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# Spray and combustion characteristics of gasoline and diesel in a direct injection compression ignition engine

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## HIGHLIGHTS

- ▶ Spray and combustion characteristics were investigated for gasoline and diesel.
- ▶ Similar liquid penetration length between two fuels under non-evaporating condition.
- ▶ Shorter liquid length and narrower spray angle for gasoline at evaporating condition.
- ▶ Premixed-dominant combustion with blue chemiluminescence for gasoline.
- ▶ Diffusion-dominant combustion with highly luminous soot incandescence for diesel.

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## ABSTRACT

The spray and combustion characteristics of gasoline and diesel were investigated in a direct injection compression ignition engine equipped with a common rail injection system. The spray evolution was observed under a non-evaporating condition in a constant volume chamber and under an evaporating condition in an optical engine. Under the non-evaporating condition, the liquid penetration length was similar between the gasoline and diesel. The gasoline spray exhibited a relatively larger spray cone angle than that of diesel spray. However, the gasoline spray exhibited a significantly shorter liquid penetration length and narrower spray angle than that of the diesel spray under the evaporating condition. The maximum liquid penetration length was maintained constant regardless of the injection pressure for each fuel at the evaporating condition. The diesel spray formed wall wetting through the fuel impingement on the combustion chamber due to the long liquid penetration length at an early injection timing of −32 crank angle degree after top dead center (CAD ATDC).

A series of combustion experiments was performed in order to investigate the performance and emissions in a metal engine and the flame characteristics in an optical engine. A low load condition (indicated mean effective pressure (IMEP) of approximately 0.45 MPa) was tested under an injection timing range from −40 to 0 CAD ATDC. The maximum thermal efficiency was similar between the two fuels with injection in close vicinity of the TDC. The gasoline combustion created a larger amount of hydrocarbon, carbon monoxide, and comparable nitric oxides ($NO_x$) but had a lower soot emission compared with diesel combustion. However, the $NO_x$ emission of the gasoline combustion was significantly reduced with the premixed charge compression ignition (PCCI) combustion via early injection. The direct combustion visualization demonstrated that the natural luminosity (NL) of the gasoline combustion was dominated by the chemiluminescence from the premixed burn while the NL of the diesel combustion was primarily attributed to the soot incandescence from the diffusion burn. However, the PCCI combustion via the early injection was dominated by the chemiluminescence from the premixed burn for both fuels.



## 1. Introduction

Compression ignition (CI) engines are the most fuel-efficient engines developed for transportation purposes, largely due to their relatively high compression ratio and lack of throttling losses [1]. However, conventional diesel engines have relatively high emissions of nitrogen oxides ($NO_x$) and soot emissions. Thus, a number of advanced combustion technologies have been studied in an attempt to reduce the harmful emissions while maintaining the high level of thermal efficiency. Many researchers have proposed that homogeneous charge compression ignition (HCCI) and premixed

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**Nomenclature**

| | |
|---|---|
| $t_{inj}$ | injection timing command (CAD ATDC) |
| $L$ | liquid penetration length (mm) |
| $A$ | spray cone angle (deg) |
| HCCI | homogeneous charge compression ignition |
| CAD | crank angle degree |
| PCCI | premixed charge compression ignition |
| ATDC | after top dead center |
| CI | compression ignition |
| IMEP | indicated mean effective pressure |
| HC | hydrocarbon |
| SOI | start of injection |
| CO | carbon monoxid |
| SINL | spatially integrated natural luminosity |
| $NO_x$ | nitrogen oxides |
| fps | frame per second |

charge compression ignition (PCCI) are promising techniques for simultaneous reduction of $NO_x$ and soot emissions [2–4]. PCCI combustion adopts moderately early injection timing in order to improve the combustion phase controllability [5] and high HC and CO emissions which are the most challenging problems in HCCI combustion.

Although advanced diesel combustion technologies can reduce the harmful emissions with high thermal efficiency, there is a problem in the preparation of premixed mixture prior to combustion. Conventional diesel fuels, with high cetane numbers (CNs) of greater than 40, are highly prone to auto-ignition with a shorter ignition delay before the fuel and air mix properly [6]. In addition, the volatility of diesel is insufficient to create a well-premixed mixture [7].

Gasoline-like fuels generally have properties of high volatility and superior resistance to auto-ignition. These fuels have been tested extensively focusing on performance and emissions at high load operation [6,8–11]. It has been demonstrated that gasoline-like fuels exhibit a long ignition delay in pPPCI combustion [6]. Gasoline-like fuels are advantageous in achieving high load condition with very low fuel consumption and very low $NO_x$ and soot emissions.

Analysis of spray characteristics has been an important issue in CI engine research [12–14]. Payri et al. [15] recently examined the injection rate and liquid penetration length of gasoline and diesel sprays under non-evaporating conditions using a common rail injection system. They reported that the gasoline and diesel sprays exhibited very similar spray momentums and spray behaviors for the liquid penetration length and spray cone angle. Zheng et al. [16] conducted a direct visualization of the spray and combustion using gasoline-like fuels in a constant volume vessel under a high pressure and high temperature. They reported that gasoline-like fuels exhibited a longer ignition delay and lift-off length compared with those of the diesel fuel [8–10].

It is valuable to investigate the physical mechanism and relationship for the spray and combustion characteristics because most engine research to date has focused on the improvement of the fuel consumption and exhaust emissions. In this study, the gasoline and diesel spray characteristics were investigated under both non-evaporating and evaporating conditions. The engine performance and emissions were tested with the injection timing. In addition, the direct combustion visualization assisted understanding of the different flame characteristics in the gasoline and diesel combustion.

## 2. Experimental setup and conditions

### *2.1. Macroscopic spray visualization under non-evaporating conditions*

A schematic diagram of the optical diagnostics system used for the non-evaporating macroscopic spray is shown in Fig. 1a. The definitions of liquid penetration length and spray cone angle are also described in Fig. 1a. The spray images were obtained via direct photography of the Mie-scattering technique. Optical access to the pressure chamber was available through three quartz windows with diameters of 90 mm. The chamber was pressurized with nitrogen at room temperature. For the direct photography of Mie–scattered light, a spark light (MVS-2061-CE96, EC&G Optoelectronics) was directed toward the sprays through a window, while a charge coupled device (PCO CCD Imaging, SensiCam) camera captured the scattered light through the perpendicular window. The spray images were captured using an imaging system that was synchronized with the spark light source.

### *2.2. Engine performance and optical diagnostics for the evaporating spray and combustion*

A four stroke, single cylinder diesel engine was used to test the gasoline and diesel. The engine specifications are listed in Table 1. The engine had a bore of 100 mm and a stroke of 125 mm with a compression ratio of 17.4:1. A schematic diagram of the experimental setup is shown in Fig. 1b. The engine speed was controlled using an 82 kW alternating current (AC) dynamometer.

Identical common rail-injection systems were used for the gasoline and diesel. The properties of the gasoline and diesel used in the experiments are listed in Table 1. The gasoline fuel has a lower density and viscosity than those of diesel. The distillation temperature range is also significantly lower for gasoline compared with diesel. The fuel pressure in the common rail was adjusted using a pressure controller (ZB-1200, Zenobalti Co.). The fuel injection duration was controlled using a programmable injector driver (IDU 5000B, Zenobalti Co.). Ambient air was introduced through an intake surge tank in order to settle fluctuation. An external air compressor was used to boost the intake air.

In-cylinder pressure was measured using a piezoelectric pressure transducer (6052C, Kistler). The in-cylinder pressure data were acquired with a crank angle (CA) resolution of 0.2° and was averaged over 100 cycles in order to calculate the heat release rate.

The analysis of exhaust gas emission was undertaken using an exhaust gas analyzer (MEXA 1500D, Horiba) consisting of a non-dispersive infrared absorption (NDIR) analyzer for the carbon monoxide (CO) and carbon dioxide ($CO_2$) measurements, a chemiluminescence detector (CLD) analyzer for the $NO_x$ measurements and a flame ionization detector (FID) analyzer for the hydrocarbon (HC) measurements.

The engine was modified to allow optical access to the combustion chamber. A cross-sectional view of the optical engine system is shown in Fig. 1c. An elongated piston was equipped to enable the mounting of a 45° mirror beneath the piston quartz window. The bottom of the piston bowl (top of the piston–crown window) was flat, rather than convex as in the production engine, so that undistorted images of the combustion event could be captured while the optical engine was designed to have a geometric