

Low Emissions and High-Efficiency Diesel Combustion Using Highly Dispersed Spray with Restricted In-Cylinder Swirl and Squish Flows

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ABSTRACT

A new clean diesel combustion concept has been proposed and its excellent performance with respect to gas emissions and fuel economy were demonstrated using a single cylinder diesel engine. It features the following three items: (1) low-penetrating and highly dispersed spray using a specially designed injector with very small and numerous orifices, (2) a lower compression ratio, and (3) drastically restricted in-cylinder flow by means of very low swirl ports and a lip-less shallow dish type piston cavity.

Item (1) creates a more homogeneous air-fuel mixture with early fuel injection timings, while preventing wall wetting, i.e., impingement of the spray onto the wall. In other words, this spray is suitable for premixed charge compression ignition (PCCI) operation, and can decrease both nitrogen oxides (NOx) and soot considerably when the utilization range of PCCI is maximized.

However, in diffusive combustion, especially at full load, a low-penetrating spray potentially causes higher soot emissions and results in lower maximum torque. In this case, item (2) is applied to recover full-load performance. The lower compression ratio enables diffusive combustion

phasing to be advanced more with an earlier injection timing because of a larger margin between the compression-end pressure and the allowable maximum in-cylinder pressure. This results in lower soot emissions because enough time is created to oxidize soot before the end of the combustion period.

A lower compression ratio often leads to worse cold-condition engine performance aspects, such as cold startability, unburned hydrocarbons, and white smoke. Item (3) is applied to compensate for such practical problems. Drastically weakened in-cylinder flow keeps the compression-end temperature to the same level as a conventional engine with an ordinary compression ratio by decreasing heat-flux escaping through the chamber wall (i.e., heat-loss).

Although a weak in-cylinder gas motion might lead to higher soot emissions due to slower fuel-air mixing, it should be noted that the highly dispersed spray of item (1) enables PCCI-dominant combustion in which the fuel-air mixing process is less dependent on in-cylinder flow.

In this way, these three items act mutually to compensate for each other's drawbacks, while maximizing their advantages.

Consequently, NO_x emissions in the New European Driving Cycle (NEDC) can be reduced drastically to less than 1/4 of the level of a conventional engine, or less than half of the Euro 6 standard without deteriorating fuel consumption, full-load torque, or cold-condition performance.

INTRODUCTION

Carbon dioxide (CO₂) is becoming a world-wide concern that is considered to be one of the most significant causes of global climate change. For this reason, CO₂ reduction is an urgent challenge particularly in the traffic and transportation sectors. From this point of view, the diesel engine is one of the most promising practical power source options owing to its higher thermal efficiency, and much research has been conducted to steadily improve its performances [e.g. 1, 2, 3, 4, 5, 6, 7, 8, 9]. However, diesel engines emit more NO_x and soot than gasoline engines and therefore require costly additional devices, such as emissions aftertreatment systems, high-pressure injection systems, EGR, and turbocharging systems to meet severe emissions regulations like the Euro 6 standard. This cost disadvantage is one reason preventing diesels from growing in popularity throughout the world. This study imposed the following hardware limitations to keep costs low: 1) no use of a de-NO_x aftertreatment catalyst, 2) the use of a single stage turbocharger, not a two-stage turbocharger, 3) the use of a commercially available common rail injection system with a maximum injection pressure of up to 180 MPa. Under constraints 1) to 3), a new combustion concept was developed that can reduce NO_x to less than half the level stipulated by the Euro 6 emissions standards in the NEDC (i.e., 40 mg/km or less) without deteriorating fuel consumption. This study constructed the basic idea of the concept, and the fundamental performance of the engine based on this concept was confirmed using a single-cylinder engine. Subsequently, more realistic feasibility tests such as full-load performance, and startability and emissions in cold conditions were conducted using a multi-cylinder engine.

METHODOLOGY

Critical engine hardware parameters impacting engine performance include the fuel injector specifications (diameter and number of nozzle orifices, spray umbrella angle, and the like), compression ratio, swirl ratio, and piston cavity configuration. The best combination of these parameter settings was explored to establish this novel combustion concept. In order to reach an optimal solution as fast as possible through numerous choices, this development utilized an in-cylinder visualization technique with an optically accessible engine [10], 3D-CFD incorporating the KIVA code, and the UniDES [11] in-house code that performs a 0-dimensional cycle simulation for a comprehensive analysis, in addition to general engine tests.

Generally, a single cylinder engine is more convenient to use than a multi-cylinder engine due to its more precise control of engine conditions such as the EGR ratio, input fuel amount, intake pressure, and so on. However, it is not appropriate for more practical tests such as full torque, cold startability, or emissions. Therefore, both types of engines were used in this study. The specifications of the test engines are shown in Table 1. The conventional engine is the one used in the Toyota Avensis, which is an inline 4-cylinder engine with a displacement of 2.2 L. It is equipped with a common rail injection system. The new combustion system proposed in this study is drastically changed from the conventional engine. It has a lower compression ratio, specially designed injectors with very small and numerous orifices, a lip-less shallow type piston cavity, and a very low swirl ratio using a new straight-port cylinder head. A schematic diagram of the single-cylinder engine is shown in Figure 1. The engine is equipped with motor driven compressor to enable intake pressure to be set freely. A soot filter and cooler is attached to the exhaust gas recirculation (EGR) system to properly condition the EGR gas. A schematic diagram of the multi-cylinder engine is shown in Figure 2. A low pressure loop (LPL) EGR system was added to the original Avensis engine, which has a conventional high pressure loop (HPL) EGR system. Cylinder pressure was measured by a piezoelectric pressure transducer. Combustion noise was measured by the 450 model AVL combustion noise meter. Soot was measured by the 415 model AVL smoke meter.

Table 1. Specifications of test engines

	Conventional	Developed engine
Displacement [L]	0.557/cylinder	
Bore × stroke [mm]	86×96	
Compression ratio	15.8	14
Injector: Diameter [mm] × number Umbrella angle [deg.]	φ0.113×9 155	φ0.08×16 140
Piston cavity type	Lip-bowl	Lip-less shallow
Intake port type	Helical	Straight
Swirl ratio	2.2	0.5

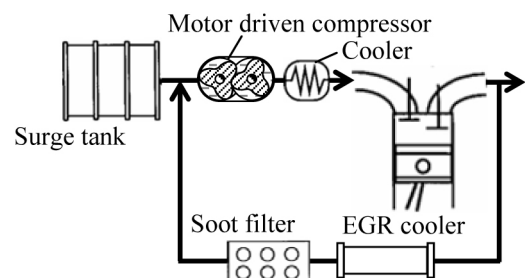


Figure 1. Schematic diagram of single-cylinder engine system used for basic tests

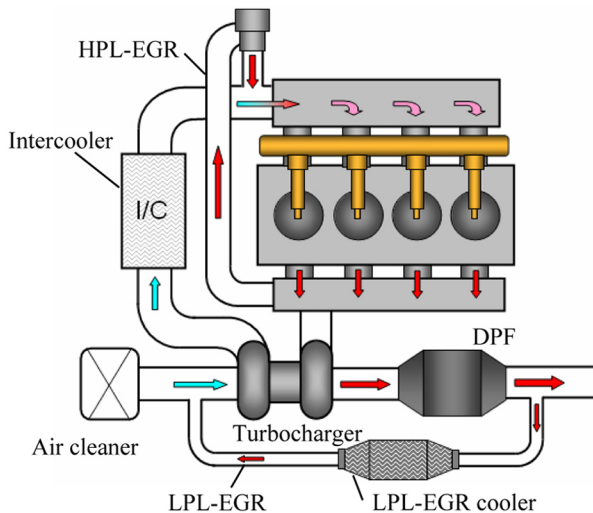


Figure 2. Schematic diagram of multi-cylinder engine system used for full-torque and cold startability tests

A schematic diagram of the optically accessible engine is shown together with the imaging system in Figure 3 [10]. It is a long-piston version of the single-cylinder engine with the same basic specifications such as bore, stroke, and connecting rod length as the base engine. The spray and combustion images in the cylinder were taken by a high-speed video camera.

The primary models used in KIVA and UniDES [11] are summarized in Table 2 and 3, respectively. The calculated results are used in the later discussions.

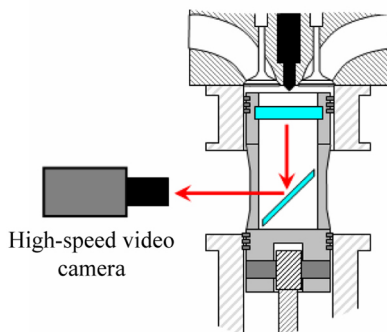


Figure 3. Schematic diagram of optically accessible engine with imaging system

Table 2. Primary models used in KIVA

Physical process	Model
Droplet breakup/atomization	Surface-wave-growth[12]
Ignition	Shell model[13]
Combustion	Laminar-turbulence char. Time model[14]
Turbulence	RNG k-ε model[14]

Table 3. Primary models used in UniDES [11]

Physical process	Model
Spray formation	Multi-zone model
Air entrainment	Hiroyasu model [15]
Droplet size	Kawamura's equation [16]
Ignition	SHELL model [13]
Combustion	Laminar-turbulence char. Time model [14]
Turbulence	Ikegami model [17]
Heat-loss	Woschni model [18]

CONSTRUCTION OF NEW COMBUSTION CONCEPT

Figure 4 shows the essence of the proposed combustion concept, which consists of three items. A specially designed injector, which can produce a low-penetrating and highly dispersed spray, is used in the concept. This spray is more advantageous for the utilization of PCCI. At the same time, a lower compression ratio and restricted in-cylinder flow motion are also employed. The purpose of the former is to improve full-load torque even when a low-penetrating spray is used, and the latter is to reduce heat loss for better fuel economy and to recover engine performance in cold conditions even with the low compression ratio. The following sections will discuss in more detail the positive and negative effects of each of the three items, and how the negative effects of each were overcome.

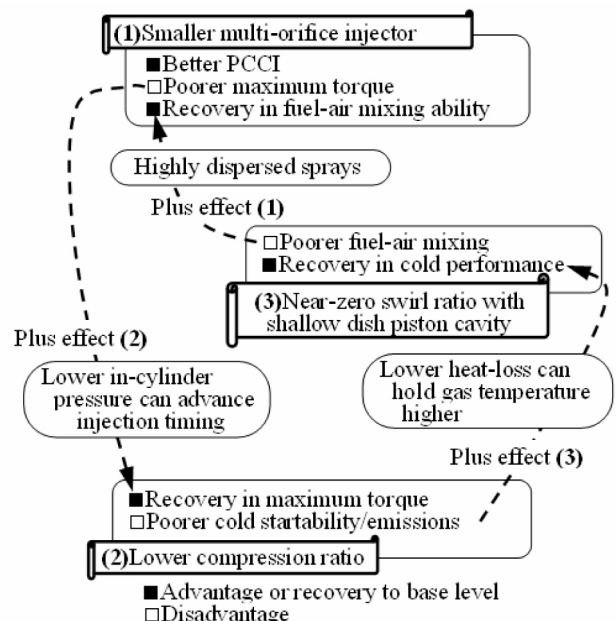


Figure 4. Essence of the proposed combustion concept, which maximizes synergy by mutual combinations of three items

Specially designed injector

Generally, PCCI helps to lower both NO_x and soot emissions. However, as the engine speed and load become higher, the PCCI mixture become more heterogeneous due to the lack of time for fuel-air mixing, which has the effect of increasing emissions. In this case, advancing the injection timing further to secure enough mixing time would increase the amount of unburned hydrocarbons (HC) or cause oil dilution by much heavier impingement of the liquid spray onto the chamber wall. This is because the lower ambient pressure caused by the further advancement of the injection timing makes increases spray penetration and the lower temperature slows fuel evaporation. Therefore, the spray characteristics suitable for PCCI require both lower penetration for less wall-impingement and higher atomization for faster evaporation.

Previous reports have described the spray penetration (S) and the Sauter mean diameter (D_{32}) in the following equations [15, 16].

$$S = 2.95 \times ((P_{cr} - P_a) / \rho_a)^{0.25} \times (d \cdot t)^{0.5} \quad (t > t_b) \quad [m] \quad (1)$$

$$D_{32} = 72.36 \times (P_{cr} \times 10^{-6})^{-0.38} \times d \quad [m] \quad (2)$$

where, P_{cr} : common rail pressure [Pa], P_a : ambient pressure [Pa], d : orifice diameter [m], t : time from injection start [s], t_b : droplet breakup time [s]. D_{32} can be used to identify the degree of spray atomization and a smaller D_{32} means better atomization. The equations above imply the following: (1) a lower common rail pressure (P_{cr}), is beneficial to lower penetration (S), but not to better atomization, (2) a smaller orifice diameter (D) is beneficial to both lower penetration and better atomization. Consequently, it was determined to use a very small orifice diameter of 80 micro-meters. On the other hand, such a small diameter requires a larger number of orifices to keep the same level of fuel flow rate as with a conventional injector. Thus, the proposed injector uses numerous small-diameter orifices. In this paper, this PCCI-oriented injector is called the micro-multi injector (MMI). Figure 5 shows a schematic comparison between conventional and MMI sprays to distinguish this spray concept more clearly.

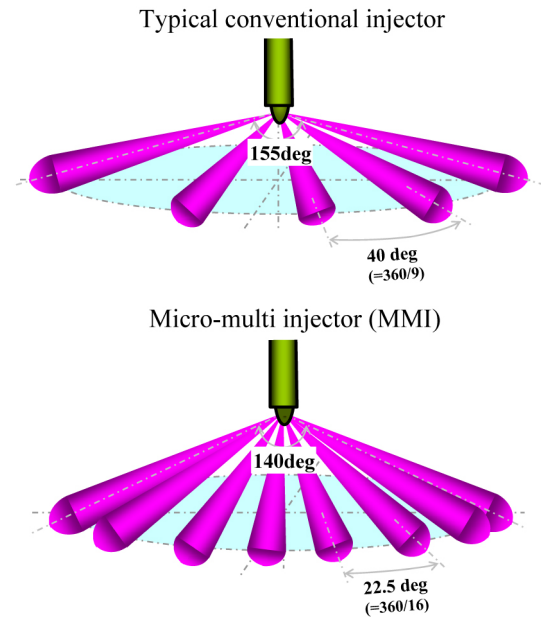


Figure 5. Schematic comparisons between conventional spray (top) and highly dispersed MMI spray (bottom)

Figure 6 compares a conventional spray (left) and an MMI-produced spray (right) at an injection pressure of 130 MPa, photographed 0.5 ms from injection start. D_{32} is 15 μ m for the conventional spray and 12 μ m for the MMI spray. This demonstrated that the MMI achieves a lower-penetrating and better-atomized spray, which is preferable for PCCI.

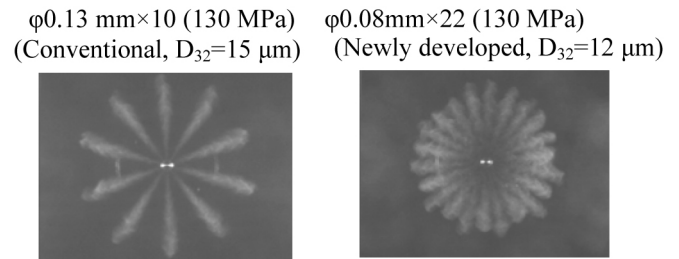


Figure 6. Comparison between conventional spray (left) and highly dispersed MMI spray (right) photographed at 0.5 ms

Figure 7 shows photographs of a combusting MMI-spray taken using the optically accessible engine from the bottom. Two cases were examined to investigate the effect of the number of orifices on the diffusive combustion of MMI spray. With a 22-orifice injector (left), the adjacent portions of the spray partially overlap each other, and seem to generate locally rich regions judging from the flame configurations. In contrast, in a 16-orifice injector (right), the spray portions are well arranged spatially with no interactions between the adjacent portions. In addition, the engine experiment also showed that the 16-orifice injector emitted

less soot than the 22-orifice injector. Accordingly, the 16-orifice injector was adopted in this study.

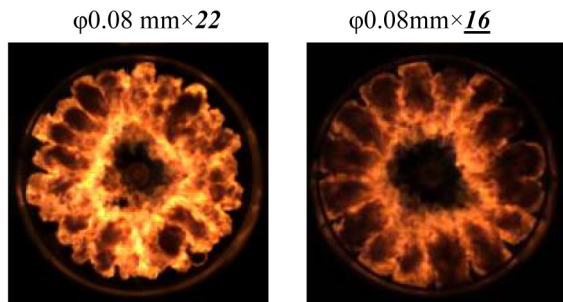


Figure 7. Effect of number of orifices on diffusive combustion, photographed from the bottom using optically accessible engine (left: number of orifices = 22, right: 16)

The spray umbrella angle was selected in consideration of hydrocarbons in PCCI and soot in diffusive combustion. A narrower cone angle is beneficial to PCCI owing to less wall impingement of the spray by creating a longer distance to the wall. It is also advantageous to full-load torque, as discussed in the following section. Consequently, a 140-degree spray umbrella angle was adopted, compared to 155 degrees for a conventional engine.

Lower compression ratio

Generally, a highly dispersed and low-penetrating spray like the MMI-spray described above tends to generate more soot in diffusive combustion, which could lead to a serious torque reduction problem at full load. This study applied a lower compression ratio to recover full-load torque performance. Figure 8 shows cylinder pressures and heat release rates of two cases with different compression ratios of 15 and 14. The allowable maximum in-cylinder pressure ($P_{\max}$) is 16 MPa in this engine, and smoke at full load was also limited to an allowable level. The injection timing and input fuel quantity were independently adjusted so as not to exceed both the limitations of allowable $P_{\max}$ and smoke level. Since the lower compression ratio case creates lower compression-end pressure, the injection timing can be advanced further owing to the larger margin between $P_{\max}$ and the compression-end pressure. In addition, the combustion was completed earlier than in the case of the higher compression ratio, resulting in lower soot. Therefore, in spite of the low-penetrating spray, full-load torque can be recovered to the conventional level.

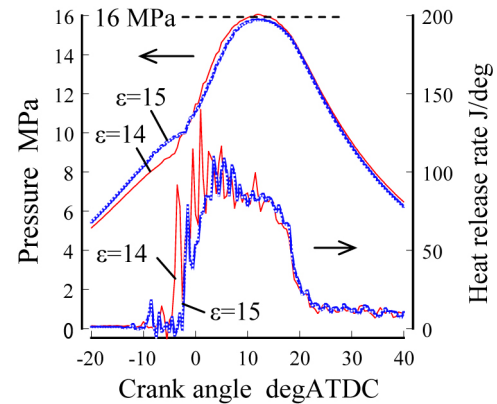


Figure 8. In-cylinder pressure and heat release rate with different compression ratios, while keeping maximum in-cylinder pressure at less than 16 MPa and smoke number at less than 0.8FSN at a full load of 2000 rpm (measured)

However, a lower compression ratio may worsen fuel economy. This section discusses the relationship between the compression ratio and thermal efficiency using the UniDES 0-dimensional in-house code [11]. Figure 9 shows a schematic overview of UniDES with its primary models. Although UniDES is based on a 0-dimensional cycle simulation, it can reproduce the fuel-air mixing process closely by combining the originally proposed multi-zone and probability density function (PDF) models. It has been reported to accurately predict basic engine performance, such as indicated specific fuel consumption (ISFC), emissions, and combustion noise [11]. UniDES was used to estimate thermal efficiencies for various compression ratios with different injection timings. Figure 10 shows a simulated example of heat release rates with changes in injection timings. It is supposed that UniDES can reasonably predict the transition of combustion characteristics based on injection timings, covering from PCCI-like combustion to diffusive combustion. Figure 11 shows the simulation results for brake thermal efficiency and heat-loss ratio for various compression ratios. The figure indicates that changes in compression ratio between 18 and 14 can hold brake thermal efficiency at almost the same level, just by selecting the appropriate injection timing. This is because even if the compression ratio is decreased within this range, the deterioration of the theoretical thermal efficiency can be canceled by reducing heat-loss. Consequently, a compression ratio of 14 was adopted in this study, which is the minimum ratio to keep the brake thermal efficiency at the same level as a conventional engine with an ordinary compression ratio.