

High-Speed Flow and Combustion Visualization to Study the Effects of Charge Motion Control on Fuel Spray Development and Combustion Inside a Direct-Injection Spark-Ignition Engine

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MSU College of EngineeringDavid L.S. Hung, Guoming Zhu and Harold Schock
Michigan State Univ.

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ABSTRACT

An experimental study is performed to investigate the effects of charge motion control on in-cylinder fuel-air mixture preparation and combustion inside a direct-injection spark-ignition engine with optical access to the cylinder. High-pressure production injector is used with fuel pressures of 5 and 10 MPa. Three different geometries of charge motion control (CMC) device are considered; two are expected to enhance the swirl motion inside the engine cylinder whereas the third one is expected to enhance the tumble motion. Experiments are performed at 1500 rpm engine speed with the variation in fuel injection timing, fuel pressure and the number of injections. It is found that swirl-type CMC devices significantly enhance the fuel-air mixing inside the engine cylinder with slower spray tip penetration than that of the baseline case without CMC device. Combustion images show that the flame growth is faster with CMC device compared to the similar case without CMC device.

INTRODUCTION

Improvement in fuel efficiency and reduction in exhaust emissions are the main goals behind the new developments in internal combustion engines (Mittal et al., 2010). The concept of direct-injection spark-ignition (DISI) engine has the potential to achieve such goals. In this technology, fuel is directly injected into the engine cylinder, which offers great flexibility to control the fuel injection timing, its duration and the number of injections. Note that the fuel-air mixture

preparation in the combustion chamber is one of the key factors that influence the in-cylinder combustion characteristics and hence the engine performance (Hung et al., 2007). Therefore, optimizing the fuel-air mixture homogeneity is an important parameter for the engine designers. In general, a homogeneous fuel-air mixture is achieved by injecting the fuel during the intake stroke. In addition, the use of a charge motion control (CMC) device is an important factor that affects the flow (Mittal and Schock, 2010) and hence the fuel-air mixing and combustion inside the engine cylinder. It is expected that the CMC device imparts an angular momentum to the charge entering the engine cylinder.

Several studies have been reported to investigate the influence of charge motion control on the engine performance. Clarke and Stein (1999) combined the variable valve timing with the charge motion control valve (CMCV). Variable valve timing was obtained using the dual equal variable camshaft timing (VCT) strategy. The combination of dual equal VCT with a CMCV allows an engine to be operated either at or near stoichiometric or at lean conditions, which allows the use of a NO_x trap for the purpose of further reducing air pollutants. The authors indicated that the synergy between the CMCV and the dual equal VCT allows the fuel consumption to be less than the fuel consumption during lean operation at standard valve timing. This is due to the fact that CMCV increases the in-cylinder charge motion, and hence improves the combustion and the ability to handle the charge dilution, which occurs from increased levels of internal

exhaust gas recirculation resulting from valve timing retard. [Li et al. \(2000\)](#) investigated the effects of swirl control valve on in-cylinder flow using a laser doppler anemometry technique. [Mittal and Schock \(2010\)](#) used molecular tagging velocimetry to study the influence of charge motion control on in-cylinder flow inside an internal combustion engine assembly. [Kim et al. \(2005\)](#) investigated the effects of injection timing and intake port flow control on fuel wetting inside the engine cylinder. They found that a tumble mixture-motion plate inside the intake port significantly reduced cylinder liner and piston top fuel wetting. This is because the use of the tumble mixture-motion plate provided more turbulence, which effectively enhanced the mixing during the intake process. [Lee and Heywood \(2006\)](#) studied the effects of CMCV on combustion characteristics and hydrocarbon emissions. The authors concluded that CMCV improved mixture preparation due to increased swirl and tumble intensities which enhanced fuel transport, distribution, and evaporation. CMCV in the closed condition allowed reduced fuel injection and retarded spark timing strategies that reduced hydrocarbon emissions significantly during the cold start due to greater fuel evaporation and faster burning rate.

Overall, previous investigations show that a charge motion control device is an important factor that controls the combustion process, and hence, influences the engine performance ([Mittal and Schock, 2010](#)). However, to the best of authors' knowledge, visualization studies of the charge motion control device on in-cylinder fuel-air mixture preparation and combustion are not available. Therefore, an experimental study is performed to investigate the effects of charge motion control on in-cylinder fuel-air mixture preparation and combustion inside a direct-injection spark-ignition engine. Experiments were performed at 1500 rpm engine speed with the variation in fuel injection timing, fuel pressure and the number of injections. In the following sections, a detail of experimental setup is first outlined, followed by the results of various tests performed. Finally, concluding remarks are summarized from this work.

EXPERIMENTAL SETUP

The engine used in the present work is a four-valve, two intake and two exhaust, 0.4 liter single-cylinder spark-ignition engine. It has a bore diameter of 83 mm and stroke length of 73.9 mm. A flat-top piston with optical access is used. This provided a compression ratio of 9.75:1. [Mittal et al. \(2010\)](#) used a custom-designed piston in the same engine, which allowed the compression ratio of 13.5:1. The head accommodates a pressure transducer to record the in-cylinder pressure data. A view of the combustion chamber geometry showing intake and exhaust valves, direct-injector, spark plug and the pressure transducer is illustrated in [Figure 1](#) ([Mittal et al., 2010](#)). It should be noted that in this paper 0° crank angle corresponds to the top dead center (TDC) of the compression, and therefore, -180 crank angle degrees (CAD) corresponds

to the bottom dead center (BDC) of the intake, i.e. 180° BTDC (before top dead center). Different fuel injection timings (240°, 210° and 180° BTDC) are considered with gasoline fuel. A high-pressure direct-injection (HPDI) 7-hole injector ([Mittal et al., 2010](#)) is used with the fuel pressures of 5 and 10 MPa.



Figure 1. Optical engine combustion chamber

SETUP FOR FUEL SPRAY VISUALIZATION

[Figure 2](#) shows the experimental rig used for spray visualization tests. The laser is introduced into the cylinder through the flat-top piston with optical access. A Mie scattering technique is used to visualize the liquid phase of the fuel dispersion inside the combustion chamber. A quartz cylinder is used to provide the optical access to the cylinder for high-speed imaging. The fuel spray was imaged with a Photron APX-RS non-intensified highspeed CMOS camera with a Nikon 105 mm AF micro lens. The camera was set to operate at 10 kHz, which provided an image size of 512 × 512 pixels. At 1500 rpm engine speed, each frame corresponds to 0.9 crank angle degrees. A high repetition rate pulsed copper vapor laser, synchronized with the high-speed camera and the fuel injection timing logic, was used to illuminate the liquid fuel dispersion. For each test condition, the engine was first motored to reach the desired rpm, i.e. 1500 rpm. Once the engine was stabilized, a signal from the Cosworth engine controller was sent out to the fuel injector to trigger the start of injection at a specific crank angle position as well as to trigger the camera to start recording the specified number of images in consecutive cycles. The fuel injection duration at each test point is defined to achieve a stoichiometric air-fuel ratio based on gasoline. For each imaging test, five-injection-cycle spray images were recorded to visualize the fuel dispersion with 400 consecutive frames from each cycle.

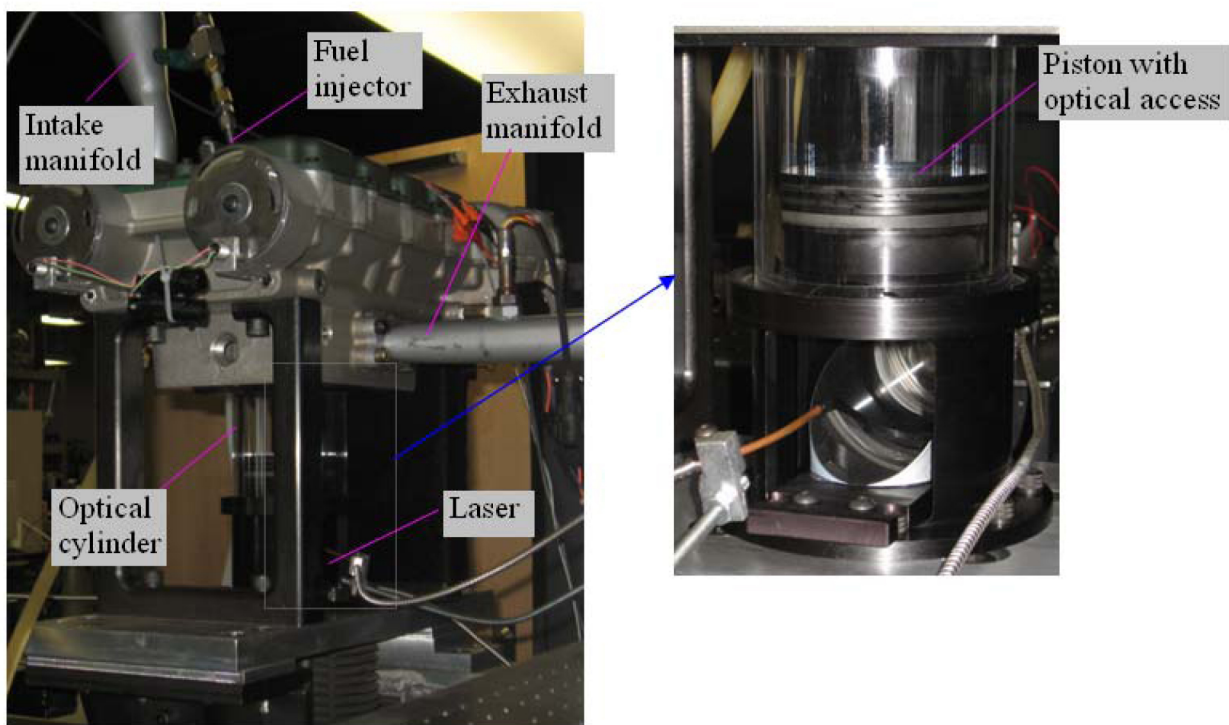


Figure 2. Experimental rig for in-cylinder fuel spray visualization

SETUP FOR COMBUSTION VISUALIZATION

Figure 3 shows the experimental rig for combustion visualization. Note that the quartz cylinder was replaced with the metal cylinder for the combustion tests. Experiments were performed at 1500 rpm engine speed with part-load condition (0.45 bar MAP). This part-load condition was selected due to optical limitations of the flat-top piston. The effects of split (or dual) injection were also studied and compared with the corresponding cases of single injection by maintaining the same relative air-to-fuel ratio (λ), inverse of fuel-to-air equivalence ratio (ϕ). With the split injection, the second injection was 90 CAD apart from the first injection and the two pulse widths (of fuel injection) were kept the same. The combustion images were captured by Photron APX-RS highspeed camera (operated at 10 kHz) through the optical piston. For each test condition, the engine was first motored to reach the desired rpm. Once the engine was stabilized, a signal from the Opal-RT engine controller was sent out to the fuel injector to trigger the start of injection at a specific crank angle position as well as to trigger the camera at the spark timing crank angle position to start recording the specified number of images in consecutive cycles. The fuel injection duration at each test point was selected to achieve the desired relative air-to-fuel ratio. For each imaging test, forty consecutive cycles were recorded to visualize the combustion process with 200 frames from each cycle. Due to optical engine limitations, the fuel supply was cut off as soon as the

camera recorded the specified number of cycles. In-cylinder pressure was recorded with one degree of crank angle resolution that has been synchronized with the imaging signal. The Kistler piezoelectric pressure transducer was used with the measurement range varying from 0 to 250 bars. The averaged in-cylinder pressure data is then used to evaluate the engine performance. Mass fraction burned (MFB) and burn durations are determined using the well-known Rassweiler-Withrow method (Rassweiler and Withrow, 1938). A linear model for the polytropic index during the combustion process is used to evaluate the pressure change due to the volume change (Mittal et al., 2009).

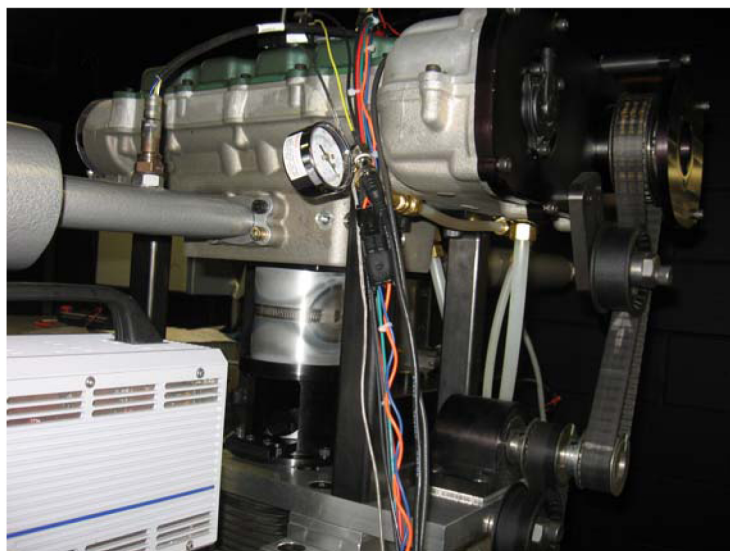


Figure 3. Experimental rig for combustion visualization

CHARGE MOTION CONTROL DEVICE

The charge motion control device was installed between the intake manifold and the intake port. The nature of CMC device influence depends on its geometry. Three different geometries of charge motion control device were considered (see Figure 4); two are expected to enhance the swirl motion inside the engine cylinder whereas the third one is expected to enhance the tumble motion. Each CMC device used in this study reduced the port cross sectional area by about 75%, i.e., the flow area about 25%. Experiments were performed with both the conditions: CMC device open (i.e. without CMC device) and CMC device closed (i.e. with CMC device).

RESULTS AND DISCUSSION

Results of fuel spray development; combustion visualization and in-cylinder pressure analyses are presented for a direct-injection spark-ignition engine with both open (without) and closed (with) charge motion control devices. Three different geometric configurations of charge motion control device are considered.

FUEL SPRAY VISUALIZATION

Figure 5 shows the spray development of gasoline with open (left column) and closed CMC devices of all three configurations, i.e. swirl-types 1 (second column) and 2 (third column) and tumble-type (right column). In each case, high-pressure direct-injection injector is used with 5 MPa of injection pressure at 1500 rpm engine speed. The start of injection (SOI) is at -240 crank angle degrees (or 240° BTDC). The size of each spray image shown in this paper is 512×512 pixels. The physical size of a pixel is about 0.19 mm. Note that the intake valves are located towards the left side of each spray image. The spray development at 223.8° BTDC shows that the spray tip penetration is faster with tumble-type CMC device compared to the baseline case with open CMC device. However, it is to be noticed that the spray tip penetration is slower with swirl-type CMC devices compared to both open and tumble-type CMC devices. Spray images at 216.6° BTDC clearly show that the spray tip penetration is even slower with swirl type-1 CMC device compared to the swirl type-2 CMC device. It is interesting to note that no significant difference is observed in spray

development when the baseline case (with open CMC device) is compared with the tumble-type CMC device. However, the fuel dispersion is wider with swirl-type CMC devices. Also, note that the intensity values in these images (with swirl-types 1 and 2) are relatively low compared to the intensity values in spray images of both open and tumble-type CMC devices (see the presence of more liquid fuel towards the left side of the piston top with both open and tumble-type CMC devices). This clearly shows that the air-fuel mixing improves with swirl-type CMC devices with reduced piston top impingement.

Figure 6 shows the spray development of gasoline with open and closed CMC devices of all three configurations. In each case, an HPDI injector is used with 5 MPa of injection pressure at 1500 rpm engine speed. The start of injection is at 180° BTDC. Similar to the results observed with SOI at 240° BTDC, the spray tip penetration is slower with swirl-type CMC devices than that of open and tumble-type CMC devices. Note that the fuel dispersion is wider with swirl- and tumble-type CMC devices than that of open CMC device. Therefore, it is expected that the air-fuel mixing improves with the charge motion control device.

SPRAY TIP PENETRATION

Figure 7 shows the effects of injection pressure (at 5 and 10 MPa) and the injection timing (at 240° and 180° BTDC with 5 MPa of injection pressure) on spray tip penetration with open CMC device. The penetration length is determined as the axial location of the spray tip from the injector tip. As expected, the spray tip penetration is faster with the injection pressure of 10 MPa than that of 5 MPa. Note that the spray tip penetration is slower when fuel injection starts at 180° BTDC than that of 240° BTDC injection timing due to upward movement of the piston. Figure 8 shows the effects of different configurations of charge motion control devices on spray tip penetration. As observed in spray images (of Figure 5), the spray tip penetration is faster with tumble-type CMC device compared to open and swirl-type CMC devices. Note that the spray tip penetration is slowest with swirl type-1 CMC device compared to open and other configurations (swirl type-2 and tumble-type) of CMC devices.

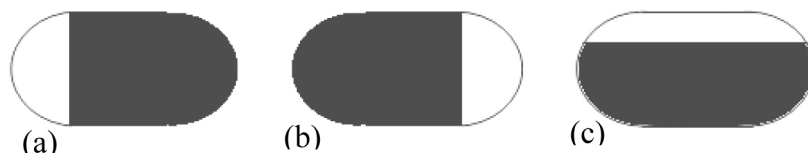


Figure 4. Three different types of charge motion control devices: (a) Swirl type-1, (b) Swirl type-2 and (c) Tumble type CMC devices