

Spray Pattern Optimization for the Duratec 3.5L EcoBoost Engine

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ABSTRACT

A systematic methodology has been employed to develop the Duratec 3.5L EcoBoost combustion system, with focus on the optimization of the combustion system including injector spray pattern, intake port design, piston geometry, cylinder head geometry. The development methodology was led by CFD (Computational Fluid Dynamics) modeling together with a testing program that uses optical, single-cylinder, and multi-cylinder engines. The current study shows the effect of several spray patterns on air-fuel mixing, in-cylinder flow development, surface wetting, and turbulence intensity. A few sets of injector spray patterns are studied; some that have a wide total cone angle, some that have a narrow cone angle and a couple of optimized injector spray patterns. The effect of the spray pattern at part load, full load and cold start operation was investigated and the methodology for choosing an optimized injector is presented. The preliminary dynamometer data obtained from a direct injection engine equipped with the optimized 6-hole injector indicated good mixing at part load and Wide-Open Throttle (WOT) operating conditions. Higher combustion efficiency and lower CO emissions at most operating points indicate better mixing for the DI engine than the baseline PFI engine. The optimized injector was able to demonstrate lower smoke emissions than the baseline injector.

INTRODUCTION

Increasing the fuel economy of engines is critical to meeting CAFE standards, to achieving a reduction of CO₂ emissions and for improving the global environment. An increasing fuel cost is another factor that shifts consumer mindset towards cars with significant fuel economy improvements. Among the methods that achieve improvements in fuel economy and performance

(higher torque and power) are direct injection, turbocharging and downsizing. Downsizing in particular is a promising strategy to lower the fuel consumption of internal combustion engines, Potteau et al. [1]. Gasoline engine downsizing shifts the speed/load operating points to a more efficient region through the reduction of engine capacity while maintaining the full load performance with increased pressure charging, Hancock et al. [2]. In a turbocharged internal combustion engine the inducted air is compressed to a higher density and pressure than the ambient air allowing for more air to be inducted into the cylinder and more fuel to be burned, which in turn increases the maximum power of the engine, Heywood [3]. Turbocharging significantly improves power and torque compared to naturally aspirated engines. Turbocharging makes engine downsizing possible, where a similar or higher power can be extracted from a smaller displacement engine. The traditional disadvantages of turbocharging, turbo lag and knock, are mitigated by the synergy between turbocharging and direct injection. Direct injection reduces the charge air temperature and knock sensitivity allowing higher compression ratios that lead to improved thermal efficiency. The combination of turbocharging with direct injection in a gasoline engine can significantly improve low-end torque and transient behavior due to improved volumetric efficiency and residual scavenging, potentially serving as the key to the successful introduction of real downsized engine concepts, Lang et al. [4], Kleeberg et al. [5]. Duratec 3.5L EcoBoost is one example of such downsized turbocharged direct injection engines.

In direct injection (DI) engines the fuel is injected directly into the cylinder, therefore fuel preparation is an essential factor affecting fuel economy and engine performance. Direct injection reduces the charge air temperature and knock sensitivity allowing higher compression ratios that lead to improved thermal

efficiency. DI gasoline engines are able to achieve both improved fuel economy and improved power. Han et al. [6] outlined the technical challenges that exist in the development of a direct injection combustion system. Wirth et al. [7] outlined challenges encountered in the development of a turbocharged direct injection combustion system. Noyori and Inoue [8] compared the impact of three types of injectors: swirl injectors, fan injectors and multi-hole injectors, on smoke emissions. They concluded that the multi-hole injectors can reduce smoke emissions because of enhanced evaporation, resulting in a shorter liquid length.

The multi-hole injectors allow considerable flexibility in choosing the number of holes and in arranging the holes. The fuel spray pattern is one the critical components of a direct injection gasoline engine as it plays a major role in achieving good air-fuel mixing and good combustion characteristics. There are several competing requirements for the optimization of the spray pattern. One primary requirement is strong cross-cylinder penetration to achieve homogeneous fuel-air mixtures. However, excessive penetration of the fuel spray will increase fuel impingement on the chamber walls, which could cause reduced engine durability due to oil dilution. In addition, the wetting of piston surface by a strongly penetrating fuel spray is believed to be a major source of soot emissions. Cold-start operation, as well, requires a spray pattern with less penetration in order to achieve stratified-charge operation for fast catalyst light-off. An optimized injector spray pattern can reduce cylinder wall wetting which leads to a reduction in oil dilution. It also reduces soot emissions over a wide window of fuel injection timing.

SPRAY PATTERN OPTIMIZATION

The 3-D CFD code MESIM was extensively used in the port optimization process. MESIM is an extension of KIVA-3V, Amsden [9, 10], with enhanced dynamic mesh moving algorithm and improvements in the multi-hole direct injection spray and film models, Han and Xu [11]. MESIM was also updated with a level set method, G-equation based, combustion model linked to the chemical kinetics software CHEMKIN, Liang and Reitz [12], Liang et al. [13]. Combustion simulation results are not presented in this paper. MESIM was successfully used at Ford Motor Company for the combustion system development and optimization of a light stratified-charge direct injection engine, Yi et al. [14-16], Han et al. [6, 17], and for the development of a spray-guided direct injection engine, VanDerWege et al. [18], Iyer et al. [19]. In the EcoBoost direct injection engine development a similar methodology was adopted. Yi et al. [20] and Kapp et al. [21] present an overview of the entire optimization process of the 3.5L V6 EcoBoost combustion system; Iyer et al. [22] discussed the port optimization process and Xu et al. [23] and Reiche et al. [24] present the cold start optimization process.

The combustion chamber, the intake port and the exhaust port were discretized and a structured

hexahedral mesh was generated with ICEMCFD. The typical mesh size for engine simulations that was used here is 2-3 mm and the computational domain contains approximately 250,000 computational cells when the piston is at BDC. The computational mesh moves dynamically following the valve lift profile and the piston motion. The flow and energy equations are solved coupled with an RNG k- ϵ turbulence model, Han and Reitz [25]. Initial and boundary conditions were setup with the use of existing dynamometer data or from a quasi-dimensional code, GESIM, simulations, Dai et al. [26]. At the inflow and outflow boundaries a constant pressure boundary condition was set. Simulations were initiated before IVO and were continued through the remainder of the exhaust stroke, the intake stroke and the compression stroke. The gasoline spray breakup model is based on the work done by Yi and Reitz [27].

The injector used in this study is a side-mounted multi-hole injector. The multi-hole injectors lend themselves very well to spray optimization because of the ability to design the number of holes and the hole placement that enables individual targeting of the spray jets. Each jet can be defined by two angles, the off-axis angle, the angle between the axis of the injector and the axis of the plume, and the azimuthal angle, defined between the axis of the injector and the axis of the plume in a top view. Usually the injector mounting angle is dictated by packaging considerations.

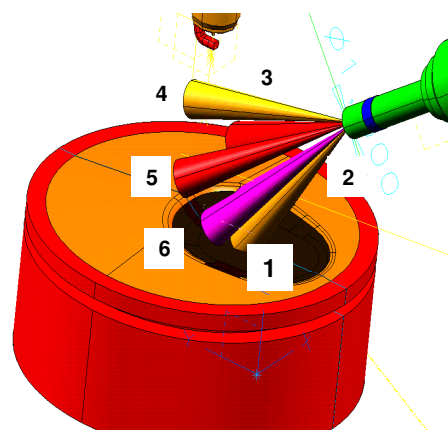


Fig. 1 Multi-hole injector.

The spray jets of a 6-hole injector were optimized to avoid valve wetting and to have reduced liner and piston wetting. The top three spray jets (jets 3, 4, and 5 in Fig. 1) were designed to provide good mixing in homogeneous-charge operation. Jet 4 was designed to target between the intake valves and to reach as far as possible on the exhaust side but at the same time have reduced liner wetting. Jets 3 and 5 were oriented such as to have minimal valve wetting, as valve wetting is a major source of soot. Jets 1, 2, and 6 were designed to provide good combustion stability for light stratified-charge at cold-start operation by making sure they are contained in the shallow piston bowl. This way a mixture cloud is formed and maintained in the bowl that enables combustion stability at cold-start operation. Jets 1, 2, and 6 were also designed to have reduced piston wetting, especially jet 1.

The spray pattern of the 3.5L EcoBoost combustion system was optimized to improve air-fuel mixing, to reduce wall wetting (for reduced smoke emissions) and to improve combustion stability during cold-start. 3D CFD methods were employed to optimize the injector spray pattern to satisfy a wide range of combustion system objectives. Several spray pattern iterations were analyzed at part load, full load, and cold start operating conditions. The air/fuel mixing preparation process, surface wetting, and combustion characteristics at cold start operation were compared for all spray pattern iterations. CFD was used for the screening of different spray pattern designs. Based on CFD results, three optimized spray pattern iterations were tested in the single-cylinder engine out of which one optimized spray pattern was chosen for multi-cylinder engine tests.

EFFECT OF SPRAY PATTERN AT PART LOAD OPERATION

The initial spray pattern optimization study was performed at part load, 1500 rpm/2.62 bar BMEP. Then, the optimized spray patterns were investigated at high speed full load and under cold start conditions.

In the attempt to optimize the spray pattern for improved mixing one has to take into consideration the effect of the spray pattern on surface wetting as it impacts the engine smoke level. One goal in our study was to reduce the smoke level with the optimized injector. The amount of piston, liner and valve wetting are known to contribute directly to the smoke level. The Duratec 3.5L engine parameters are bore=92.5 mm and stroke=86.7 mm. The simulation setup conditions at part load used in this study are listed in Table 1.

Table 1. Setup conditions at 1500 rpm/2.62 bar BMEP

Specification	
Intake cam duration	248
Exhaust cam duration	248
IVO (BTDC)	30
EVC (ATDC)	10
Injector flow rate (cc/s)	20
Injection pressure (bar)	50
Start of injection (ATDC)	429
MAP (kPa)	44.5

EFFECT OF WIDE SPRAY PATTERNS ON MIXING AND SURFACE WETTING - Wider spray patterns improve air-fuel mixing as they distribute the fuel better in the combustion chamber. This section compares the baseline spray pattern iteration 5 to a couple of wider spray patterns, iterations 10 and 14.

Fig. 2 shows the spray patterns investigated here, viewed from the intake side of the chamber. Fig. 3 shows the effect of the spray pattern on the evolution of the tumble ratio (TR) throughout the engine cycle. It can be noticed that spray iteration 10 enhances the tumble flow development in the cylinder. The negative sign of the tumble ratio indicates the direction of rotation. The absolute value of the tumble ratio indicates the magnitude of the tumble flow for the different configurations. The tumble ratio in MESIM was calculated using the definition from the KIVA code:

$$TR = \frac{M_y}{I_y} = \frac{\sum_{\text{cells}} \rho_i V_i [(z_i - z_{\text{center}})u_i - (x_i - x_{\text{center}})w_i]}{\frac{\pi N}{30} \sum_{\text{cells}} \rho_i V_i [(z_i - z_{\text{center}})^2 + (x_i - x_{\text{center}})^2]} \quad (1)$$

Where M_y is the angular momentum about the y-axis; I_y is the mass moment of inertia about the y-axis; N is the rotational speed (rot/min); (u_i, v_i, w_i) is the velocity at the cell i ; $(x_{\text{center}}, y_{\text{center}}, z_{\text{center}})$ is the center of mass of the cylinder; (x_i, y_i, z_i) is the coordinate of cell i ; V_i is the volume of cell i ; and ρ_i is the density of cell i .

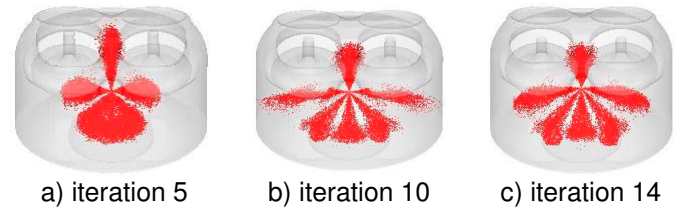


Fig. 2 Baseline spray pattern (iteration 5) and two wide spray patterns (iterations 10 and 14).

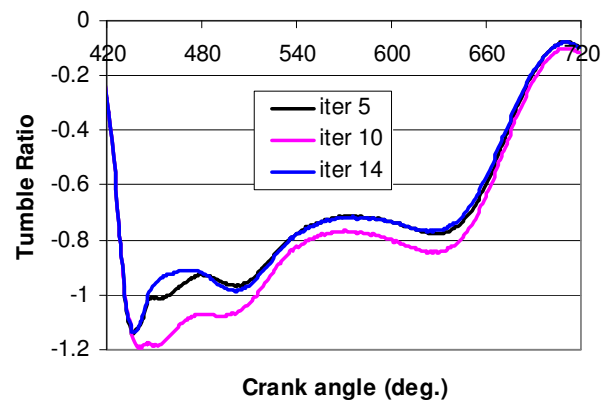


Fig. 3 Spray pattern effect on tumble motion, 1500 rpm/2.62 bar BMEP.

Fig. 4 shows that the spray pattern has a large effect on the in-cylinder velocity shown in a central cutting plane at two crank angles (CA) (the intake side is on the right side of the image). The widest spray pattern, iteration 10, is more favorable to tumble flow development and it also generates higher turbulence intensity at TDC. Table 2 summarizes the effect of the wider spray patterns on the tumble ratio at BDC and turbulence intensity at TDC.

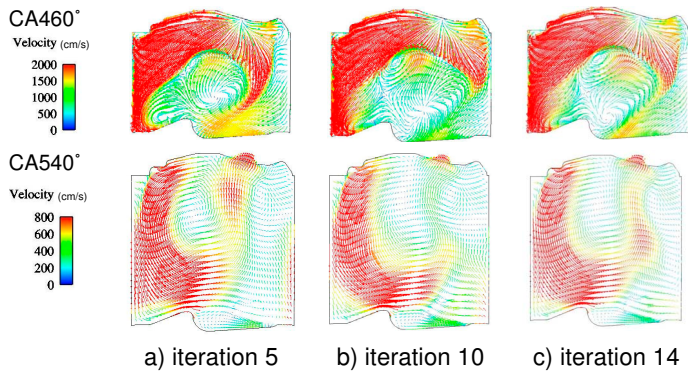


Fig. 4 Velocity in central plane of the wide fuel spray patterns, 1500 rpm/2.62 bar BMEP.

Table 2. Effect of wide fuel spray patterns on tumble ratio (TR) at BDC and turbulence intensity (u') at TDC, 1500 rpm/2.62 bar BMEP.

1500rpm/ 2.62 bar BMEP	TR @ BDC	u' @ TDC (m/s)
Iteration 5	0.78	1.78
Iteration 10	0.82	1.79
Iteration 14	0.78	1.77

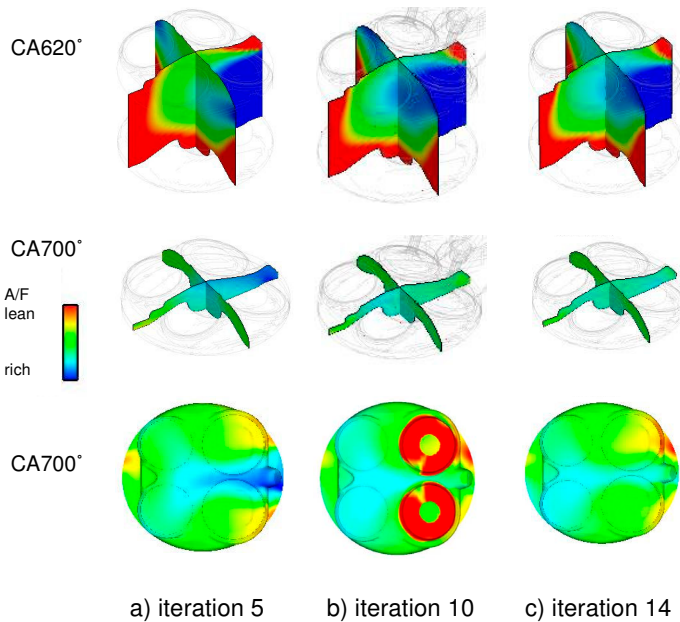


Fig. 5 Effect of wide spray patterns on mixing, 1500 rpm/2.62 bar BMEP

Fig. 5 shows the air-fuel (A/F) ratio contours generated using spray pattern iterations 5, 10, and 14, at 1500 rpm/2.62 bar BMEP. The A/F contours are shown in two central cutting planes (the intake side is on the right side of the image) at two crank angle values, 620° and 700° ATDC, and the third row shows a top view of the combustion chamber. It can be noticed that spray pattern iterations 10 and 14 improve mixing homogeneity

considerably compared to the baseline iteration 5 and that is because they have a wider spray jet distribution. A noticeable difference in the mixing between the sprays is seen in the first row that shows the A/F ratio contours at the crank angle (CA) of 620°. Iterations 10 and 14 have a wider mixing length; the stoichiometric mixture region reaches further towards the exhaust side (left side of the image).

Fig. 6 shows the histogram or probability density function (PDF) of the equivalence ratio distribution in the cylinder by volume at CA=20° BTDC. The PDF distribution shows the percentage of the in-cylinder volume occupied by mixtures with a certain equivalence ratio. A narrow and tall distribution curve of the in-cylinder equivalence ratio indicates better mixing homogeneity. It can be noticed that both iterations 10 and 14 improve mixing compared to iteration 5 as they have a smaller cylinder volume occupied by very rich or lean mixtures. Spray pattern iteration 14 improves mixing the most out of the three cases studied here.

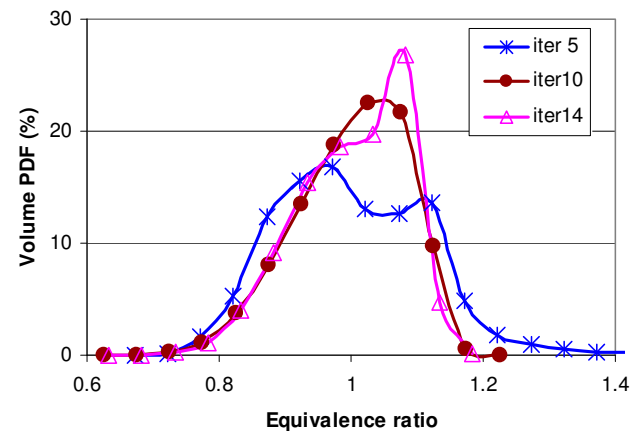


Fig. 6 Effect of wide spray patterns on mixing, 1500 rpm/2.62 bar BMEP.

Table 3 shows the effect of the spray pattern on piston, liner and valve wetting. It shows the amount of fuel that forms a film on the surface normalized by the injected fuel. It can be noticed that both wide spray patterns, iterations 10 and 14, reduce piston and liner wetting compared to the baseline iteration 5. Iteration 14 has negligible valve wetting however, iteration 10 has considerably larger valve wetting than baseline iteration 5. It is confirmed in Fig 2b that two jets of the wide spray pattern iteration 10 clip the sides of the intake valves and this leads to its high valve wetting, as seen in Table 3.

Table 3. Effect of wide spray patterns on surface wetting, 1500 rpm/2.62 bar BMEP.

Spray iteration	Piston wetting (%)	Liner Wetting (%)	Valve wetting (%)
Iteration 5	0.024	0.015	0.002
Iteration 10	0.010	0.007	0.604
Iteration 14	0.000	0.001	0.000