

Development and Optimization of the Ford 3.5L V6 EcoBoost Combustion System

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ABSTRACT

Recently, Ford Motor Company announced the introduction of EcoBoost engines in its Ford, Lincoln and Mercury vehicles as an affordable fuel-saving option to millions of its customers. The EcoBoost engine is planned to start production in June of 2009 in the Lincoln MKS. The EcoBoost engine integrates direct fuel injection with turbocharging to significantly improve fuel economy via engine downsizing. An application of this technology bundle into a 3.5L V6 engine delivers up to 12% better drive cycle fuel economy and 15% lower emissions with comparable torque and power as a 5.4L V8 PFI engine.

Combustion system performance is key to the success of the EcoBoost engine. A systematic methodology has been employed to develop the EcoBoost engine combustion system. Instead of a trial-and-error approach, the EcoBoost combustion system development was focused on the fundamental physics with emphasis on the optimization including injector spray pattern, piston geometry, and intake port design, and innovation of operating strategies. The development methodology was led by 3-dimensional CFD modeling together with experiments that used optical, single cylinder, and multi-cylinder engines. As a result, a higher quality design has been achieved through only a few hardware iterations.

INTRODUCTION

In response to the rapidly changing North American automotive industry, Ford has identified EcoBoost as Ford's near term fuel economy / CO₂ sustainability technology, with mid and long term emphasis on Hybrids, Diesels, and Plug-In Hybrids (Kapp et al., 2008). As previously reported by Borrmann et al. (2002), turbocharged direct-fuel-injected engines have a number of functional advantages relative to both naturally-aspirated and turbocharged port-fuel-injected engines. It significantly improved power and torque and achieves consistent real-world fuel economy benefit, regardless of drive cycle as compared with turbocharged port fuel injected engines.

The specifications of the 3.5L V6 EcoBoost engine are shown in Table 1. The engine is equipped with intake VCT (iVCT) and targeted to deliver at least 340 hp (254 kW) at 5650 rpm and 340 lb-ft (461 N-m) of torque from 2000 to 4000 rpm. Both the peak power and torque are quite competitive with premium V8 naturally aspirated engines with displacements around 5.0L.

Table 1: Specifications of the D35 EcoBoost Engine Combustion System

Specification	3.5L V6 EcoBoost iVCT
Bore	92.5 mm
Stroke	86.7 mm
Bank Angle	60 deg
Bore Spacing	106 mm
Compression Ratio	10.0:1
Intake Duration	236 deg
Intake Valve Opening	40 deg BTDC
Intake Valve Closing	16 deg ABDC
Exhaust Duration	236 deg
Exhaust Valve Opening	46 deg BBDC
Exhaust Valve Closing	10 deg ATDC
Fuel Recommendation	95 RON
Maximum Engine Speed	6300 RPM
Maximum Cylinder Pressure	100 bar (mean + 3 sigma)

In order to meet the above engine performance targets and engine durability requirement, the combustion system is required to meet the following attributes:

- 1) Capable of operating at 17 bar BMEP over a wide engine speed range between 1500 rpm to 5500 rpm.
- 2) Capable of achieving fuel-air mixing homogeneity equivalent to its naturally-aspirated PFI counterpart at part-load conditions to ensure fuel economy targets; and good mixing at full load conditions to ensure engine performance.
- 3) Capable of PZEV engine-out emissions
- 4) Capable of minimizing smoke emissions and fuel liner impingement for low oil dilution, to ensure engine durability.

Figure 1 shows the 3.5L V6 EcoBoost combustion system configuration. A 6-hole fuel injector is mounted beneath the intake ports. The spark-plug is located at the

centre of the chamber. The piston has a shallow piston bowl that is mainly designed to ensure stable combustion during engine cold-start operations.

Development of a direct-injection spark-ignition (DISI) engine for optimal performance is technologically challenging. One reason is that the functional requirements for fuel-air mixture formation are fundamentally different between stratified-charge and homogeneous charge operation. In the EcoBoost engine, homogeneous charge mode is used almost exclusively, except at the cold-start operation where stratified-charge operation is found to be very beneficial. Local stratification at cold-start condition provides reliable ignition and stable combustion with reduced emissions. The mixture formation is affected by many complex interactions of fuel spray, in-cylinder flow motion, and piston geometry.



Figure 1 Schematic of the 3.5L V6 EcoBoost Combustion System.

A CFD-based methodology has been employed to develop the 3.5L V6 EcoBoost combustion system. This approach focuses on the fundamental physics to optimize the combustion system, through injector spray pattern, piston geometry, intake port design, and innovative operating strategies. This methodology was led by 3-dimensional CFD modeling together with a testing program involving optical, single cylinder, and multi-cylinder engines as shown in Figure 2. The CFD modeling has been carried out with a Ford in-house developed code, named MESIM, with leading-edge spray and combustion physical models (Yi et al. 2001, 2002, 2004, Han et al., 2002, Liang et al. 2007, Amsden et al, 1985, 1989, 1997). Figure 3 depicts a flowchart

of the EcoBoost combustion system development methodology. It starts with detailed optimization of three key components, the intake ports, the spray pattern, and the piston crown design, using both CFD modeling and optical engine experiments. The optimized designs together with engine operating strategy were tested and confirmed in both single cylinder and multi-cylinder engines before finally being integrated into the vehicle. As shown in Table 2, one obvious advantage of such an approach is that it significantly reduces the development time and cost while achieving a high quality design all thanks to the physical insights of the details and interactions of the in-cylinder phenomena.

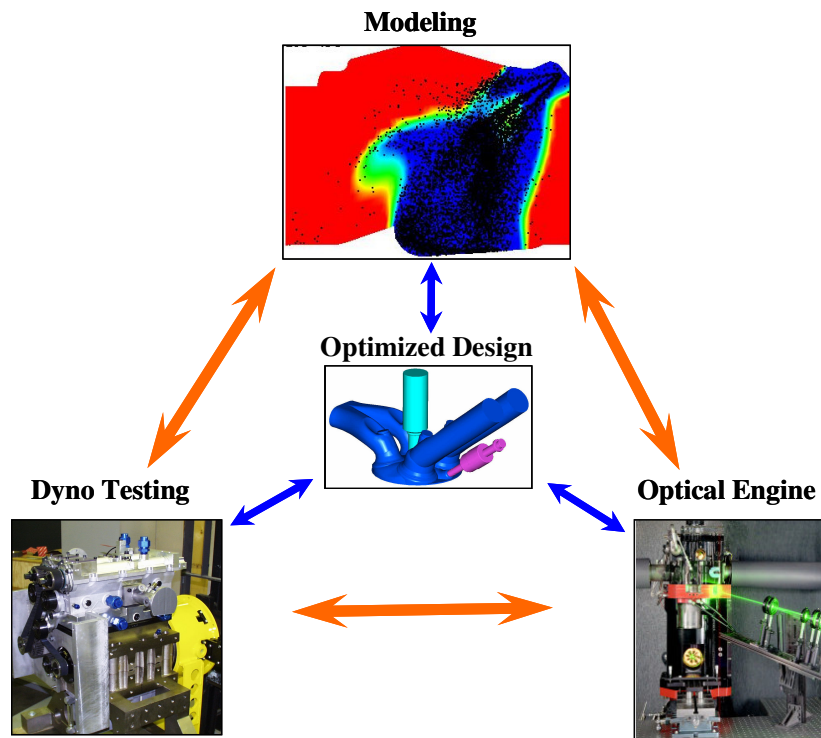


Figure 2 A systematic combustion system development methodology that integrates CFD modeling, optical engine experiments, and dynamometer testing

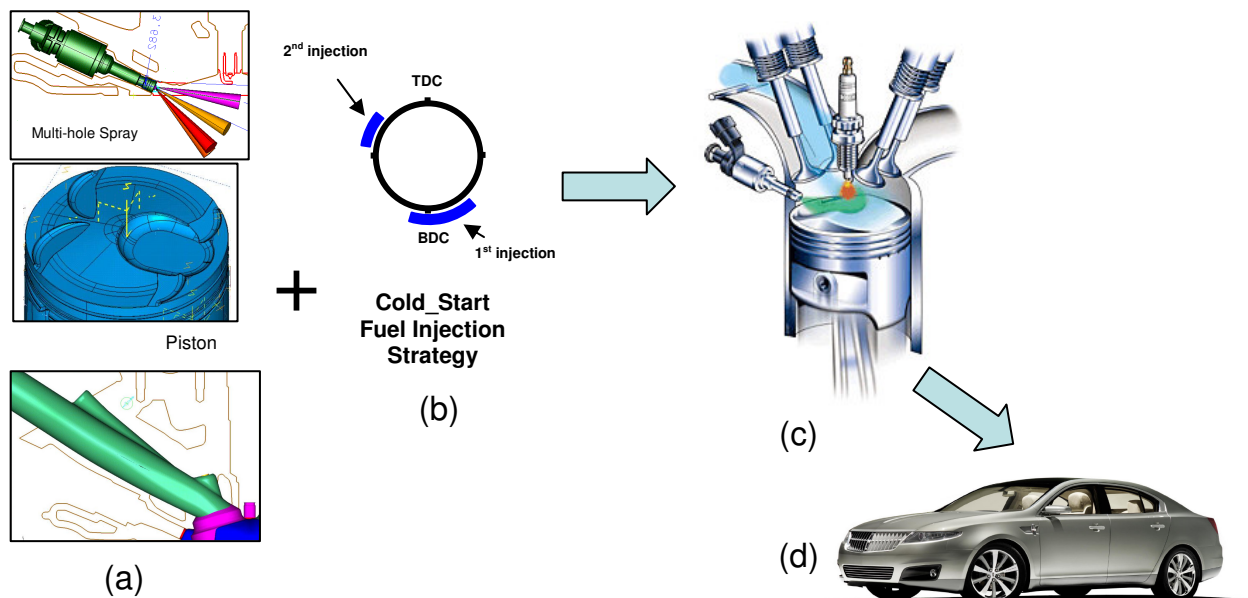


Figure 3 3.5L V6 EcoBoost combustion system development flow chart.

INTAKE PORT OPTIMIZATION

As discussed by Wirth et al. (2000) and Yi (2009), there are some unique aspects for the turbo-charged DISI intake port design, as compared with naturally aspirated DISI combustion systems. Similar to the naturally aspirated DISI engines, there are two major factors that drive the intake port optimization in the turbo-charged DISI engines. The first is the requirement for port flow capacity to improve engine power. The second is the in-cylinder flow motion that affects fuel droplet trajectory and fuel-air mixing, as well as the turbulent intensity, that affects burn rate, combustion stability, heat transfer between wall and in-cylinder gas, and engine knock tendency. Unfortunately, the port flow capacity and in-cylinder flow motion requirements are generally competing. An intake port with higher flow capacity generally results in lower in-cylinder flow motion, and an intake port with higher flow motion generally yields lower flow capacity. Typically intake port design requires a balance of these effects. Variable flow controlling devices have also been used to separate these two requirements by improving flow capacity at high load operations and generating high flow motion where the mixing is critical but flow capacity can be sacrificed.

Wurms et al. (2002) designed a variable tumble plate in their 2.0L Audi FSI combustion system. In this system, the intake port was optimized for maximum flow capacity, but is split into upper and lower sections by a plate. When the engine operates at high load or full load conditions, the control valve is open, allowing flow through both sections for maximum flow capacity. At operating points where high tumble motion is needed, the valve is closed, blocking the lower part of the intake port. The flow is forced through the upper section, generating high tumble motion.

The relative importance of flow capacity and charge motion are different between naturally-aspirated and turbocharged DISI engines. The flow capacity in turbocharged engines is less important, however the charge motion is essential. Much higher tumble motion is needed to enhance engine burn rate to mitigate the elevated knock tendency in turbocharged engines. High tumble motion will also promote turbulence mixing. However, higher tumble motion also increases the heat transfer between cylinder wall and in-cylinder mixture. This may hurt fuel economy due to increased heat loss from in-cylinder gas to the wall early in the expansion stroke. At high load condition, this may make the engine more knock limited due to increased heat absorption from the wall, in the intake and early compression strokes. Higher tumble motion also has the potential to make the fuel-air mixing homogeneity more challenging.

Shown in Figure 4 are CFD-predicted in-cylinder flow velocity and fuel spray distributions in the cylinder center plane at BDC, and the air-fuel ratio distribution at 20 degrees before TDC, in a DISI engine. The operating condition is high speed, high load. It indicates that the intake flow keeps fuel spray from penetrating to the exhaust side (as shown in left figure), making the mixture on the exhaust side very fuel lean at the time of combustion (right figure). The interaction between intake flow and fuel-spray will be aggravated at boosted conditions, due to increased momentum of the intake flow with increased air density, and with higher tumble motion, due to the increased strength of the rotational flow motion.

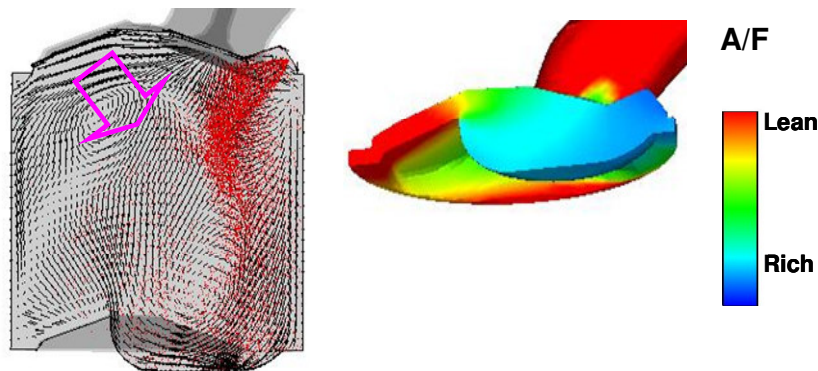


Figure 4 CFD-predicted spray evolution and fuel-air mixing in a boosted DISI engine. Left: velocity and spray droplet distribution at BDC; Right: air fuel ratio contour at 20 degrees before TDC. The operating condition is high speed and high load.

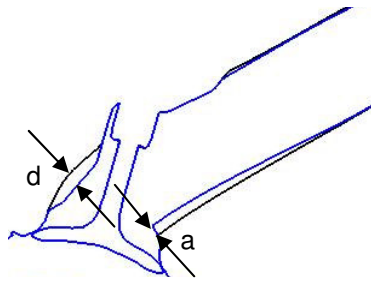


Figure 5 Port design parameters 'a' and 'd' shown on Port 5 (blue) vs. GTDI iVCT XO (black).

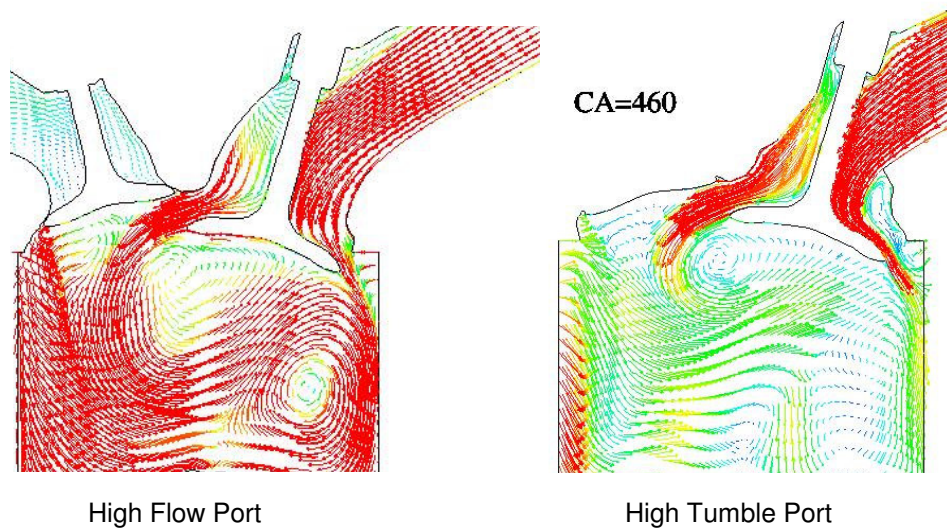


Figure 6 Near-valve flow structure comparison between high flow port (left) and high tumble port (right). The high flow port has a more evenly distributed velocity flow field around the valve seat area, while the high tumble port has a more biased flow structure with flow separation on the short side of the port; 5000 rpm, 19 bar BMEP.

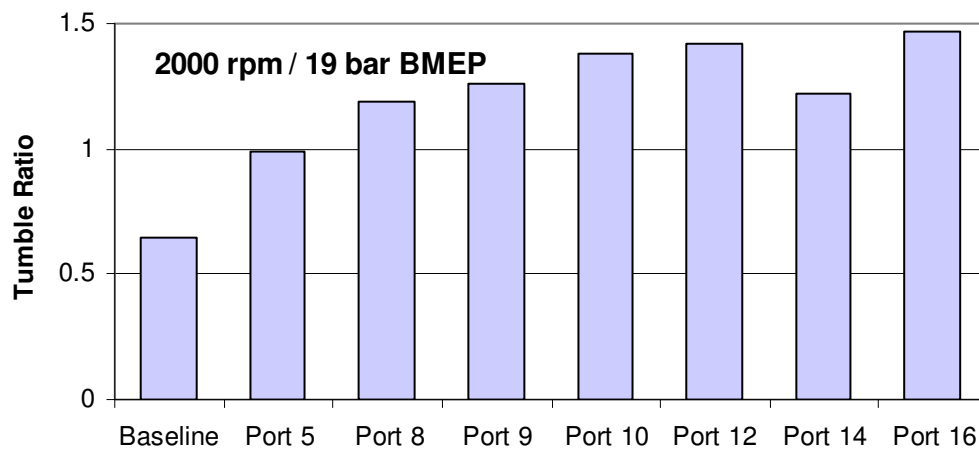


Figure 7 CFD-calculated tumble ratio at BDC at 2000 rpm, boosted conditions for different intake port designs.