the exhaust side of the cylinder at $CA=690^\circ$. In the

meantime, the top jet of fuel injection generates a recirculation flow at the exhaust side that carries the rich mixture back towards the center of the cylinder around the spark plug. At the time of ignition, a stratified fuel-air mixture is formed in the cylinder with most of the rich mixture located at the exhaust side and the lean mixture at the intake side of the cylinder. The ideal distribution of a rich mixture pocket around the spark plug like the one shown in Figure 5 is not realized.

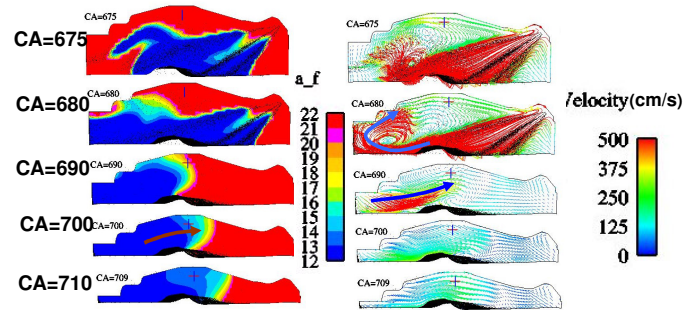


Figure 8. Evolution of A/F contours and velocity vectors for the case of $EOI=40^\circ$ using the SC single injection

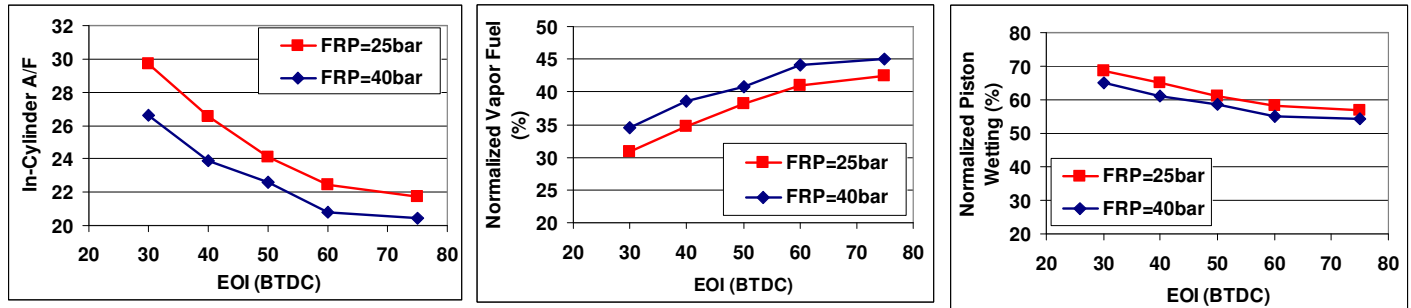


Figure 9. Effect of injection timing (EOI) at two fuel rail pressure (FRP) levels

Figure 9 shows the effect of injection timing (EOI) and fuel pressure on overall in-cylinder A/F, normalized in-cylinder vapor fuel and normalized piston wetting (normalized by the injected fuel mass) based on the simulation results of the injection timing sweep at two fuel pressure levels. As can be seen, advancing injection timing decreases the piston wetting, increases the in-cylinder fuel vaporization, and therefore decreases the in-cylinder A/F. The results also show that the higher fuel pressure decreases the surface wetting and improves the fuel vaporization due to better atomization, as expected.

Because the stratified-charged fuel-air mixture cloud is developed when using this SC single injection strategy, even though advancing injection timing improves the fuel vaporization, it can not be concluded that the optimal fuel distribution around spark plug can be achieved by simply advancing the injection timing. The simulation results of the injection timing sweep (not shown here) show that the injection time $EOI=40^\circ$ produces the richest mixture cloud around the spark plug and is the optimal injection timing at this condition. However, for a wide range of injection timing other than $EOI=40^\circ$, the model results reveal that even with a large enrichment factor, the fuel air mixture around the spark plug is still lean due to poor mixture distribution with these injection timings. The model results are consistent with the engine test observation in terms of the optimal injection timing and benefit of higher fuel pressure. The results also explain that the lack of fuel-air mixture around the spark plug is the main reason causing misfires with low enrichment factors.

The performance of homogenous-charge (HC) single injection is examined with simulations using three fuel pressures: 25, 40 and 60 bar and a high enrichment factor. The evolutions of normalized in-cylinder vapor fuel, piston and liner surface wetting (normalized by the injected fuel mass) are shown in Figure 10. As illustrated by the spray images in Figure 5, with a HC strategy, liquid fuel impinges directly on the piston and liner surfaces and forms liquid films on these wall surfaces during the intake stroke. Due to the cold surface temperature, once the liquid films are formed, the fuel vaporization rate from the wall film is very low. This is evidenced by the very slow decrease of the surface wetting from $CA=480^\circ$ to $CA=600^\circ$ shown by the charts in Figure 10. After $CA=600^\circ$, some liquid films on the liner surface are scrubbed to the piston top surface due to the movement of the piston and the location of the liner film. The transfer of the liquid film leads to the increase of the piston wetting and decrease of the liner wetting. As can be seen from these charts, the HC single injection strategy produces high surface wetting (~55%-60% of total injected fuel) and therefore results in a low fuel vapor generation (~33%-38%). Additionally, it is observed that at intake valve closing (IVC= 608° in this case), the fuel vapor is reduced from its peak value. The reduction is due to loss of fuel vapor to the intake port from the push back flow generated during the compression stroke with late IVC. As a result of heavy surface wetting and strong push back flow, in-cylinder vapor fuel is reduced when using a single HC strategy. Therefore, it is required to inject more fuel (high enrichment factor) to achieve a combustible mixture. The results in Figure 10 also show that the effect of increasing fuel pressure is the decrease of total surface

wetting and increase of fuel vapor due to the balance between better spray atomization and higher spray

penetration. Therefore, a higher fuel pressure produces a richer in-cylinder mixture.

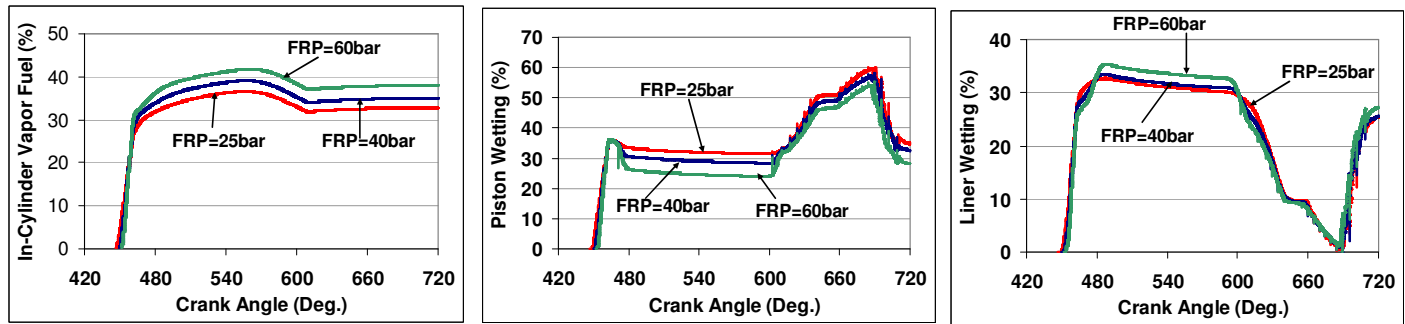


Figure 10. Model results of normalized in-cylinder vapor fuel, piston and liner surface wetting using HC single injection strategy

The model predicted fuel-air mixing is shown in Figure 11 as A/F contour plots in two central cross-section planes through combustion chamber at CA=10° BTDC. The mixture is seen to be quite lean but relatively well mixed. The A/F results confirm that the higher the fuel pressure, the richer the mixture in the cylinder. This correlates with more stable combustion observed in the engine test. The model results also explain that the observed misfire is mainly caused by the lean fuel air mixture when using a low enrichment factor or low fuel pressure, and is not caused by poor mixing. The low fuel vaporization due to heavy surface wetting is the fundamental reason that this strategy requires higher enrichment factors in order to have stable combustion.

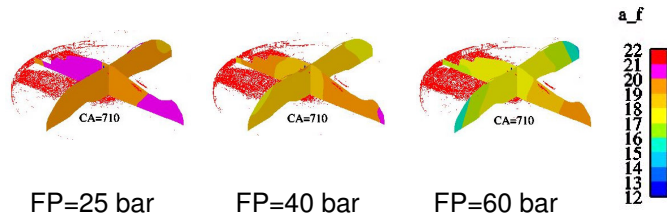


Figure 11. Fuel-air mixture distributions at three fuel pressure levels using the HC single injection strategy

Based on above discussions, achieving stable combustion using single injection strategies at 200RPM

is difficult. On one hand, the SC injection strategy tends to form the rich fuel-air mixture at the exhaust side of the cylinder instead around the spark plug. It also produces high piston surface wetting. On the other hand, the HC injection strategy produces a homogeneous but lean mixture in cylinder due to heavy surface wetting and lost vapor fuel to push back flow. Model results reveal the mechanism behind the unstable combustion and high HC emission observed in engine tests using these injection strategies.

To explore the alternative strategies, the intake-compression (I-C) split injection with injection times EOI=260°, 30°, and compression-compression (C-C) split injection with injection times EOI=75°, 30° are evaluated with the same amount of injected fuel mass. Figure 12 shows the comparison of normalized fuel vapor, piston and liner surface wetting between two split injections and the single injection. The results show that C-C split injection reduces surface wetting from 65% to 59% and increases in-cylinder vapor fuel from 35% to 40%. While I-C split injection produces similar overall surface wetting as C-C split injection, the improvement on the fuel vapor is smaller for I-C split injection due to lost vapor fuel to the intake port during push back.

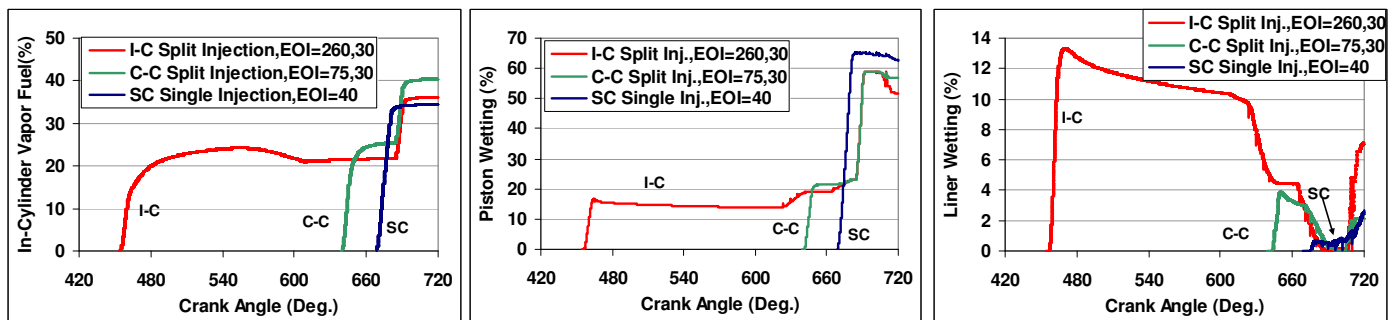


Figure 12. Comparisons of normalized in-cylinder vapor fuel, piston and liner wetting between I-C, C-C split injection and SC single injection

Figure 13 shows A/F contour at CA=680° (before the second injection pulse) for the I-C split injection case.

The velocity-vector plots in the center during the second injection are also shown in this figure. The