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Research Article

Study on Premixed Combustion in a Diesel Engine with Ultra-Multihole Nozzle

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This study proposed a new low-temperature premixed combustion mode to achieve the simultaneous reduction of NO_x and soot emissions in a volume production diesel engine of CA6DF by reconstructing key systems. Some developments of this diesel engine are as follows. A straight port and large diameter combustion chamber of a low compression ratio was developed. Inlet ports of a high induction swirl ratio were developed. A cooled EGR was developed. Especially, an ultra-multihole (UMH) nozzle was developed. It has two layers of injection holes and a large flow area. Two sprays of the upper and under layers meet in the space of the combustion chamber. The results showed that the operation range of this diesel engine to achieve the better low-temperature premixed combustion is as follows. The speed can cover from the idle speed to the rated speed. The load can reach to 50% of the full load of the corresponding external characteristics speed. The NO_x and soot emissions of this operation range are simultaneously largely reduced, even by 80%–90% at most test cases, while keeping the brake-specific fuel consumption (BSFC) from being significantly deteriorated.

1. Introduction

The conventional diesel engine combustion is dominated by the heterogeneous mixture diffusion combustion. The flame front is the diffusion flame with the stoichiometric equivalence ratio mixture. Thus, the maximum combustion temperature is very high, which is beneficial to NO_x generation. The inherent characteristic of the conventional diesel engine combustion determines the existence of a minimum NO_x emissions value [1]. And the occurrence of locally low oxygen concentration zones is a precursor for the formation of soot. Therefore, the ideal excess air coefficient λ of the conventional diesel premixed combustion should meet with $0.6 < \lambda < 0.8$, or $\lambda > 1.5$ in order to prevent the formation of soot and a large amount of NO_x emissions [2].Thus, it is necessary to explore a new way to change a heterogeneous diesel-air mixture, stoichiometric equivalence ratio mixture, and diffusion flame properties in order to meet increasingly stringent emissions regulations. The low-temperature premixed combustion is just a new combustion mode that can reduce NO_x and soot emissions simultaneously [3–11]. In recent years, homogeneous charge compression ignition (HCCI) combustion, umbrella spray combustion, modulated kinetic combustion, multiple stage diesel combustion, homogeneous charge diesel combustion, multiple pulse injection HCCI combustion, and low-temperature combustion, all belong to this category. Their common target is to achieve a low-temperature premixed combustion by forming a lean and highly-mixed or partially mixed premixed fuel-air mixture prior to ignition. All these new combustion ways can be generalized into two categories, namely, HCCI and PCCI (premixed charge compression ignition) combustion.

The first category is HCCI combustion. It can achieve a chemical and physical homogeneous mixture prior to ignition by an early fuel injection. HCCI combustion is close to the ideal constant volume Otto cycle. Not only it is close to an instantaneous combustion, but there is no diffusion flame. Thus, NO_x and soot emissions are very low. The homogeneous mixture results in negligible soot emissions, and low temperature leads to ultra-low NO_x emissions. There are two typical stages in HCCI combustion. The first stage is called the cold combustion or low-temperature oxidation reaction [12] when the long carbon chains begin to break. The second stage is the succeeding main combustion which is called the high temperature oxidation reaction. The cold combustion leads to the high-temperature in the cylinder until the main combustion starts (the second obvious heat release peak) for the high air-fuel ratios (AFRs). EGR is a way to decrease the oxygen concentration and to control the burning rate by thermal storage, especially when the mixture is close to the stoichiometric equivalence ratio at high loads.

The second category is PCCI combustion. PCCI combustion is distinguished from HCCI combustion since PCCI involves in-cylinder injection and features significant equivalence ratio gradients with the fuel and air mixture. Fuel in PCCI combustion usually is directly injected in the combustion chamber in the vicinity of top dead centre. This injection strategy prevents fuel from adhering to the cylinder wall compared with that of HCCI combustion [14]. The low temperature combustion that is achieved by a high EGR level leads to low NO_x emissions, and the premixed (or partially premixed) combustion that is achieved by prolonging the ignition delay leads to low soot emissions. However, the ability to reduce NO_x and soot emissions of PCCI is slightly weaker, HC and CO emissions are lower, and the requirements for uniformity of a fuel-air mixture and the preparation of a lean mixture are not stringenter compared with those of HCCI. PCCI is usually more suitable for diesel fuel engines [15], and not dependent on an early fuel injection. The early fuel injection is easy to wet the liner wall or combustion wall when the piston is still in a low position in the cylinder, which results in the inevitable increase of HC and CO emissions. Moreover, HCCI of an early fuel injection can not control the ignition timing. However, it would become possible to control the ignition timing by PCCI of a late fuel injection.

Figure 1 shows NO_x-soot maps [13] of the conventional diesel engine, PCCI and HCCI combusiton, with the combustion temperature as the horizontial axis and the mixture equivalence ratio as the vertical axis. The lower right of the figure is the high NO_x concentration area, and the corresponding upper left is the high soot concentration area. The different combustion ways lead to different NO_x-soot emissions generated areas. The NO_x and soot emissions of the conventional diesel combustion are relatively high, and there is a trade-off relationship between them. The fuel-air mixture concentration in the combustion chamber is very unevenly distributed in a wide range because the combustion begins before the end of the fuel injection. Although the whole air-fuel ratio is generally relatively high, there are still existing a rich mixture in the local areas and a lot of stoichiometric equivalence ratio mixture. HCCI combustion, however, begins with a homogeneous lean mixture to avoide the high NO_x and soot concentration generated areas at the same time. Therefore, NO_x and soot emissions can be reduced to very low values. The fuel-air mixture of PCCI, however, is not as lean and homogeneous as that of HCCI.

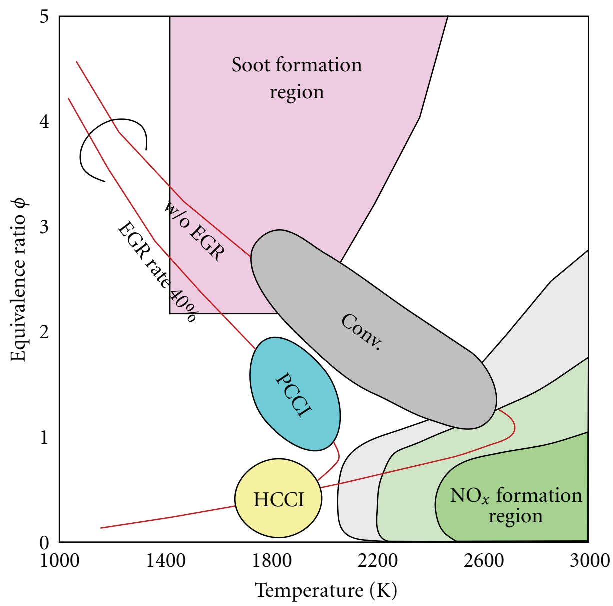


Figure 1: NO_x-soot map of conventional Diesel, PPCI, and HCCI combustion [13].

But it meanwhile avoids the high NO_x and soot concentration generated areas compared with the conventional diesel combustion. Thus, NO_x and soot emissions of PCCI are very lower than those of the conventional diesel combustion, but slightly higher than those of HCCI.

Although the low-temperature premixed combustion can simultaneously reduce NO_x and soot emissions, there are several problems to be solved before it is applied in practice [16, 17].

(1) *Ignition and Combustion Phase Control.* The ignition and combustion phase must be controlled in order to control the maximum combustion pressure and avoid the combustion efficiency from reducing.

(2) *The Preparation Method and Quality of the Premixed Mixture.* It is critical to form a homogeneous or relatively homogeneous premixed mixture before ignition for achieving the better premixed combustion.

(3) *Premixed Combustion Load Adaptability.* Determining how to make the premixed combustion extend to high loads without causing a knocking combustion and how to make it expand to low loads without causing a flame out is one of key issues.

(4) *The High HC and CO Emissions.* The existing premixed combustion has higher HC and CO emissions than those of the conventional combustion because of an early fuel injection in HCCI and a high level of EGR in PCCI combustion.

This study wants to explore a new way of the low-temperature premixed combustion to simultaneously largely reduce NO_x and soot emissions in a diesel engine, and tried to overcome these drawbacks in the existing premixed combustion modes of diesel engines as stated above. Its main ideas are as follows.

(1) The low-temperature combustion that is achieved by a high level of EGR largely reduces NO_x emissions. Moreover, the cooled EGR is beneficial to achieving the premixed combustion due to prolonging the ignition delay. Therefore, it can also control the increase of soot emissions.

(2) The premixed combustion that is achieved by combing the UMH nozzle with a high injection pressure reduces soot emissions. The total flow area of the UMH nozzle is much larger than that of the conventional nozzle. Moreover, compared with the conventional nozzle, the holes diameter of the UMH nozzle can be decreased while the flow area is large. The UMH nozzle has two layers of holes. Two sprays of the upper and under-layer meet in the space of the combustion chamber. This avoids fuel sprays impingement on the wall of the combustion chamber or the cylinder liner, which can control HC emissions.

(3) The premixed combustion of a controllable combustion phase that is achieved by using a late fuel injection avoids a knocking combustion. The combustion begins at the downward stage of the piston because the fuel injection is near the top dead center. Therefore, the maximum cylinder pressure can not rise too high and a knocking combustion can be avoided even in the high load.

2. Experimental Setup

The key systems (intake and exhaust systems, combustion system, fuel injection system, etc.) of a volume production diesel engine have been reconstructed according to the design ideas of the premixed combustion. The test systems of the cylinder pressure, exhaust, and other performance have been established. At last, the test bench of the diesel premixed combustion has been built.

2.1. The Engine. The original engine used in this experiment is a commercial vehicle diesel engine, and its key specifications are shown in Table 1.

2.2. Inlet Port. Ricardo swirl ratio [19] (swirl ratio = charge swirl speed at end of induction/engine crankshaft speed) of the original inlet port is 2.8. Generally, the induction swirl ratio value should be matched with the fuel injection pressure and the number of nozzle holes in order to achieve the better performance in diesel engines. The induction swirl ratio is inversely proportional to the number of nozzle holes when other parameters keep unchanged in the conventional diesel engine, that is, the induction swirl ratio must be reduced when the number of nozzle holes increases. Otherwise, the adjacent sprays may be overlap to form a rich mixture when the swirl ratio is too high, which can lead to an

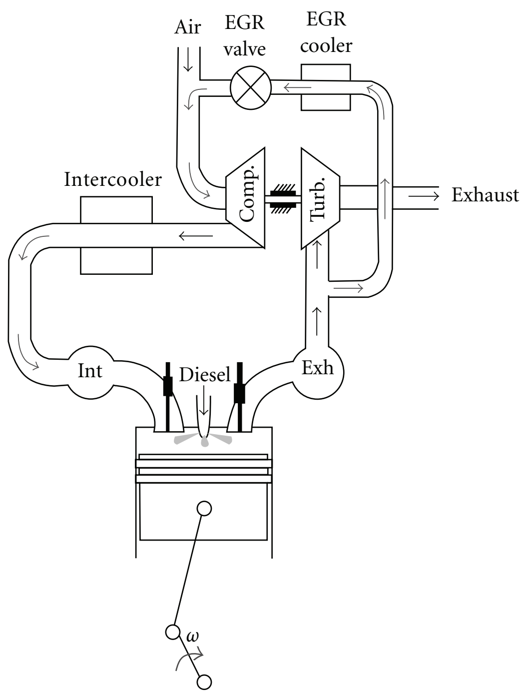


Figure 2: Cooled EGR system [18].

incomplete combustion. A low induction swirl ratio can be used when a fuel injection pressure is high. However, a large amount of EGR and a substantialfuel injection delay should be used in order to achieve the low-temperature premixed combustion in this research, which results in the adverse consequences of HC, CO, brake-specific fuel consumption (BSFC), and so forth. Therefore a high induction swirl ratio must be used to accelerate the fuel-air mixing and promote the combustion, which can eliminate these adverse consequences [3, 4].

At last, two kinds of inlet ports were developed. Their Ricardo swirl ratios are 3.6 and 4.5, respectively, and the corresponding flow coefficients are 0.298 and 0.27, respectively. The induction swirl ratio of two inlet ports is much higher than that of the original inlet pot.

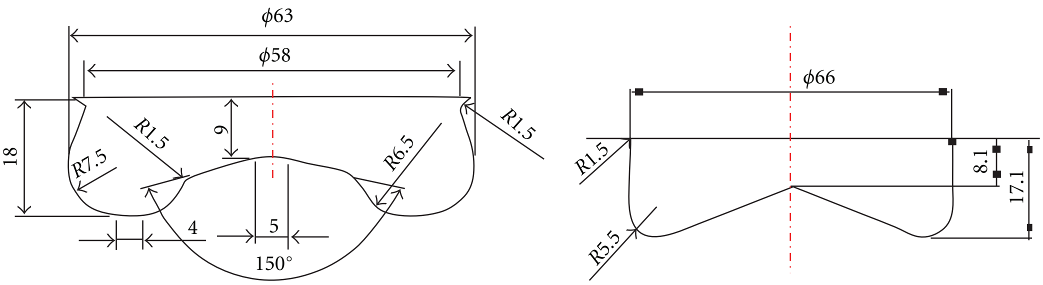
2.3. EGR System. The low pressure EGR system can acquire a high level of EGR as shown in Figure 2 [18]. But the point that the exhaust gas takes is before the turbine in order to achieve a much higher EGR rate in this research. Two CO_2 gas analyzer units are used to test CO_2 volume concentrations of the exhaust gas and the mixture of the fresh charge and exhaust gas, respectively, in the experiment. Then, a relative EGR rate of the fresh charge is calculated with the following formula:

$$\mathrm{EGR}(\%) = \frac{CO_{2,I} - CO_{2,A}}{CO_{2,E} - CO_{2,I}}, \qquad (1)$$

where E, I, and A denote exhaust gas, inlet gas, and atmosphere, respectively. CO_2 volume concentration of the atmosphere is generally 0.04%.

Table 1: Specifications of the original engine.

Model	CA6DF2
Type	In-line, turbocharged, intercooled
Cylinder number − Bore × Stroke (mm)	6 − 110 × 115
Rated Power/Speed (kW/r/min)	155/2300
Maximum Torque/Speed (N·m/r/min)	680/1400
Minimum Stable Speed and Torque (N·m/r/min)	580∼620/1000
Minimum BSFC (g/kW·h)	205
Emission Standard	EURO II
Combustion Chamber	ω Reentrant
Ricardo Swirl Ratio/flow coefficient of Inlet Port	2.8/0.32



(a) Combustion chamber of original engine

(b) Combustion chamber of improved design

Figure 3: Combustion chamber schematic.

2.4. Combustion Chamber. In order to achieve the low-temperature premixed combustion, the improved design guideline of a combustion chamber is as follows.

(1) *Reducing the Compression Ratio Appropriately.* The low compression ratio can prolong the ignition delay, which is beneficial to achieving the premixed combustion.

(2) *Using a Large Diameter and Straight Port Combustion Chamber.* It is necessary to use a large diameter combustion chamber to prevent the fuel spray from hitting the combustion chamber wall when the fuel injection pressure is high, which is beneficial to controlling HC emissions. In addition, the combustion chamber shape should be favorable for the rapid spread of the fuel into the top area of the combustion chamber when the fuel injection timing is delayed to near TDC, which promotes the rapid mixing of the fuel and air to accelerate the combustion. The simulation showed the straight port combustion chamber has this characteristic compared with the original combustion chamber. Therefore, a large diameter and straight port combustion chamber is developed.

The combustion chamber of the original engine has a clear squish lip, and the compression ratio is 17.7, while the compression ratio of the improved combustion chamber with a straight port shape is 16.5 as shown in Figure 3.

2.5. Fuel Injection System. The design guideline of the fuel injection system is to shorten the fuel injection duration in order to achieve the premixed combustion. Therefore, a high pressure common rail injection system is used to replace the original mechanical in-line pump, and the UMH nozzle with a large flow area is used to replace the conventional nozzle.

2.5.1. UMH Nozzle. It is critical to develop a UMH nozzle in this research. The necessary prerequisite for achieving a premixed combustion is injecting all the fuel into cylinder prior to ignition. Therefore, the design guideline of the UMH nozzle is that the total flow area should be much larger than that of the conventional nozzle in order to shorten the fuel injection duration. And holes diameter should be decreased as possible as it can in order to ensure the spray quality. The developed UMH nozzle is shown in Figure 4 [20].

It includes needle and needle body. There are two layers of nozzle holes in the top of needle body. The injection holes cone angle (here defined as the angle of cone consisting of all the axes of injection holes on the same layer) of the under-layer holes is larger than that of the upper-layer holes ($\alpha 2 > \alpha 1$, half of the angle difference is defined as the impingement angle $\beta = (\alpha 2 - \alpha 1)/2$). On the one hand, the design of two layers of nozzle holes greatly increases the total flow area of the nozzle holes. This makes it possible to inject all the fuel into cylinder before ignition. Therefore, it creates a necessary prerequisite for achieving the premixed combustion. On the other hand, two sprays from the upper layer and the under layer meet in the space of the combustion chamber because the $\alpha 2$ is larger than the $\alpha 1$. This not only avoids fuel sprays impingement on the wall of the combustion chamber or the cylinder liner, but also strengthens sprays turbulence, which promotes fuel-air mixing. The UMH

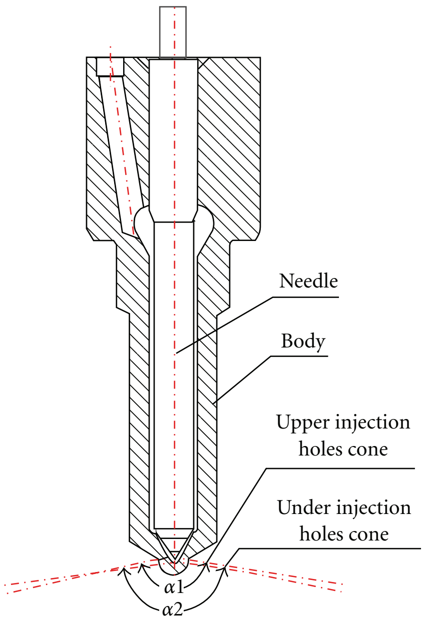


FIGURE 4: Schematic of UMH Nozzle [20].

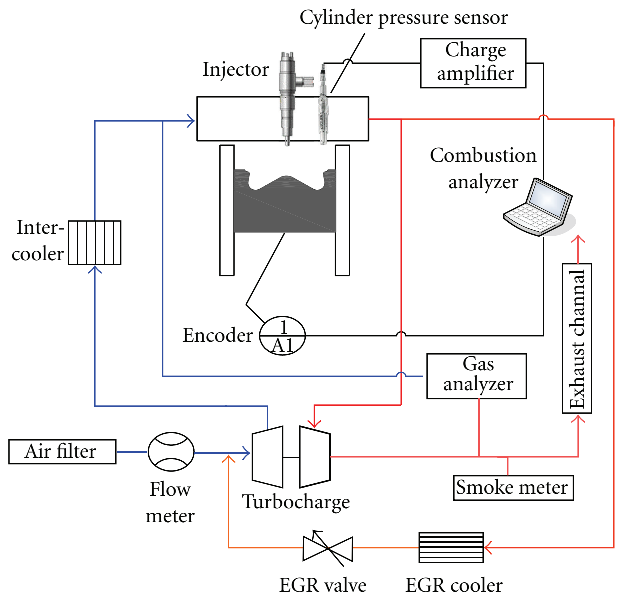


FIGURE 6: Experimental apparatus.

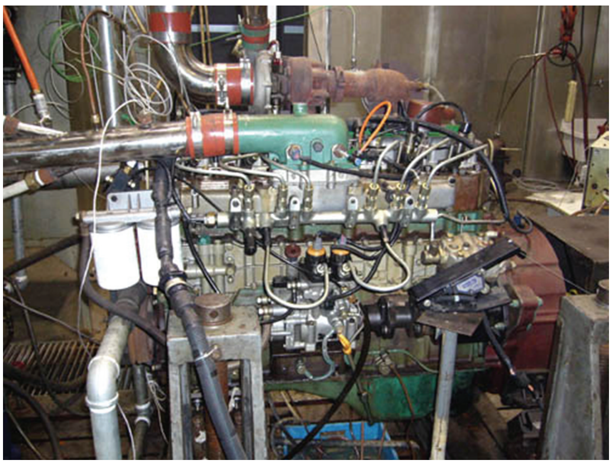


FIGURE 5: Reconstructed test engine.

nozzle exhibits shorter spray penetration and bigger spray angle resulting from sprays interaction than the conventional nozzle, which helps prepare a more homogeneous mixture [20]. The specifications of the designed UMH nozzles are shown in Table 2.

2.5.2. High-Pressure Common Rail Fuel Injection System. A common rail fuel injection system replaces the original mechanical in-line fuel injection system. The fuel injection pressure and injection timing of the common rail system can be flexibly adjusted at any operation condition, which provides very convenient ways to investigate the diesel engine premixed combustion.

2.6. Reconstructed Test Engine. The reconstructed test engine is shown in Figure 5, while the schematic of the experimental apparatus is shown in Figure 6. The EGR cooler is arranged in the EGR tube between the upstream of the exhaust gas turbine and the inlet of the compressor. Both the intercooler and EGR cooler are water cooled, and the cooling capacity can be set by adjusting the circulating water.

2.7. Experimental Apparatus. The cetane number of diesel fuel is 51. The coolant temperature is set to 80 ± 3°C. The inlet air temperature after the intercooler is maintained at 40 ± 3°C during the whole experiment. The exhaust gas is measured by HORIBA MEXA-7100 gas analyzer, the smoke by AVL 415S smoke meter, and the cylinder pressure by KISTLER pressure transducer. Table 3 lists the main equipments and instruments in the engine test.

3. Research Methodology

The premixed combustion of the test diesel engine with the UMH nozzle focuses on the premixed combustion mechanism and the final exhaust gas level. The main technical measures are as follows. A large number of EGR is used to achieve the low-temperature combustion in order to reduce NO_x emissions. The UMH nozzle is used to replace the conventional nozzle, and it is combined with the high injection pressure to shorten the injection duration, which facilitates the premixed combustion in order to reduce soot emissions. The fuel injection timing should be delayed to near TDC, which is possible to achieve a controllable combustion phase in order to avoid a knocking combustion and deteriorating the combustion efficiency. Therefore, the main research methods of the UMH nozzle diesel engine premixed combustion experiment are as follows.

Table 2: Specifications of nozzles.

Type		Number of layers/number of holes per layer	Nozzle holes diameter (mm)	Total flow area of nozzle holes (mm^2)	cone angle $\alpha 1/\alpha 2$ (°)	Impingement angle β (°)
Original diesel nozzle		1/8	0.17	0.1815	150	/
UMH nozzle	Case 1	2/7	0.17	0.3167	148/156	4
	Case 2	2/8	0.16	0.3215	150/150 146/154 140/170	0, 4, 15

Table 3: Specifications of the experimental apparatus.

Name		Type	Precision	Manufacture company
Hydraulic dynamometer		WE41N	±0.2%	Institute of Electromechanical Technology, Hangzhou Yike
Hangzhou Yike Monitor	Mass of intake air	EIM 030ID	0.8%	Hangzhou Yike Electromechanical Technology Co., Ltd
	Torque		±0.3%	
	Revolution speed		±0.1%	
	Exhaust gas temperature		±2%	
	Pressure		±0.2%	
Fuel meter		7351CST	0.8%	AVL Company in Austria
Wet and dry thermometer		272-A	±2%	Shanghai Medical Instrument Factory
AVL smoke meter		415S	±2%	AVL Company in Austria
AVL combustion analyzer		620	—	
Gas analyzer		MEXA-7100 DEGR	Meet GB17691-1999 Requirements	HORIBA Company in Japan
Cylinder pressure transducer		6125B	−16pC/bar	KISTLER Company in Sweden
Mass flow meter		Sensyflow	±1.5% FS	SENSYCON Company

(1) Based on the combustion analysis, compare the UMH nozzle diesel engine premixed combustion with the conventional combustion to conclude the mechanism of the premixed combustion.

(2) Based on the measurement and analysis of the exhaust gas, compare the UMH nozzle diesel engine premixed combustion with the conventional combustion to obtain a relationship between NO_x and soot emissions.

3.1. Combustion Analysis. Figure 7 shows the cylinder pressure, heat release rate, fuel injection rate, and several important parameters. The angle when the fuel starts to inject into the chamber is defined as the injection starting point θ_1 (°CA ATDC). The angle when the fuel injection closes is defined as the injection end point θ_2 (°CA ATDC). The angle when the ignition starts is defined as the combustion starting point θ_3 (°CA ATDC). The interval angle (θ_3-θ_1) between the injection starting point and the combustion starting point is defined as the ignition delay τ_i (°CA). The interval angle (θ_2-θ_1) between the fuel injection close and the start is defined as the injection duration ϕ_z (°CA). The interval angle (θ_3-θ_2) between the combustion starting point and the injection close point is defined as the premixed degree duration τ_{pmix} (°CA).

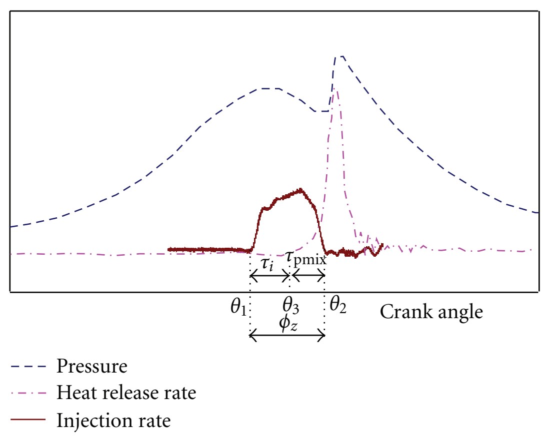


Figure 7: Definition of combustion parameters.

Here, the premixed degree of the fuel-air mixture is quantitatively analyzed with the interval angle between the combustion starting point and the injection close point. This angle is defined as the premixed degree duration

(τ_{pmix}) of the fuel-air mixture [21]. When the τ_{pmix} value is negative, it indicates that not all the fuel is injected into the cylinder before ignition. It is impossible to achieve a complete premixed combustion because some of the fuel is still of a diffusion combustion. Therefore, it also belongs to the conventional combustion. When the τ_{pmix} value is zero, it indicates that all the fuel has just injected completely into the cylinder before ignition. But it is also impossible to achieve a better premixed combustion because there is little time left for the later injected fuel to get mixed with air. Only when the τ_{pmix} value is positive, the bigger value allows the longer time for fuel-air mixing to be able to achieve a more homogeneous premixed combustion. It can form a homogeneous lean-mixture if the excess air coefficient of the operating condition is big. Therefore, NO_x emissions are reduced due to the low maximum combustion temperature, and soot emissions are also reduced due to the homogeneous lean mixture. Thus, the premixed degree duration τ_{pmix} of the fuel-air mixture defined above is a very important parameter that plays an important role in comparing the conventional combustion and the premixed combustion.

3.2. Measurement and Analysis of Exhaust Gas. It is necessary to measure HC, CO, NO_x, soot emissions, BSFC, and so forth in the engine test. Besides, the analysis of the heat release rate is carried out based on the cylinder pressure.

The experiments include the original engine combustion and the diesel engine premixed combustion of different UMH nozzles, combustion chambers, and inlet ports. The fuel injection timing, injection pressure, and EGR rate are varied in order to obtain a better performance during the whole experiment.

There is a trade-off relationship between NO_x and soot emissions in the conventional combustion. This research aims at achieving a low-temperature premixed or partially premixed combustion to simultaneously reduce NO_x and soot emissions by adjusting the EGR rate, injection timing, injection pressure, UMH nozzle, and other parameters.

4. Results and Analyses

Table 4 shows the four selected operating conditions in the whole experiment. The excess air coefficients of these selected operating conditions are different because their speed and load are different. Therefore, in the whole experiment, the needed EGR rate, injection pressure, and injection timing of these operating conditions are different in order to achieve their respective lower NO_x and soot emissions and avoid a significant deterioration of HC, CO emissions, and BSFC performance.

4.1. Effect of Combustion Chamber on Premixed Combustion. The premixed combustion performances of the original reentrant combustion chamber are compared with those of the straight port combustion chamber (as shown in Figure 3) in the test. The UMH nozzle of 1616-4 (1616-4 means the hole number of 16, hole diameter of 0.16 mm, and the impingement angle of 4°) and the inlet port of 3.6 Ricardo swirl ratio are used. The A and B of Table 4 are set to test cases. The engine performances of two different combustion chambers are optimized by adjusting the injection pressure, injection timing, and EGR rate in order to achieve their respective lowest NO_x and soot emissions, meanwhile avoid a significant deterioration of HC, CO emissions, and BSFC.

Table 4: Test operating conditions.

Operating condition	Speed (r/min)	Torque (N·m)	Mean effective pressure* (MPa)
A	1000	62	0.119
B	1000	155	0.297
C	1000	215	0.412
D	1400	300	0.575

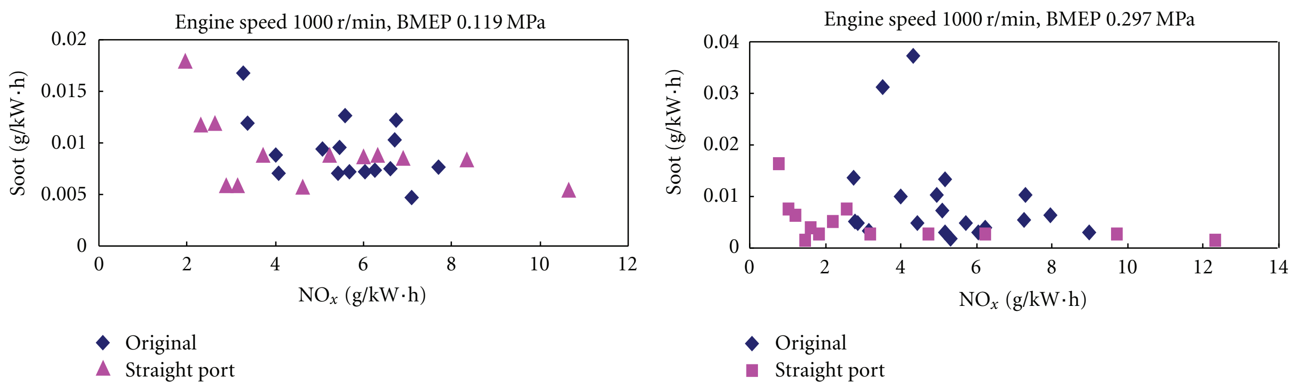
* Mean effective pressure = torque × τ/(318.3 × Vs).
Where τ means stroke number, four stroke is 4 and two stroke is 2.
Vs means displacement (l).

The straight port combustion chamber can simultaneously largely reduce NO_x and soot emissions, compared with the original reentrant combustion chamber in both A and B operating conditions as shown in Figure 8.

The main differences between the original combustion chamber and the straight port combustion chamber are the compression ratio and structure types. Compared the straight port combustion chamber with the original combustion chamber, the low compression ratio can prolong ignition delay and the large diameter can use a high injection pressure to shorten fuel injection duration, which is beneficial to achieving a better premixed combustion. Table 5 shows combustion parameters of two different combustion chambers that correspond to the lowest point of NO_x and soot emissions of Figure 8 in both A and B operating conditions. The τ_{pmix} value of the original combustion chamber is negative as shown in Table 5, which indicates that the ignition starts before all the fuel is injected into the cylinder. While the τ_{pmix} of the straight port combustion chamber in A and B operating conditions are 5.1° and 4.1°, respectively, which leaves long time for fuel-air to mix prior to ignition. Therefore, it can achieve a better homogeneous premixed combustion. In addition, the large diameter of the straight port combustion chamber can use a high injection pressure, so it is possible to use a high level of EGR. Consequently, NO_x and soot emissions can be simultaneously reduced substantially compared with the original combustion chamber.

4.2. Effect of Induction Swirl Ratio on Premixed Combustion. In order to achieve a low-temperature premixed combustion in diesel engines, it is necessary for a high induction swirl ratio to enhance the fuel-air mixing when a high level of EGR and a late fuel injection are used [3, 4].

The comparison between the effects of three inlet ports of different swirl ratios (Ricardo swirl ratios are 2.8, 3.6, and 4.5 resp.) on the diesel engine premixed combustion is carried out. The 1616-4 UMH nozzle and straight port combustion chamber (as shown in Figure 3) are used. The B and D operating conditions of Table 4 are selected to test cases.



(a) A operating condition

(b) B operating condition

FIGURE 8: Effect of combustion chamber on NO_x and soot emissions.

TABLE 5: Effect of combustion chamber on combustion parameter.

Operating condition	Type of combustion chamber	Intake boost pressure KPa	Injection pressure MPa	EGR rate %	Excess air coefficient λ	Injection starting point θ_1 °CA ATDC	Injection end point θ_2 °CA ATDC	Combustion starting point θ_3 °CA ATDC	Ignition delay $\theta_3 - \theta_1$ °CA	τ_{pmix} $\theta_3 - \theta_2$ °CA	*CA50 °CA ATDC
A	Original	3.5	55	10	3.55	0	4.5	4.4	4.4	−0.1	17.6
	Straight port	0	90	80	2.81	−1.5	3	8.1	9.6	5.1	16.2
B	Original	7.8	80	15	1.88	3	8.6	7.5	4.5	−1.1	25
	Straight port	0	110	80	1.68	−1.5	4.1	8.2	9.7	4.1	17.9

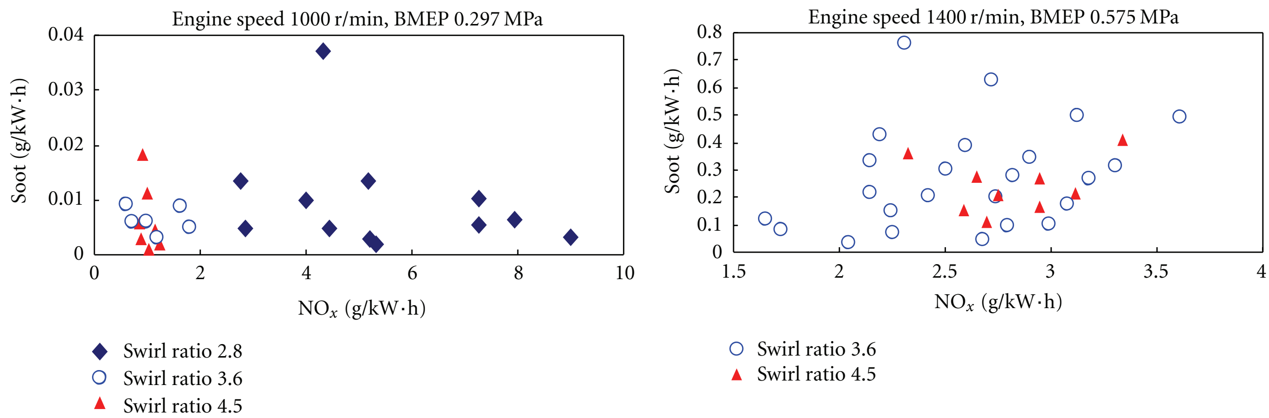
*CA50 denotes the crank angle of 50% total heat release rate.

The engine performances of three inlet ports are optimized by adjusting the EGR rate, injection pressure, and injection timing in order to achieve their respective lowest NO_x and soot emissions, meanwhile avoid a significant deterioration of HC, CO emissions, and BSFC performance.

The results are shown in Figure 9. The specific data corresponding to the lowest NO_x and soot emissions in Figure 9 are shown in Table 6. Table 6 shows that NO_x and soot emissions of 2.8 induction swirl ratio are the worst in the B operating condition. The NO_x and soot emissions of 3.6 induction swirl ratio are close to those of 4.5 induction swirl ratio. The HC, CO emissions, and BSFC of 3.6 induction swirl ratio are better than those of 4.5 and 2.8 induction swirl ratios. For the D operating condition, NO_x and soot emissions of 3.6 induction swirl ratio are better than those of 4.5 induction swirl ratio. The HC, CO emissions, and BSFC of 3.6 induction swirl ratio are also better than those of 4.5 induction swirl ratio. This reason is that adjacent sprays may be overlap to form a rich mixture when the swirl ratio is too high, which can lead to an incomplete combustion. But the performance for 2.8 induction swirl ratio is very bad at condition D. Especially HC and CO emissions are too high to test.

4.3. Effect of UMH Nozzle Parameters on Premixed Combustion. There are three variable parameters in the UMH nozzle, including the holes diameter, holes number, and impingement angle. The general requirement for the UMH nozzle design is that the flow area should be larger than that of the original nozzle in order to inject all fuel into cylinder before ignition. The large diameter and increase number of holes can obtain a large flow area. But the holes diameter should be smaller than that of the original nozzle to ensure a better spray atomization and a smaller SMD (Sauter mean diameter) of the spry droplets. The increase number of holes is limited by the nozzle structure and should be matched with the induction swirl ratio. In general, nozzle parameters have a significant effect on engine performances [22, 23]. The effect of the UMH nozzle on engine performances is more complicated because the structure of the UMH nozzle is more complicated than that of the conventional nozzle. Therefore, the effect of the UMH nozzle on the diesel engine premixed combustion was mainly investigated by experiments.

4.3.1. Effect of Impingement Angel. The effects of the UMH nozzle of three different impingement angles on the diesel engine premixed combustion are compared in the tests. The impingement angles are 0°, 4°, and 15°, respectively. The 1616 type UMH nozzle, the inlet port of 3.6 Ricardo swirl ratio, the straight port combustion chamber and B operating condition of Table 4 are selected in the tests. The engine performances of three different impingement angles are optimized by adjusting the EGR rate, injection pressure and



(a) B operating condition

(b) D operating condition

Figure 9: Effect of swirl ratio on NO_x and soot emissions.

Table 6: Effect of swirl ratio on engine performance.

Operating condition	Swirl ratio	Injection pressure MPa	Injection starting point °CA ATDC	EGR rate %	Soot g/kW·h	NO_x g/kW·h	HC g/kW·h	CO g/kW·h	BSFC g/kW·h
B	2.8	80	3	15	0.0049	2.86	14.05	24.36	308.0
	3.6	110	−1.5	80	0.0012	1.49	5.08	20.03	274.8
	4.5	110	−2	40	0.0009	1.04	5.20	20.21	277.9
D	3.6	110	2	28	0.0402	2.48	0.53	10.15	253.4
	4.5	110	2	28	0.1081	2.69	0.62	11.86	269.8

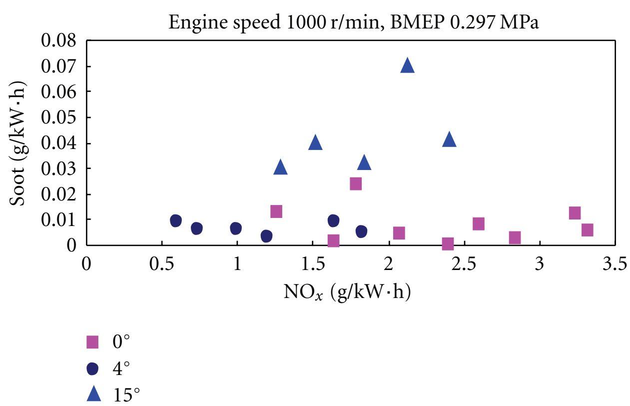


Figure 10: Effect of the impingement angle on NO_x and soot emissions.

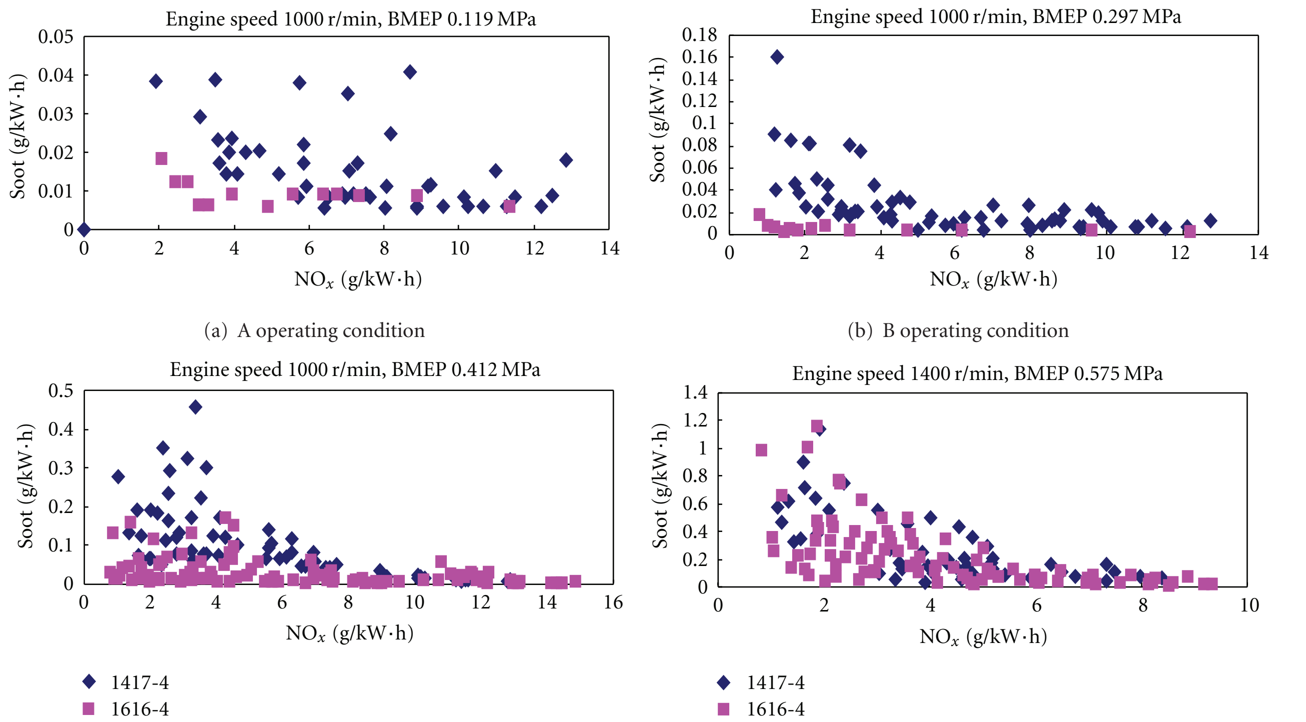
injection timing in order to achieve their respective lowest NO_x and soot emissions, meanwhile avoid a significant deterioration of HC, CO emissions, and BSFC.

The variation of NO_x and soot emissions with the impingement angles is shown in Figure 10. It can be seen that the UMH nozzle of 4° impingement angle can simultaneously achieve the lowest NO_x and soot emissions, while the UMH nozzle of 15° impingement angle is the worst.

4.3.2. Effect of Holes Number and Diameter. The 1417-4 (holes number is 14, holes diameter is 0.17 mm, and impingement angle is 4°) and 1616-4 (holes number is 16, holes diameter is 0.16 mm, and impingement angle is 4°) UMH nozzles, the inlet port of 3.6 Ricardo swirl ratio, the straight port combustion chamber, and the A, B, C, D operating conditions of Table 4 are selected in the tests. The engine performances of two different UMH nozzles are optimized by adjusting the EGR rate, injection pressure and injection timing in order to achieve their respective lowest NO_x and soot emissions, meanwhile avoid a significant deterioration of HC, CO emissions, and BSFC. It can be seen form Figure 11 that in all the four operating conditions, NO_x and soot emissions of the 1616-4 UMH nozzle are much better than those of the 1417-4 UMH nozzle.

The main performance differences of the two UMH nozzles are caused by differences in the holes number and diameter. The small holes diameter can obtain the better spray atomization compared 1616-4 nozzle with 1417-4 nozzle, which is beneficial to forming more homogeneous mixture. It can be inferred from Figure 11 that the nozzle of 16 holes is more suitable to the inlet port of 3.6 Ricardo swirl ratio than that of 14 holes.

4.4. Performance Comparison between Diesel Engine with UMH Nozzle Premixed Combustion and Original Engine Combustion. Figure 12 shows the NO_x and soot emissions comparison between the diesel engine with the UMH nozzle premixed combustion and the original engine combustion in the four operating conditions of A, B, C, and D. The 1616-4 type UMH nozzle, the inlet port of 3.6 Ricardo swirl



(a) A operating condition

(b) B operating condition

(c) C operating condition

(d) D operating condition

FIGURE 11: Effects of UMH nozzle holes number and diameter on NO_x and soot emissions.

ratio, and the straight port combustion chamber (as shown in Figure 3) are selected in the premixed combustion tests.

The original engine performances are tested to provide the comparison base for the UMH nozzle premixed combustion. The diesel engine with the UMH nozzle premixed combustion provides a scatter of data due to using different EGR rates, injection timings, and injection pressures as shown in Figure 12. The specific datum corresponding to the lowest NO_x and soot emissions in Figure 12 are shown in Table 7.

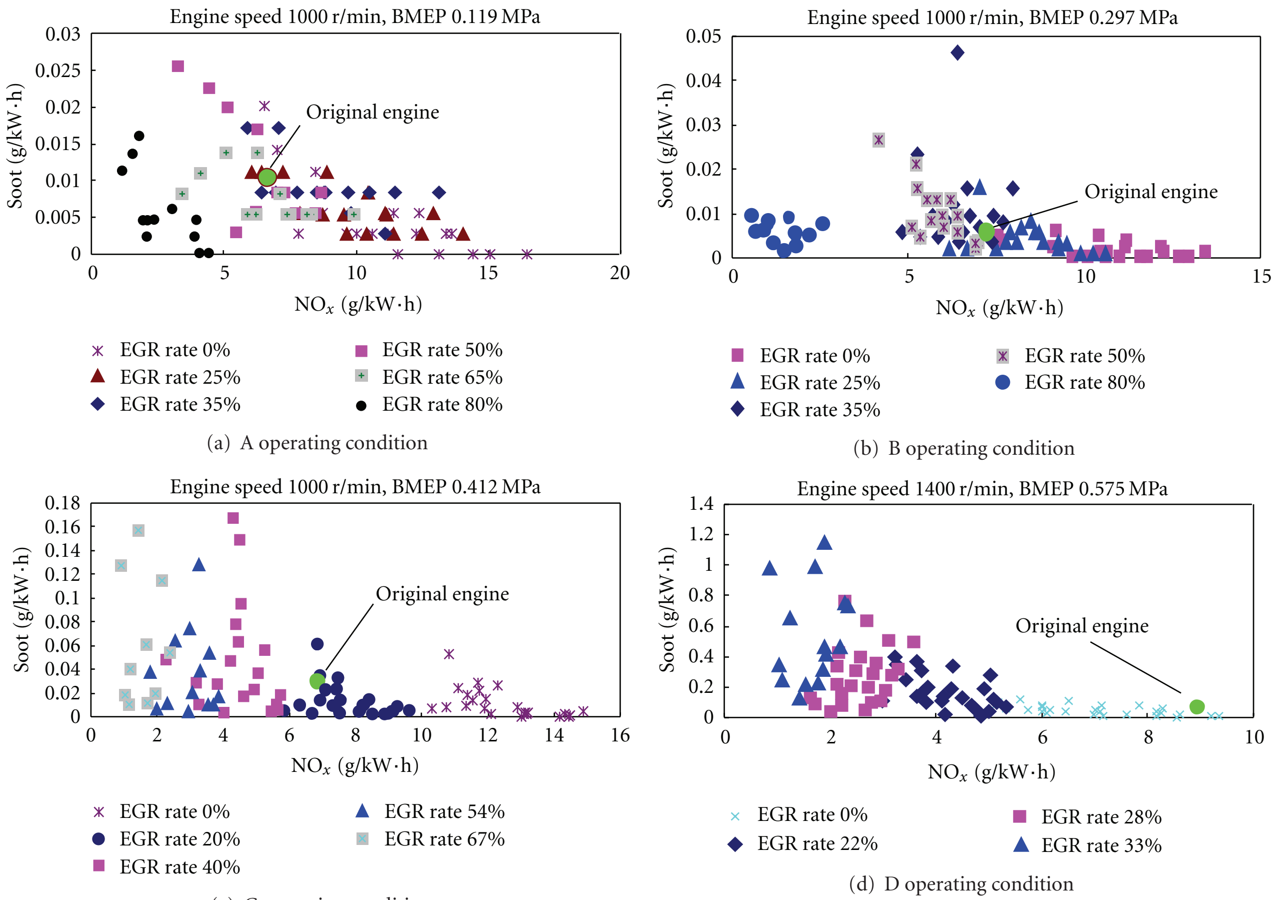
All the fuel can be injected into the combustion chamber before ignition, even under relatively high load operating conditions, because the flow area of the UMH nozzle is much larger than that of the original nozzle. This leaves long time for fuel-air to mix prior to ignition, which provides necessary guarantees for achieving a premixed combustion. Therefore, the more ideal low-temperature premixed combustion is achieved by combining the UMH nozzle with high EGR rate, high injection pressure, and optimal injection timing. It can be seen form Figure 12 that NO_x emissions are very high and soot emissions are very low without EGR. Then NO_x emissions decrease with the EGR rate increasing, but soot emissions increase. The NO_x emissions drop substantially as the EGR rate continuously increases. Meanwhile, the more ideal homogeneous charge premixed combustion is achieved by means of the high injection pressure and optimal injection timing. Therefore, soot emissions drop to very much low values. Eventually, NO_x and soot emissions are simultaneously largely reduced compared with the original engine. The reason is that the τ_{pmix} of all operating conditions becomes long (the τ_{pmix} is about 0.3–5.1°CA as shown in Table 7) due to the combination of the UMH nozzle with EGR, high injection pressure, and optimal injection timing. Thus, it leaves long time for fuel-air to mix before ignition, which allows the fully mixing of fuel-air to achieve the more homogeneous lean-mixture combustion.

The NO_x and soot emissions of the four operating conditions are largely reduced compared with those of the original engine as shown in Table 7. Meanwhile, BSFC has almost no increase in A operating condition, but there is a slight increase in B, C, and D operating conditions. The reason is that A operating condition has the maximum excess air coefficient (which is beneficial to achieving a more ideal homogeneous lean-mixture premixed combustion) and the CA50 value of the closest to the TDC (which is beneficial to improving the combustion efficiency) compared with the other operating conditions. However, HC and CO emissions of the premixed combustion are much higher than those of the original combustion due to the high level of EGR. But, they are lower than those of the original combustion in lower load operating condition when the EGR rate is not much higher according to the previous research (as shown in [20]).

Figure 13 compares the cylinder pressure and heat release rate between the original engine combustion and the premixed combustion in A, B, C, and D operating conditions. It can be seen that the cylinder pressure of the diesel engine with the UMH nozzle premixed combustion is lower than that of the original engine combustion. This is because the premixed combustion has the low compression ratio and the low boost pressure, especially the late fuel injection timing

Table 7: Engine performance comparison between premixed combustion and original engine.

Operating condition	Combustion way	EGR rate %	Excess air coefficient λ	Boost Pressure kPa	Injection pressure MPa	Injection starting point °CA ATDC	Soot g/kW·h	NO_x g/kW·h	HC g/kW·h	CO g/kW·h	BSFC g/kW·h	CA50	τ_{pmix} °CA
A	Conventional combustion	0	4.41	6.3	50	0	0.0103	6.72	9.21	29.71	400.5	15.45	−1.2
	Premixed combustion	80	2.81	0	90	−1.5	0.0059	3.10	14.85	56.44	402.0	16.2	5.1
B	Conventional combustion	0	3.5	9.4	50	0	0.0103	7.28	1.05	6.20	262.2	16.35	−3.3
	Premixed combustion	80	1.68	0	110	−1.5	0.0012	1.49	5.08	21.23	274.8	17.87	4.1
C	Conventional combustion	0	2.75	13	50	0	0.0289	6.89	0.47	2.31	246.5	16.3	−5.1
	Premixed combustion	67	1.4	1.6	120	0	0.0099	1.17	3.79	19.67	263.0	20.3	3.4
D	Conventional combustion	0	2.25	40.2	75	−2	0.0558	8.97	0.58	1.46	243.9	19.0	−6.8
	Premixed combustion	28	1.41	31.4	110	3	0.0291	2.05	1.36	14.19	261.2	20.95	0.3



(a) A operating condition

(b) B operating condition

(c) C operating condition

(d) D operating condition

FIGURE 12: NO_x and soot emissions comparison between premixed combustion and original engine combustion.

TABLE 8: Test operating conditions for premixed combustion.

Speed r/min	Torque N·m	Speed r/min	Torque N·m	Speed r/min	Torque N·m	Speed r/min	Torque N·m	Speed r/min	Torque N·m	Speed r/min	Torque N·m
1000	62	1430	75	1770	70	2110	67	2300	64	750	5
	155		188		175		167		160		
	217		263		245		233		224		
	310		375		350		333		320		

in high load compared with the original combustion. The main difference in the heat release rate is that there is the diffusion combustion in the later period of the original engine combustion. But the heat release rate of the premixed combustion is a single smooth curve, and there is the cold flame combustion with a certain crank angle in the early combustion period, especially in low-speed and low-load operating conditions. In addition, the premixed combustion process is delayed with the load increasing. The fuel-air mixture before ignition is not completely homogeneous because the τ_{pmix} becomes smaller with the load increasing as shown in Table 7, which leads to fast combustion once ignition. Therefore, the maximum heat release rate is much higher than that of the original engine combustion as shown in D operating condition of Figure 15.

4.5. Experimental Investigation on Operation Range of Premixed Combustion. In order to determine the operation range of the better premixed combustion after using the methodology in this research, the selected test speeds include 1430 r/min, 1770 r/min, and 2110 r/min of EURO III emissions tests, idle speed, the minimum operating speed of 1000 r/min in external characteristics, and the rated speed of 2300 r/min. The torques corresponding to the speeds are shown in Table 8.

The torque corresponding to each speed is 10%, 25%, 35%, and 50% of the full load in external characteristics respectively. The purpose is that NO_x and soot emissions simultaneously decrease largely and BSFC increase is not more than about 10% compared with the original engine at any operating condition. The optimized results of the

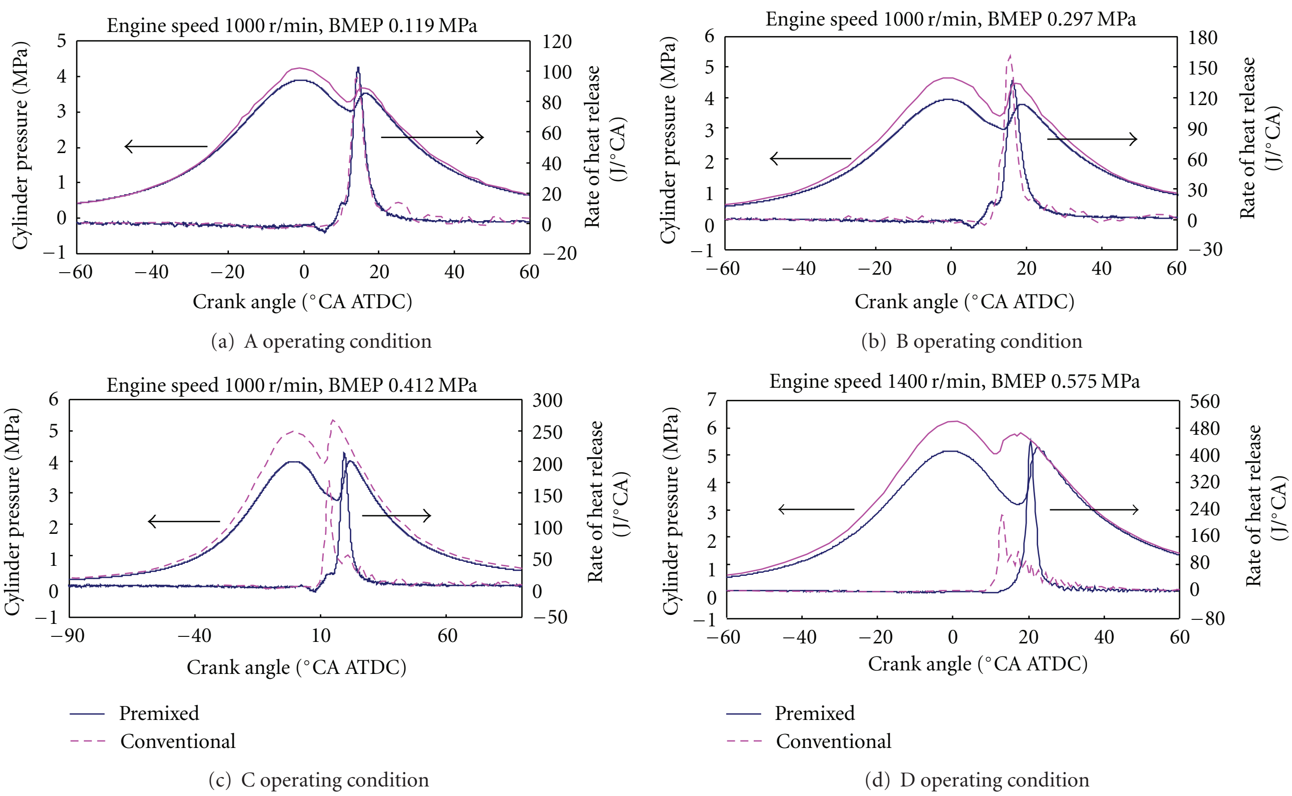
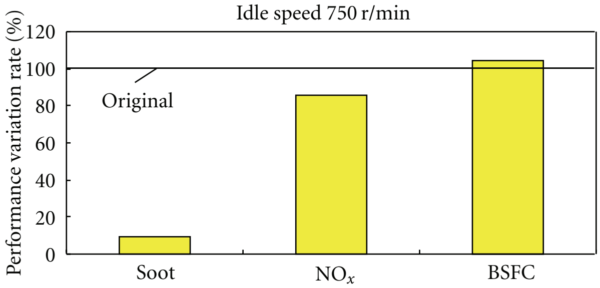


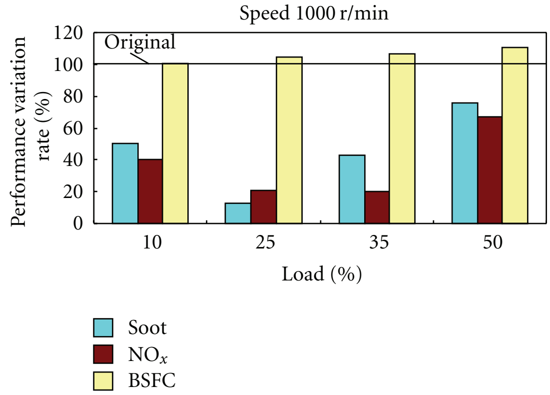
FIGURE 13: Cylinder pressure and heat release rate comparison between premixed combustion and original engine combustion.

TABLE 9: Test result at different test cases.

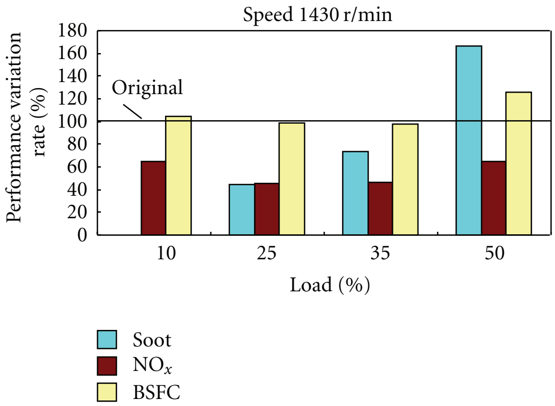
speed r/min	Torque N.m	Load%	EGR Rate%	Injection pressure MPa	Injection starting point °CA ATDC	Excess air coefficient λ	Soot g/kW·h	NO_x g/kW·h	HC g/kW·h	CO g/kW·h	BSFC g/kW·h	CA50 °CA ATDC
1000	62	10	84	90	−1.5	2.81	0.0059	3.10	14.85	56.44	402.0	16.2
	155	25	81	110	−1.5	1.68	0.0012	1.49	5.08	21.23	274.8	17.9
	217	35	49	120	0	1.40	0.0099	1.17	3.79	19.67	263.1	20.3
	310	50	26	120	2.5	1.43	0.0516	4.04	1.07	11.41	252.9	19.4
1430	75	10	72	110	−2	2.53	0.0000	3.36	31.96	57.07	431.6	24.1
	188	25	60	110	−3	1.59	0.0223	1.87	4.10	13.96	269.9	19.6
	263	35	37	110	0	1.50	0.0236	1.91	1.19	8.76	256.7	20.7
	375	50	24	120	3	1.35	0.2831	2.97	0.35	10.75	249.5	21.1
1770	70	10	65	90	−3	2.48	0.0000	3.68	25.15	46.81	470.9	25.2
	175	25	51	100	−1	1.66	0.0069	1.48	3.81	15.98	314.3	24.2
	245	35	45	110	1	1.47	0.0263	1.17	2.69	12.98	299.5	31.9
	350	50	28	110	4.5	1.48	0.0238	2.09	3.11	8.15	281.5	32.8
2110	67	10	69	110	−2.5	2.24	0.0054	3.77	38.17	50.84	553.6	22.8
	167	25	60	110	−2.5	1.58	0.0192	1.10	6.49	17.67	334.2	23.9
	233	35	44	110	0.5	1.54	0.0257	1.40	3.97	11.05	307.9	27.3
	333	50	28	110	4	1.74	0.0317	2.11	0.87	5.27	284.7	29.1
2300	64	10	38	110	−1	2.95	0.0126	7.11	11.91	27.22	574.7	21.2
	160	25	50	120	−1	1.77	0.0137	2.09	5.35	12.62	348.8	22.4
	224	35	42	110	1.5	1.66	0.0219	1.74	2.51	10.42	336.9	29
	320	50	37	120	4	1.53	0.0366	1.49	0.61	8.28	314.5	33.4
750	5	—	38	80	1	5.72	0.0675	28.52	268.27	542.46	3027.9	18



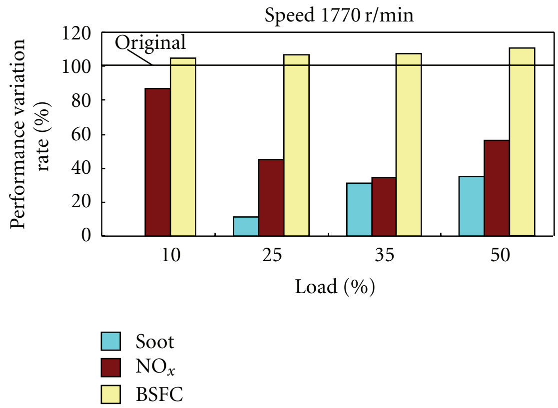
(a) idle speed



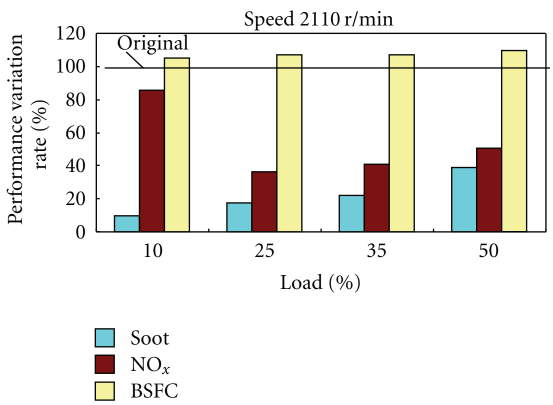
(b) 1000 r/min



(c) 1430 r/min



(d) 1770 r/min



(e) 2110 r/min

FIGURE 14: NO_x, soot, and BSFC of premixed combustion variation rate compared with the original engine at different test operating conditions.

premixed combustion are shown in Table 9, and the performance variation compared with the original engine is shown in Figure 14.

It is impossible to show the comparison of the 2300 r/min between the original and premixed combustion because the original performance was not done. But it can be estimated that both NO_x and soot emissions of the premixed combustion should be reduced largely. It can be seen from Table 9 and Figure 14 that NO_x and soot emissions of all the other operating conditions largely decrease except the operating condition of 1430 r/min and 375 N·m, while BSFC variation is different. In general, BSFC of the low load almost does not increase, or even decreases. The BSFC of the high load increase slightly, and some even reach 10% compared with that of the original engine. The detailed analyses are as follows.

(1) *10% of the Full Load in Operating Conditions at All Speeds.* The decrease of NO_x emissions is not significant compared with the original engine combustion because NO_x

emissions of the original engine at this load are already relatively low. The soot emissions of all the other speeds substantially decrease with the extent of over 90% except the speed of 1000 r/min. The soot emissions are zero to achieve smokeless combustion at 1430 r/min and 1770 r/min. The BSFC increases within 7%, but has almost no increase at 1000 r/min.

(2) *25% and 35% of the Full Load in Operating Conditions at All Speeds.* It can be seen that NO_x and soot emissions significantly decrease compared with the original engine. The maximum values are only 10%–20% of the original values at some operating conditions. The increase of BSFC is about 7%, but it even slightly decreases at 1430 r/min.

(3) *50% of the Full Load in Operating Conditions at All Speeds.* The results showed that the excess air coefficient λ has a low threshold to keep HC, CO, and BSFC from being deteriorated while greatly reduces NO_x and soot emissions. This low threshold is about 1.4, which means the low-temperature premixed combustion was restricted to the low load. But, the excess air coefficient λ of 50% of the full load at 1430 r/min is 1.35 as shown in Table 9, namely, this value is too low to form the better premixed combustion. Therefore, soot emission is higher than that of the original combustion. But NO_x and soot emissions largely decrease at the other speeds and the increase of BSFC is about 10%.

(4) *The Idle Speed.* The NO_x and soot emissions of the idle speed decrease by 18.8% and 91.6%, respectively, and BSFC increases by 4.8%, compared with those of the original engine. This means that it is also possible to achieve the better low-temperature premixed combustion at the idle speed.

Figure 15 shows the operating range of the better low-temperature premixed combustion achieved by combined the UMH nozzle with high level of EGR, high injection pressure, and optimal injection timing. The speed can cover from the idle speed to the rated speed, and the load can reach to 50% of the full load of the corresponding external characteristics speed (except for 1430 r/min). The NO_x and soot emissions of the premixed combustion simultaneously largely decrease even by 80%–90% at most operating conditions in the above-mentioned operating range, compared with those of the original engine. Especially some operating conditions have smokeless. The increase of BSFC is about 10% compared with the original engine, and it does not increase obviously at the low load and even slightly decreases in the certain operating conditions.

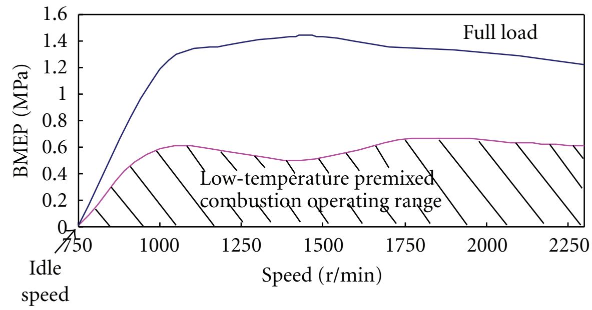


Figure 15: Operation range of low-temperature premixed combustion.

5. Conclusion

(1) The large diameter and straight port combustion chamber of a low compression ratio are more suitable for the premixed combustion in this study compared with the original combustion chamber.

(2) The premixed combustion with a late fuel injection strategy needs the high induction swirl ratio to accelerate the fuel-air mixing and promote the combustion, which is beneficial to improving the HC, CO, emissions and BSFC performance.

(3) The UMH nozzle has a large flow area of holes, which is beneficial to homogeneous mixture preparation for the premixed combustion in this research diesel engine.

(4) The better premixed combustion range achieved is as follows. The speed can cover from the idle speed to the rated speed. The load can reach to 50% of the full load of the corresponding external characteristics speeds except for 1430 r/min.

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References

[1] D. W. Dicked et al., "NO_x control in heavy-duty diesel engines—what is the Limit?" SAE Paper 980174, 1998.

[2] C. Arcounmanis et al., "Mixture formation and combustion in the D.I. diesel engine," SAE Paper 972681, 1997.

[3] S. Kimura, O. Aoki, H. Ogawa, and S. Muranaka, "New combustion concept for ultra-clean and high-efficiency small DI diesel engines," SAE Paper 1999-01-3681, 1999.

[4] S. Kimura, O. Aoki, Y. Kitahara, and A. Eiji, "Ultra-clean combustion technology combining a low-temperature and PremiXed Combustion Concept for Meeting Future Emission Standards," SAE Paper 2001-01-0200, 2001.

[5] T. J. Jacobs and D. N. Assanis, "The attainment of premixed compression ignition low-temperature combustion in a compression ignition direct injection engine," *Proceedings of the Combustion Institute*, vol. 31, pp. 2913–2920, 2007.

[6] T. J. Jacobs, S. V. Bohac, D. N. Assanis, and P. G. Szymkowicz, "Lean and Rich Premixed Compression ignition combustion in a light-duty diesel engine," SAE 2005-01-0166, 2005.

[7] K. Akihama, Y. Takatori, K. Inagaki, and S. Sasaki, "Mechanism of the smokeless rich diesel combustion by reducing temperature," SAE 2001-01-0655, 2001.

[8] D. Choi, P. C. Miles, H. Yun, and R. D. Reitz, "A parametric study of low-temperature, late-injection combustion in a HSDI diesel engine," *JSME International Journal, Series B*, vol. 48, no. 4, pp. 656–664, 2005.

[9] R. H. Stanglmaier and C. E. Roberts, "Homogeneous charge compression ignition (HCCI): benefit, compromises, and future engine applications," SAE Paper 1999-01-3682, 1999.

[10] F. Zhao, T. Asmus, D. Assanis, J. Dec, J. Eng, and P. Najt, "Homogeneous charge compression ignition (HCCI) engines," SAE Paper , 2003.

[11] K. Kawamoto and T. Araki, "Combination of combustion concept and fuel property for ultra-clean di diesel," SAE Paper 2004-01-1868, 2004.

[12] S. Simescu, S. B. Fiveland, and L. G. Dodge, "An experimental investigation of PCCI-DI combustion and emissions in a heavy-duty diesel engine," SAE 2003-01-0345, 2003.

[13] K. Okude et al., "Premixed compression ignition (PCI) combustion for simultaneously reduction NO_x and soot in diesel engine," SAE Paper 2004-01-1907, 2007.

[14] C. Noehre, M. Andersson, B. Johansson, and A. Hultqvist, "Characterization of partially premixed combustion," SAE Paper 2006-01-3412, 2006.

[15] B. Kobayashi, S. Najima, M. Mikami, N. Kojima, and T. Kanda, "Effects of fuel volatility on combustion characteristics of PCCI diesel engine," SAE Paper 2007-01-1862, 2007.

[16] M. Y. Kim and C. S. Lee, "Effect of a narrow fuel spray angle and a dual injection configuration on the improvement of exhaust emissions in a HCCI diesel engine," *Fuel*, vol. 86, no. 17-18, pp. 2871–2880, 2007.

[17] J. Ma, X. Lü, L. Ji, and Z. Huang, "An experimental study of HCCI-DI combustion and emissions in a diesel engine with dual fuel," *International Journal of Thermal Sciences*, vol. 47, no. 9, pp. 1235–1242, 2008.

[18] M. Zheng, G. T. Reader, and J. G. Hawley, "Diesel engine exhaust gas recirculation—a review on advanced and novel concepts," *Energy Conversion and Management*, vol. 45, no. 6, pp. 883–900, 2004.

[19] B. Challen and R. Baranescu, *Diesel Engine Reference Book*, Butterworth-Heinemann, 2nd edition, 1999.

[20] X. Miao, X. Qiao, X. Wang et al., "Study on ultramultihole nozzle fuel injection and diesel combustion," *Energy and Fuels*, vol. 23, no. 2, pp. 740–743, 2009.

[21] N. Shimazaki, T. Tsurushima, and T. Nishimura, "Dual mode combustion concept With premixed diesel combustion by direct injection near top dead center," SAE Paper 2003-01-0742, 2003.

[22] S. Marcus, K. Max, K. Peter, and L. Alfred, "Optical investigations on partially premixed diesel combustion for different operating parameters," SAE paper 2008-01-0041, 2008.

[23] S. Inderpal, Z. Lurun, D. Minq-Chia, W. Bryzik, and A. Naeim, "Effect of nozzle hole geometry on a Hsdi diesel engine- out emissions," SAE paper 2003-01-0704, 2003.