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

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1. Introduction

Conventional diesel combustion exhibits the trade-off relationship between reduction of NOx and soot emissions. The strategies for reducing soot increase NOx; for example increasing injection pressure and swirl ratio reduce soot, but increase NOx in the conventional diesel combustion. High levels of exhaust gas recirculation (EGR) also reduce NOx but increase soot. It is well known that NOx emissions are largely dependent on the equivalence ratios of air-fuel mixture. NOx emissions reach maximum value when combustion occurs near stoichiometric air-fuel ratio. However, it can be lowered when mixture is over-rich or over-lean. Generally speaking, soot emission is largely produced during the diffusion combustion, but it is very low during the premixed combustion. Therefore researchers attempt to mix fuel and air as homogeneous as possible prior to ignition to achieve the premixed combustion (Kimura et al.,1999,2001; Hasegawa & Yanagihara,2003;Takeda et al.,1996; Shimazaki et al.,1999). Studies show that homogeneous charge premixed mixture low-temperature combustion can simultaneously reduce NOx and soot emissions(Nandha & Abraham,2002;Walter & Gatellier,2002; Lewander et al.,2008;Husberg et al.,2005;Lejeune et al.,2004).

The purpose of this investigation is to develop a new low-temperature premixed combustion mode in a six-cylinder commercial vehicle diesel engine using the UMH nozzle and EGR. The UMH nozzle facilitates better mixing of fuel and air prior to ignition and the resultant realization of the premixed combustion, because of its shortened the injection duration and improved atomization compared with the conventional nozzle (Miao et al.,2009). This investigation also explores the combustion characteristics of the UMH nozzle through the experiments of selected operation conditions of 1400 r/min, 0.575MPa and 1000 r/min, 0.279 by adjusting injection timing, injection pressure and EGR rate. The results showed that NOx and soot emissions of the selected operation conditions were simultaneously largely reduced.

2. Experimental apparatus

2.1 UMH nozzle structure

Figure 1 shows the schematic of the UMH nozzle (Miao et al.,2009). It consists of a needle and a body, and has the following characteristics: (1) there are two layers of injection holes in the front part of body. Any injection hole of upper layer and the corresponding injection hole of under layer are positioned in a vertical plane; (2) 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（$\alpha 2>\alpha 1$）; and (3) it has a large enough flow area of holes such that cyclic fuel can be completely injected into the combustion chamber prior to ignition, which is a prerequisite for premixed combustion.

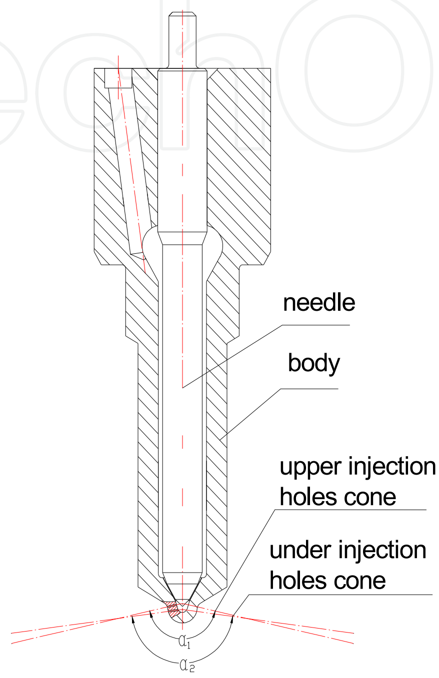


Fig. 1. Schematic of UMH nozzle

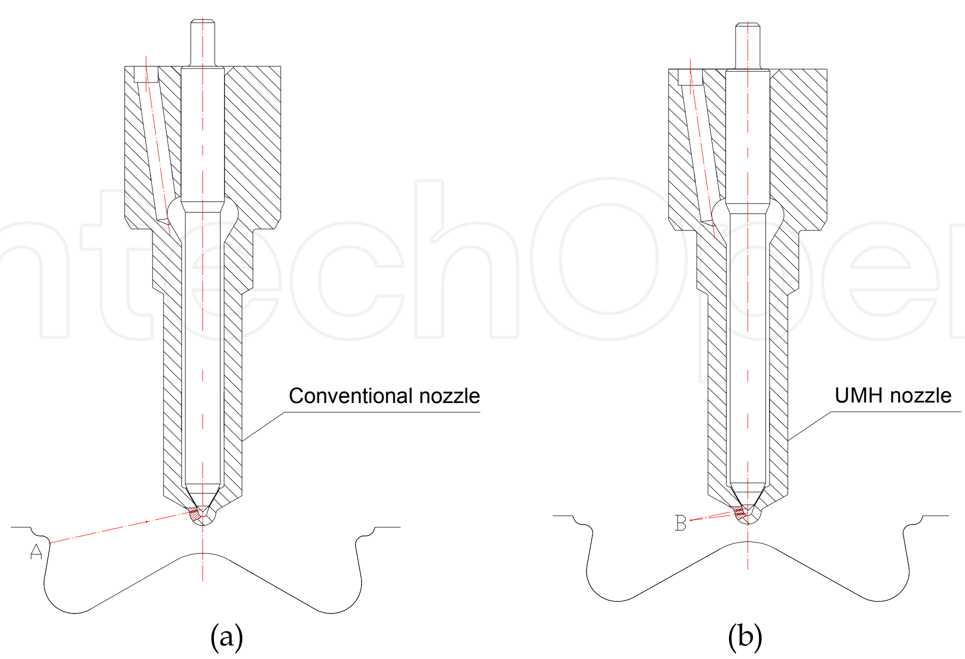


Fig. 2. Schematic comparison between the conventional and UMH nozzle (a) Conventional nozzle (b) UMH injection nozzle

Figure 2 shows the schematic comparison between the existing conventional nozzle and the UMH nozzle. There is only one layer of holes on the conventional nozzle, so sprays possibly impinge on the wall of the combustion chamber or the cylinder liner to cause high HC and CO emissions. Sprays might impinge at point A as shown in Figure 2(a). The UMH nozzle, however, has two layers of holes and the cone angle of the under-layer injection holes is larger than that of the upper-layer holes. Two sprays of upper and under-layer meet in the space of the combustion chamber, for example, at point B as shown in Figure 2(b). The results showed that the UMH nozzle exhibits shorter spray penetration than does the conventional nozzle (Miao et al.,2009). 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. Therefore, the result is a more homogeneous mixture required to perform the premixed combustion in diesel engines.

Type	Hole number	Hole diameter (mm)	Flow rate (l/min)@10MPa
Original nozzle	8	0.17	1.2
UMH nozzle	16	0.16	2

Table 1. Specifications of test nozzles

Table 1 lists the specifications of test nozzles. It can be seen that the flow rate of the UMH nozzle with smaller diameter hole is higher than that of the original nozzle by up to 67%, which helps the UMH nozzle not only shorten the injection duration and also improve fuel atomization.

2.2 Combustion experimental apparatus

The specifications of the test engine are shown in Table 2. Its operating conditions are set at 1400 r/min, 0.575MPa and 1000 r/min, 0.279MPa. The test engine is operated on the commercially available diesel fuel with the cetane number of 51 in all the test cases. The coolant temperature is set to 80±3°C. Figure 3 gives the combustion experimental apparatus. The EGR cooler and the intercooler are water-cooled, with water circulation volume and water temperature that are adjustable. The inlet air temperature after the intercooler is maintained at 40±3°C during the whole experiment. The fuel injection is performed by a high-pressure common-rail electric-controlled system on the engine. The exhaust gas emissions are measured using a HORIBA MEXA-7100 gas analyzer, smoke density (soot) measured using an AVL 415s smoke meter, and particle matter (PM) measured using an AVL 472 partial-flow particulate sampler which allows double particulate filters to be exposed. In-cylinder pressure is acquired using a KISTLER cylinder pressure sensor.

Model	CA6DF2
Type	In-line, supercharged, inter-cooled
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 torque/speed（N.m/r/min）	580～620/1000
Minimum BSFC*（g/kW.h）	205

Model	CA6DF2
Combustion chamber	reentrant
Compression ratio	16.5
Ricardo swirl ratio of inlet port	2.8

*BSFC is an acronym for 'brake specific fuel consumption'.

Table 2. Specifications of the test engine

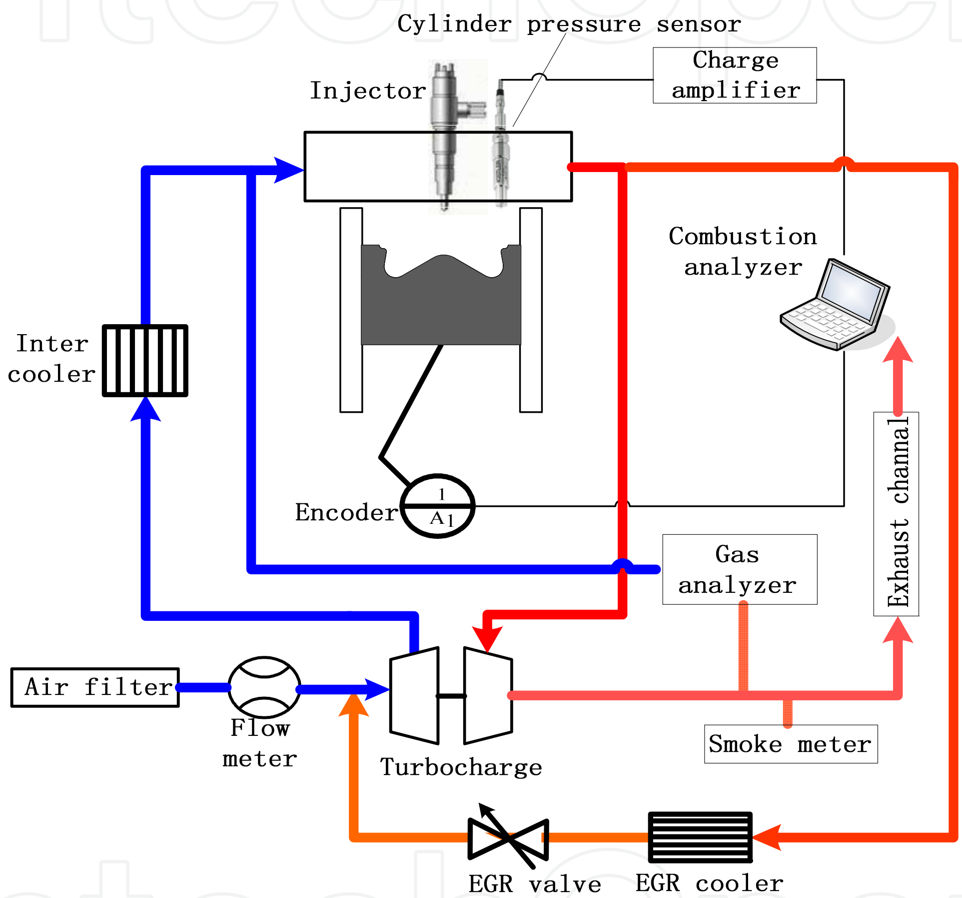


Fig. 3. Combustion experimental apparatus

EGR rate is denoted by the follow formula.

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

where, E, I, and A denote exhaust gas, inlet gas and atmosphere respectively.

3. Experimental results and discussions

The operating conditions are set at 1400 r/min, 0.575MPa, 1.18-2.29 of excess air ratio (λ) (EGR rates from 0 to 33%) (referred to as case A) and 1000 r/min, 0.29MPa, 1.68-2.92 of λ

(EGR rates from 0 to 80%) (referred to as case B). Experiments are carried out using the UMH nozzle and the original nozzle respectively. The cyclic fuel can't be completely injected into the combustion chamber before ignition because of smaller flow rate of the original nozzle, so experiment with the original nozzle can only achieve the conventional combustion. It means that this combustion can't eliminate the trade-off relationship between reduction of NOx and soot emissions, and accordingly the engine with the original nozzle is only tested in original condition. Experiments with the UMH nozzle, however, are carried out by adjusted EGR rates, injection pressures and injection timings to achieve the low-temperature premixed combustion. These optimum parameters (include EGR rate, injection pressure and injection timing) are different for case A and B to achieve the minimum values of NOx and soot emissions while keep the break specific fuel consumption (BSFC) not to be significantly deteriorated because their excess air ratios are different.

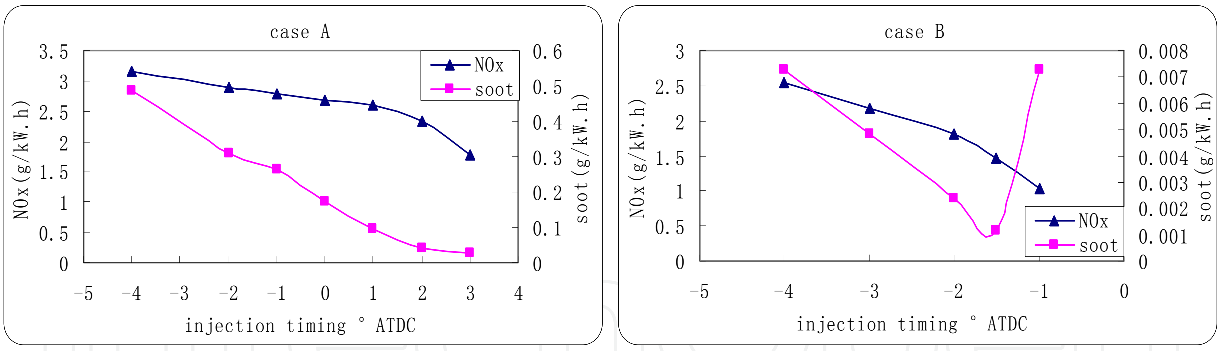
3.1 Effects of injection timing on combustion characteristics

The EGR rates of case A and B are set at 28% and 80% respectively, and the injection pressure is all 110MPa. NOx, soot, HC, CO, BSFC and cylinder pressure are measured by varying the injection timing. The results are shown in Figure 4.

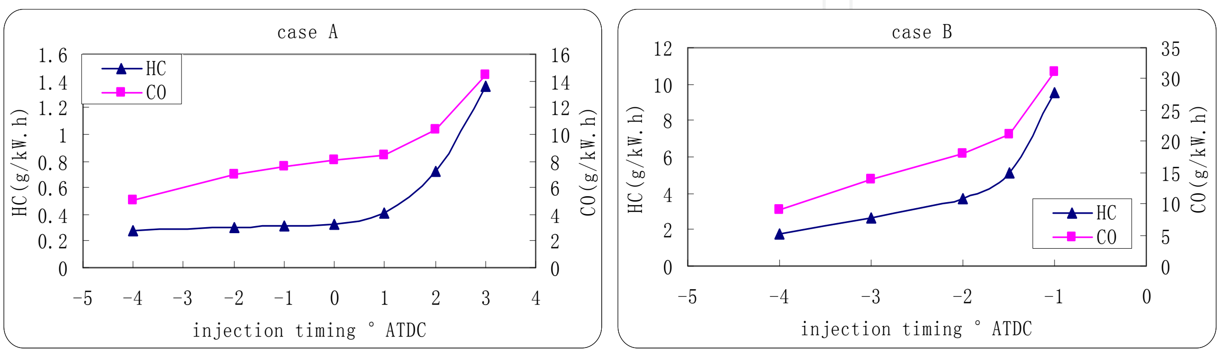
It can be seen that NOx and soot emissions are simultaneously decreased by 43% and 94% respectively with retarding the injection timing from -4°ATDC to 3°ATDC in case A. For case B, NOx and soot emissions are also simultaneously decreased by 42% and84% respectively with retarding the injection timing from -4°ATDC to -1.5°ATDC, further retarding to -1°ATDC causes continuing reduction of NOx but increase of soot.

It is not difficult to understand NOx reduction with retarding the injection timing. The heat release rates at different injection timing are shown in Figure 5. The fuel injection rate curves are also plotted in Figure 5 and set at the same start point, accordingly the corresponding heat release rate curves must be shifted. In this way, it is convenient to compare the combustion characteristics, and to distinguish if the premixed combustion at different injection timing is achieved.

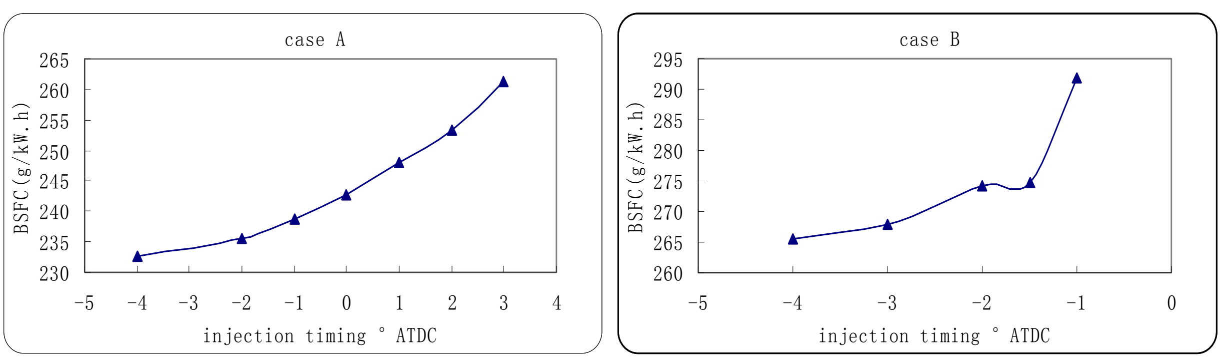
The premixed combustion of this investigation means combustion that occurs after the cyclic fuel completely injected into the combustion chamber. Therefore it is important the duration between the injection end point and the combustion start point (Shimazaki,2003). This duration affects combustion characteristics especially emissions, because it represents the degree of the premixed combustion. Here it is defined as the premixed degree duration denoted by τ_{pmix}. The cyclic fuel has not been completely injected into the combustion chamber prior to ignition when τ_{pmix} is less than zero, it means the complete premixed combustion can't be achieved, still belongs to the conventional combustion. However, the cyclic fuel has just been completely injected into the combustion chamber prior to ignition when τ_{pmix} is equal to zero, but it is short for fuel and air to completely mix, which can not form homogeneous mixture. The homogeneity of mixture tends to improve with the increase of τ_{pmix}, and accordingly soot and NOx emissions tend to decrease simultaneously when high levels of EGR were used. So τ_{pmix} is a very important parameter to help compare between the premixed combustion and the conventional combustion.



a) Effects of injection timing on NOx and soot emissions



b) Effects of injection timing on HC and CO emissions



c) Effects of injection timing on BSFC

Fig. 4. Effects of injection timing on engine performance

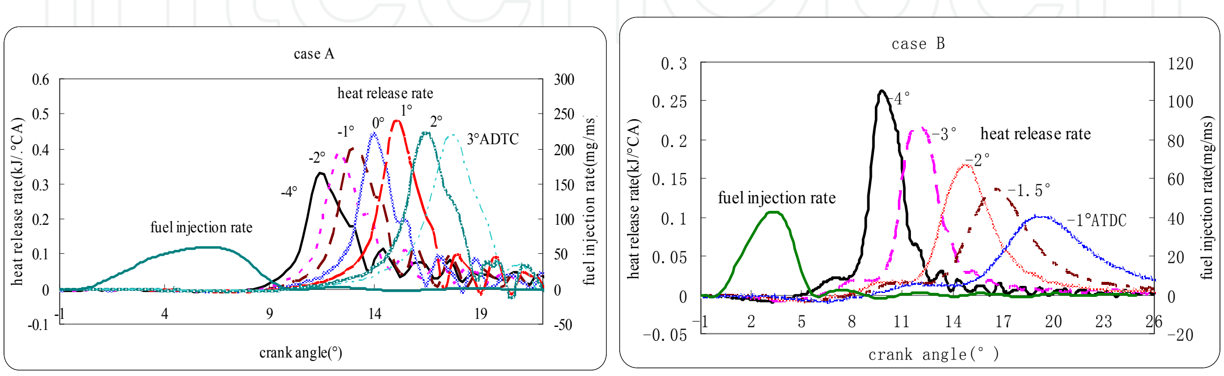


Fig. 5. Heat release rates at different injection timing

The premixed degree duration τ_{pmix} at different injection timings is shown in Table 3. It can be seen that the premixed combustion has not been achieved at -4°ATDC of the injection timing in case A. The premixed combustion has just been achieved at -1°ATDC of the injection timing while τ_{pmix} is equal to zero. The τ_{pmix} is equal to 0.3°CA at 3°ATDC of the injection timing, which means that it is longer for fuel and air to mix prior to ignition. The longer τ_{pmix} is, the more homogeneous mixture is. In this way, it is possible to achieve the homogeneous charge combustion, eventually soot and NOx emissions are simultaneously reduced to very low when high levels of EGR were used. This is different with the conventional combustion. For case B, the premixed combustion has already been achieved at -4°ATDC of the injection timing while τ_{pmix} is equal to 3.43°CA, so soot is low. The τ_{pmix} is equal to 3.9°CA with retarding the injection timing to -2°ATDC, therefore soot is already very low. But it can be seen from Figure 5 that the combustion rate is very low with further retarding the injection timing to -1°ATDC, which is not beneficial to complete combustion, and accordingly causes increase of soot.

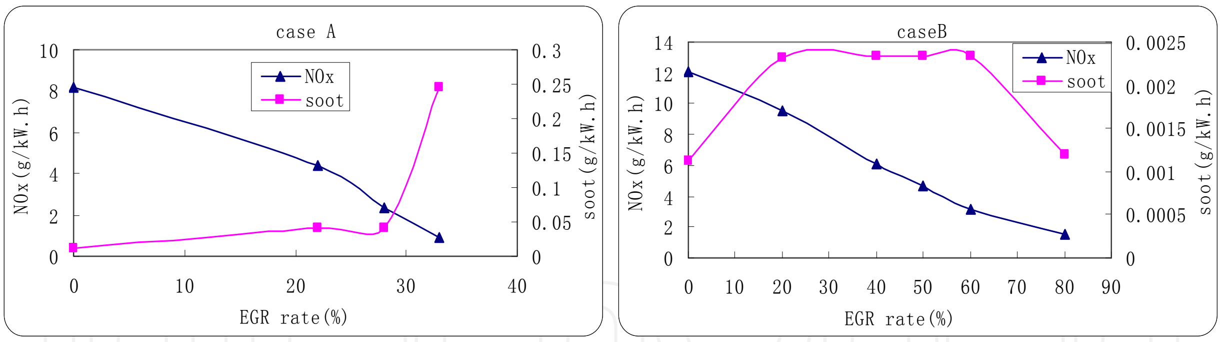
case	Injection start θ1	Injection end θ2	Combustion start θ3	Ignition delay =θ3-θ1	T_{pmix} =θ3-θ2
	°ATDC	°ATDC	°ATDC	°CA	°CA
A	-4	6	4.3	8.3	-1.7
	-2	8	6.6	8.6	-1.4
	-1	9	7.8	8.8	-1.2
	0	10	9.3	9.3	-0.7
	1	11	10.7	9.7	-0.3
	2	12	12.1	10.1	0.1
	3	13	13.3	10.3	0.3
B	-4	1.6	5.03	9.03	3.43
	-3	2.6	6.18	9.18	3.58
	-2	3.6	7.5	9.5	3.9
	-1.5	4.1	8.2	9.7	4.1
	-1	4.6	8.9	9.9	4.3

Table 3. Premixed degree duration τ_{pmix} at different injection timing

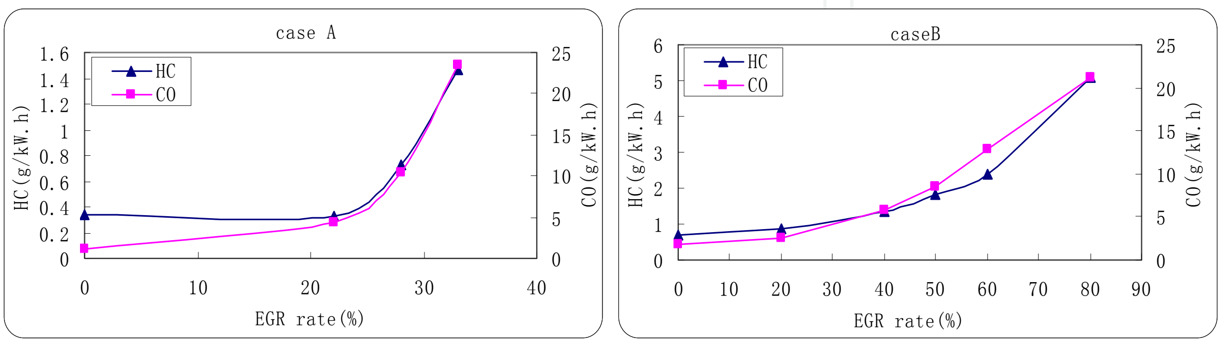
HC, CO and BSFC have a slight change until 1°ATDC of the injection timing, but the further retarding injection timing causes deterioration of these performances in case A. These performances are significantly worsened after 1°ATDC of the injection timing because the combustion period is far away from the top dead center (TDC). For case B due to higher EGR rate, HC and CO emissions are swiftly increased with retarding the injection timing to after -1.5°ATDC because of slowing combustion rate, and BSFC is also worsened due to far away from TDC of the combustion period.

3.2 Effects of EGR rate on combustion characteristics

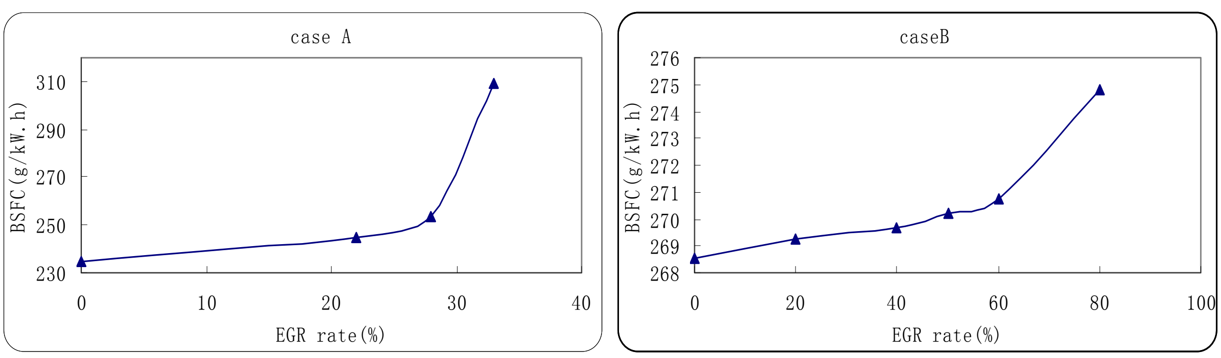
The injection timing of case A and B are set at 2°ATDC and -1.5°ATDC respectively, injection pressure is all 110MPa. NOx, soot, HC, CO, BSFC and cylinder pressure are measured by varying EGR rates. The results are shown in Figure 6.



a) Effects of EGR rate on NOx and soot emissions



b) Effects of EGR rate on HC and CO emissions



c) Effects of EGR rate on BSFC

Fig. 6. Effects of EGR rate on engine performance

It can be seen from Figure 6 that NOx is linearly decreased with the increase of the EGR rate. For case A, NOx is decreased by 71% and 89% respectively when the EGR rate is from zero to 28% and 33%. Soot is almost unchanged when the EGR rate is less than 28%, but further increasing the EGR rate causes swift increase of soot emission. For case B, NOx is decreased by 88% when the EGR rate is from zero to 80%, but soot has a complicated tendency. Firstly soot has a slight change when the EGR rate is low, then reaches the maximum value when the EGR rate is 20%, however soot begins to decrease swiftly with further increase of the EGR rate.

Figure 7 shows the heat release rates and Table 4 shows the premixed degree duration τ_{pmix} at different EGR rates. The reduction of oxygen concentration with EGR causes NOx decrease and soot increase, but on the other hand, longer τ_{pmix} due to EGR causes soot decrease. Therefore effects of EGR on soot emission are as follows. Lower EGR rates don't cause significant change of soot. But soot is deteriorated when the EGR rate is more than 28%

in case A due to lower excess air ratio (lower oxygen concentration). Longer τ_{pmix}, however, dominates combustion process compared to oxygen concentration decrease when the EGR rate is high in case B due to higher excess air ratio. There is much time for fuel and air to mix prior to ignition with the aid of EGR. This is beneficial to the formation of a homogeneous mixture. Therefore soot has been greatly decreased again when EGR rate is high in case B.

HC, CO and BSFC have a slight change when the EGR rate is less than 22%, then begin to increase with further increase the EGR rate, especially worsen when the EGR rate reaches 33% in case A. For case B, however, HC, CO and BSFC have a slight change when EGR rate is less than 30%, then begin to worsen with further increase of the EGR rate. This is because of an incomplete combustion causing with the EGR rate increase.

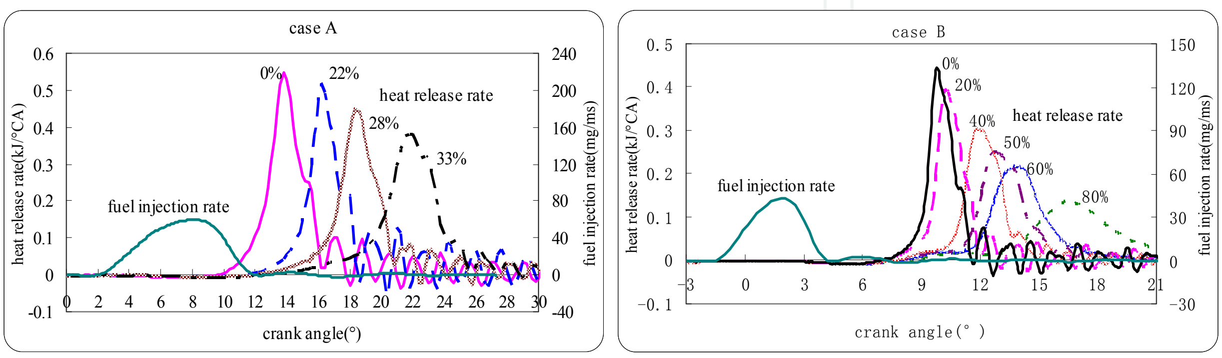


Fig. 7. Heat release rates at different EGR rates

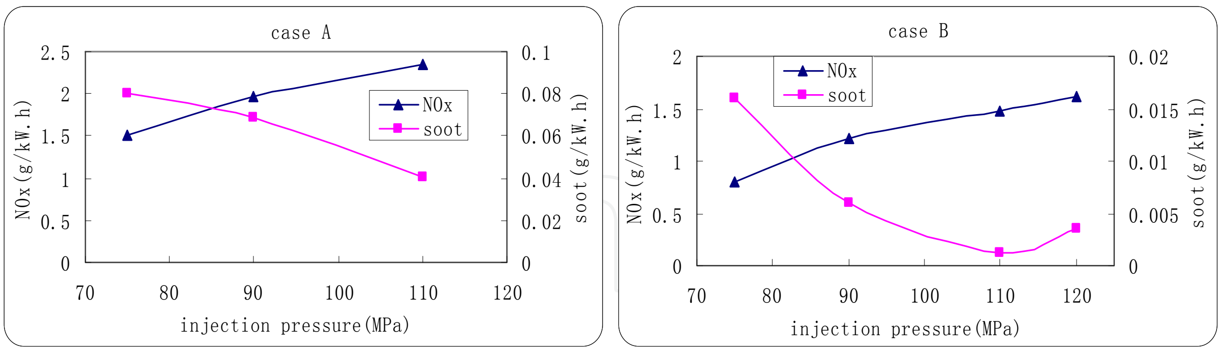
case	EGR rate	Injection start θ1	Injection end θ2	Combustion start θ3	Ignition delay =θ3-θ1	τ_{pmix} =θ3-θ2
	%	°ATDC	°ATDC	°ATDC	°CA	°CA
A	0	2	12	9.7	7.7	-2.3
	22	2	12	11.6	9.6	-0.4
	28	2	12	12.1	10.1	0.1
	33	2	12	12.6	10.6	0.6
B	0	-1.5	4.1	6.94	8.44	2.84
	20	-1.5	4.1	6.95	8.45	2.85
	40	-1.5	4.1	7.52	9.02	3.42
	50	-1.5	4.1	7.73	9.23	3.63
	60	-1.5	4.1	7.75	9.25	3.65

Table 4. Premixed degree duration τ_{pmix} at different EGR rates

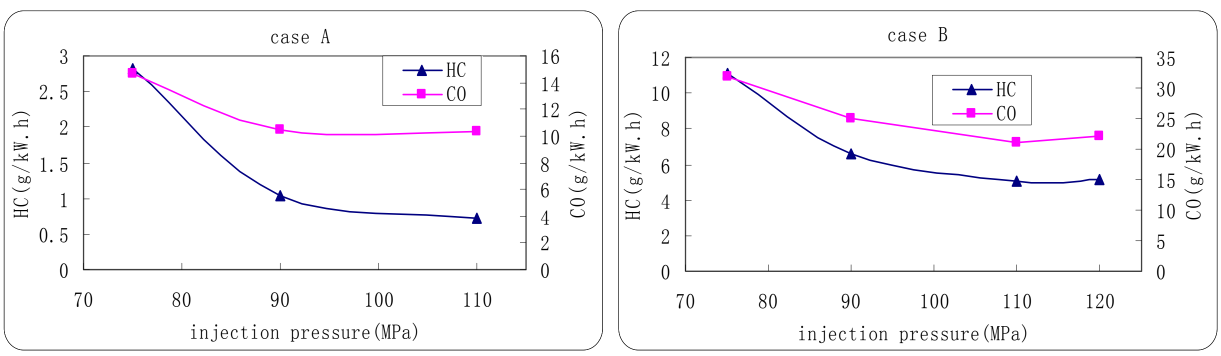
3.3 Effects of injection pressure (rail pressure) on combustion characteristics

The injection timing of case A and B are set at 2°ATDC and -1.5°ATDC respectively, EGR rates are set at 28% and 80% respectively. NOx, soot, HC, CO, BSFC and cylinder pressure are measured by varying the injection pressure. The results are shown in Figure 8.

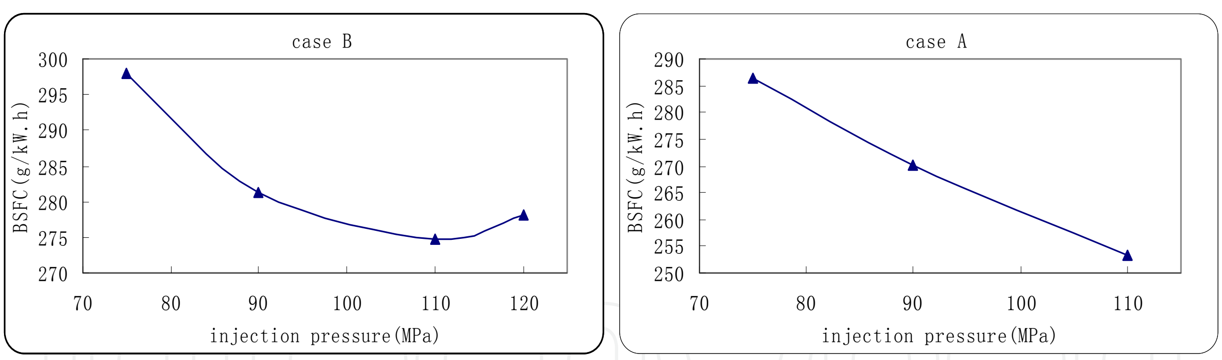
It can be seen that soot, HC, CO and BSFC have some certain reduction except NOx with the injection pressure increase in case A. For case B, these changes are almost the same as case A when the injection pressure is less than 110MPa. But these performances are deteriorated with the injection pressure further increase. This is maybe due to the spray wall-impingement with the injection pressure further increase.



a) Effects of injection pressure on NOx and soot emissions



b) Effects of injection pressure on HC and CO emissions



c) Effects of injection pressure on BSFC

Fig. 8. Effects of injection pressure on engine performance

Figure 9 shows the heat release rates and Table 5 shows the premixed degree duration τ_{pmix} at different injection pressures. It can be seen that combustion advance and heat release peak increase with the injection pressure increase, and thus NOx is increased. But the advance of injection end point leads to longer τ_{pmix} with the injection pressure increase. In case A, τ_{pmix} increases from -0.5°CA to 0.1°CA with the injection pressure increase from 75MPa to 110MPa, so soot is decreased. Additionally, BSFC can be improved due to the combustion advance with the injection pressure increase. Higher injection pressure, of course, can improve mixing of fuel and air, finally improve combustion. Therefore HC and CO emissions are also decreased. In case B, although τ_{pmix} keeps almost unchanged with the injection pressure increase from 90MPa to 110MPa, but the higher injection pressure can improve mixing of fuel and air, so soot can be significantly reduced. Over spray penetration, however, leads to bad performance when the injection pressure reaches 120MPa.

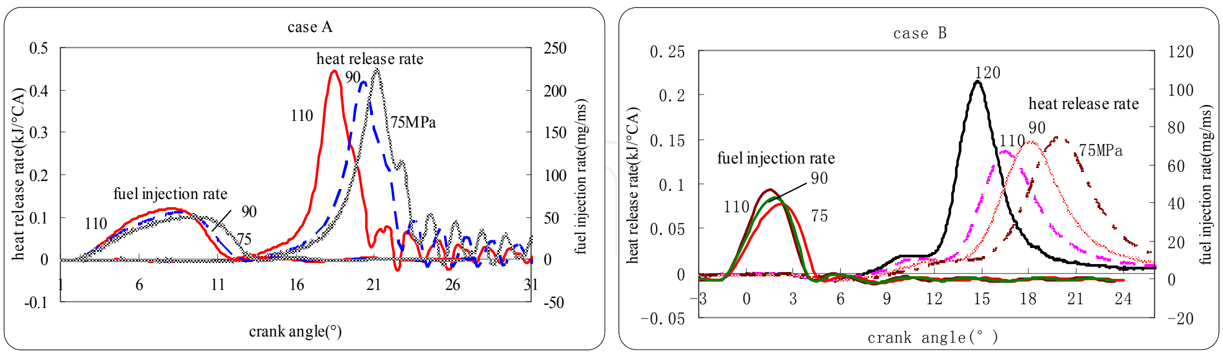


Fig. 9. Heat release rates at different injection pressure

case	Injection start θ1	Injection pressure	Injection end θ2	Combustion start θ3	Ignition delay =θ3-θ1	τ_{pmix} =θ3-θ2
	°ATDC	MPa	°ATDC	°ATDC	°CA	°CA
A	2	110	12	12.1	10.1	0.1
		90	12.5	12.8	10.8	0.3
		75	13.4	12.9	10.9	-0.5
B	-1.5	120	4	7.72	9.22	3.72
		110	4.1	8.2	9.7	4.1
		90	4.2	8.92	9.22	4.72
		75	4.7	9.5	11	4.8

Table 5. Premixed degree duration τ_{pmix} at different injection pressure

3.4 Combustion characteristics comparison between UMH nozzle and original nozzle

Figure 10 shows that the relationship of NOx and soot emissions is compared between the UMH nozzle and the original nozzle (original engine). Because engine with the original nozzle can't achieve the complete premixed combustion and still belongs to the conventional combustion, which trade-off relationship between reduction of NOx and soot emissions can't be overcome. So the engine with the original nozzle is only tested in the original condition, this result is used to compare with that of the UMH nozzle. The engine with the UMH nozzle, however, can achieve the premixed combustion to obtain simultaneous reduction of NOx and soot emissions. Therefore it is tested by adjusted the EGR rate, injection pressure and injection timing to obtain a series of values of NOx and soot emissions. It can be seen that NOx is high but soot very low without EGR, and then NOx is swiftly decreased with the EGR rate increase. In sum, for case A, NOx and soot emissions can simultaneously achieve minimum values when EGR rate is 28%, and combing suitable injection pressure and injection timing. For case B, NOx is very low when EGR rate is 80%, and soot is also simultaneously reduced to very low value combing suitable injection pressure and injection timing. Table 6 shows combustion characteristics comparison between the UMH nozzle and the original nozzle. NOx and soot emissions of the UMH nozzle are reduced by 68.2% and 20% respectively than that of the original nozzle in case A, but BSFC increases by 10.4%. For case B, NOx and soot emissions of the UMH nozzle are reduced by 78.1% and 76.2% respectively than that of the original nozzle, but BSFC only increases by 8.7%. This is because the excess air ratio of case B is higher than that of case A. Therefore case B can use higher EGR rate than case A, which leads to the achievement of much more lean-homogeneous charge combustion.

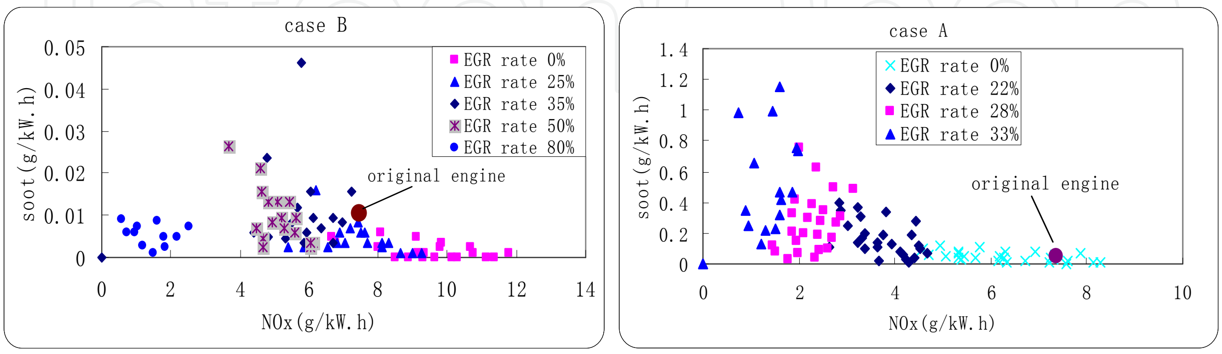


Fig. 10. NOx—soot emissions comparison between UMH and original nozzle

case	Nozzle type	EGR rate	excess air ratio λ	Injection pressure	Injection start	NOx	HC	CO	soot	BSFC	τ_{pmix}
		%		MPa	°ATDC	g/kW.h					°CA
A	Original nozzle	0	2.25	75	-2	7.35	0.54	1.38	0.05	235	-2
	UMH nozzle	28	1.49	110	2	2.34	0.73	10.33	0.04	259.4	0.1
B	Original nozzle	0	3.5	50	0	6.73	1.52	7.69	0.005	252.6	-4.6
	UMH nozzle	80	1.68	110	-2	1.47	5.08	21.15	0.001	274.8	4.1

Table 6. Combustion characteristics comparison between UMH and original nozzle

4. Conclusions

1. The UMH nozzle has a large flow area of holes, which is beneficial to homogeneous mixture preparation prior to ignition. The better premixed combustion can be achieved combing the UMH nozzle with EGR, suitable injection pressure and injection timing to obtain simultaneous reduction of NOx and soot emissions in a diesel engine.
2. For case A, NOx and soot emissions of the UMH nozzle compared with the original nozzle are simultaneously reduced by 68.2% and 20% respectively when the EGR rate is

28% (excess air ratio is 1.49), the injection pressure is 110MPa and the injection timing is 2°ATDC. For case B, however, NOx and soot emissions of the UMH nozzle compared with the original nozzle are simultaneously reduced by 78.1% and 76.2% respectively when the EGR rate is 80% (excess air ratio is 1.68), the injection pressure is 110MPa and the injection timing is -1.5°ATDC.

3. Case B can use higher EGR rate to achieve much more lean-homogeneous charge premixed combustion because the excess air ratio of case B is higher than that of case A. Therefore it can obtain remarkably simultaneous reduction of NOx and soot emissions and slight increase of BSFC by 8.7% compared with the original engine.

5. Acknowledgements

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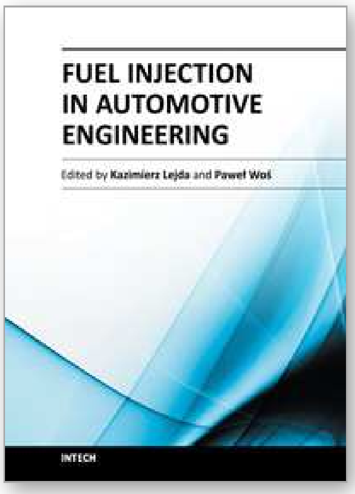

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The main topic of "Fuel injection in automotive engineering" book is fundamental process that determines the development of internal combustion engines and performances of automotive vehicles. The book collects original works focused on up-to-date issues relevant to improving injection phenomena per se and injection systems as the engine key components.