Comparison of Mixture Formation in Two Combustion Modes in Directly-Injection Diesel

XIAOLU LI, LIJUN XU, YING LI, HONGYAN CHEN, JUN WANG,
JUAN FENG, BAOPING XIAO, XUEFEI ZHAO
School of Electromechanical Engineering
China Jiliang University
Xueyuan Rd., Higher Education District, Hangzhou, Zhejiang Province
PEOPLE REPUBLIC OF CHINA

Abstract: This paper simulates the density and temperature fields in the process of mixture formation with two combustion modes in a Diesel engine. The computational simulations and experiments show that in the normal combustion mode, the density and temperature of mixtures change greatly in the combusting sprays so that there are premixed combustion and diffusion combustion simultaneously, which results in high NO_X and smoke emissions, low CO and HC emissions. Homogeneous charge compression ignition (HCCI) combustion takes place by early-injection to form the stratified homogeneous mixtures, which results in very low NO_X and smoke emissions, high CO and unburned fuel emissions.
Key-Words: Diesel engine; Mixture; Injection; Computational simulation; Emission; HCCI

1 Introduction

Diffusion combustion is the main combustion mode of normal Diesel Engine. This kind of combustion mode can't resolve the contradiction between smoke and NO_X formation conditions, which results in producing a lot of poisonous NO_X and smoke to environments and human beings [1]. Recently, the homogeneous charge compression ignition (HCCI) combustion is becoming the research hotspot in the internal-combustion engine field [2, 3]. This new internal-combustion engine combustion mode can reduce NO_X and smoke emissions simultaneously. The most important characteristic of this combustion mode is the homogeneous mixture and ignition by compression, which essential difference is the mixing state before ignition [4]. This paper obtains engine emission and thermal efficiency in these two combustion modes.

Also, the density and temperature fields in the process of mixture formation in two combustion modes are simulated to explain the experimental results in a Diesel engine.

2 Computational Conditions, Parameters of Engine and Testing Equipment

Three-dimensional model of in-cylinder flow is based on classic fluid dynamics, namely the continuity, momentum and energy equations. The dispersion droplet model (DDM) is used to simulate the process of fuel spray, which is consisted of disperse particles. The spray liquid model includes the flow, breakup, collision and evaporation sub-models [5].

The computational simulation and experiment are completed on a 1E150C Diesel engine. In its normal combustion mode, the specification of the nozzle is $6\times\Phi0.25\times150^0$, and the fuel injection advance angle is 35^0CA BTDC (crank angle). In HCCI combustion mode, the spray-angle is small as possible in order to avoid fuel impinging wall. Simultaneously, for better spray and more lean mixture, more nozzles should be manufactured, so its specification is $6\times\Phi0.25\times90^0 + 4\times\Phi0.18\times45^0$. The fuel injection advance angle of early-injection is 82 ^{0}CA BTDC. The injection durations of these two combustion modes are 25 ^{0}CA.

Engine parameters, operating parameters and calculating parameters are shown in Table 1.

The schematic of calculation grid is shown in Fig.1. Calculation starts at 121^0CA BTDC. In the calculation process, n-heptane is instead of diesel fuel [6]. When simulating the HCCI combustion, the cylinder wall temperature is assumed as 400K, cylinder top temperature 410K, and piston temperature 420K. While simulating the normal Diesel combustion, the cylinder wall temperature is assumed as 420K, cylinder top temperature 440K, and piston temperature 460K. The exhaust gas pressure is tested by a Kistler's 4045A5 low voltage sensor and 4618A0 charge amplifier. The exhaust gas pressure is shown in Fig.2. The scavenging pressure is a constant value of 111.7 kPa. All these are the boundary conditions of calculation. The emissions of NO_X, HC and CO are tested by AVL

Company's five gas analyzer Di Gas4000Light, and the smoke densities are measured by a full-automated FBY-200 smoke meter. The output power of the engine is calculated based on the data of engine torque and speed measured directly by the D650 hydraulic dynamometer and magneto-electrical speed sensor, respectively, while the brake thermal efficiency by using the data of engine output power and data of fuel consumption calibrated by the FCM-04 mass flow meter.

Table 1 Engine Conditions

Parameter	Value
Cylinder diameter/mm	150
Stroke/mm	225
Connecting rod length/mm	450
Compression ratio	15
Shape of combustion chamber	Shallow ω
Exhaust port number	3
Exhaust port shape	Rectangle
Exhaust port wide/mm	30
Exhaust port high/mm	63
Exhaust port open /^{0}CA	+109 ATDC
Exhaust port close /^{0}CA	-109 ATDC
Scavenging port number	6
Scavenging port shape	Circular
Scavenging port diameter/mm	30
Scavenge port open /^{0}CA	+121.8 ATDC
Scavenge port close /^{0}CA	-121.8 ATDC
Fuel injection temperature /K	350
Fuel quantity per cycle /g	0.117
Injection pressure / MPa	20
Injection duration /^{0}CA	25
Fuel molecular formula	C_7H_{16}
Rate /rpm	450
Scavenging pressure /kPa	111.7
Scavenging temperature /K	310

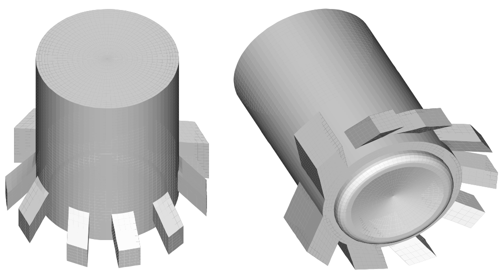


Fig.1 Schematic of Computational Grid

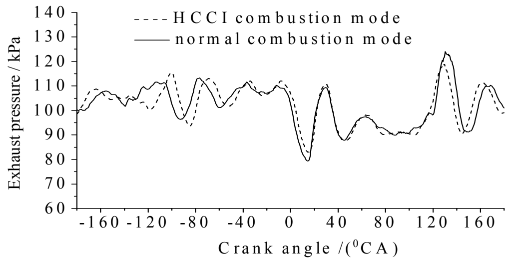


Fig.2 Exhaust Gas Pressure

3 Experiment Results and Analysis

In the experiment, the fuel consumption and rate is invariable. The experiment results are shown in Table 2, where p_e is the average effective pressure, η_{et} the effective thermal efficiency.

In the normal combustion mode, there are more NO_X and smoke emissions, volume fraction of NO_X is 652×10^{-6}, and smoke is 0.5 Bosch. The formation condition of NO_X is high temperature and rich oxygen while the formation condition of smoke is high temperature and lean oxygen [1]. In the normal combustion mode, diffusion combustion is the main combustion mode, and exist the formation condition of NO_X and smoke in the cylinder simultaneously. Owing to high temperature, there are less CO and HC emissions. In this combustion mode, the thermal efficiency is 31.32%.

In HCCI combustion mode, owing to early-injection, the stratified homogeneous mixtures are formed and burned at many points, which cause the peak temperature lower to prevent NO_X and smoke formation. In this experiment, there is none of these two emissions. Due to early-injection, the fuel impinges the cylinder wall, which results in high unburned fuel emissions. This is consistent with the experimental observation. In addition, CO and HC emissions are higher than normal combustion mode's. It is these unburned emissions resulting in low engine thermal efficiency, 18.66%. Certainly, the low thermal efficiency may also because of an advancing burning before the top dead center.

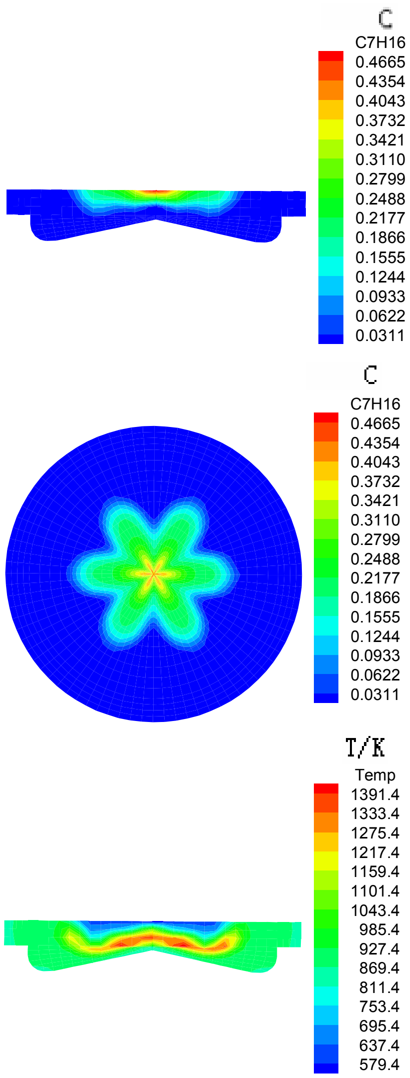
Table 2 Experimental Results

Combustion mode	Normal combustion	HCCI combustion
$\varphi_{NOx}\times10^{-6}$	652	0
Smoke / Bosch	0.5	0
$\varphi_{HC}\times10^{-6}$	10	65
φ_{CO} / %	0.01	0.32
p_e / MPa	0.47	0.28
η_{et} / %	31.32	18.66

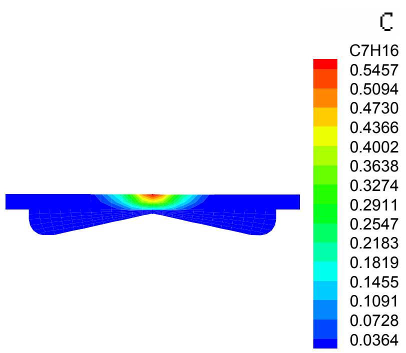
4 Calculation Results and Analysis

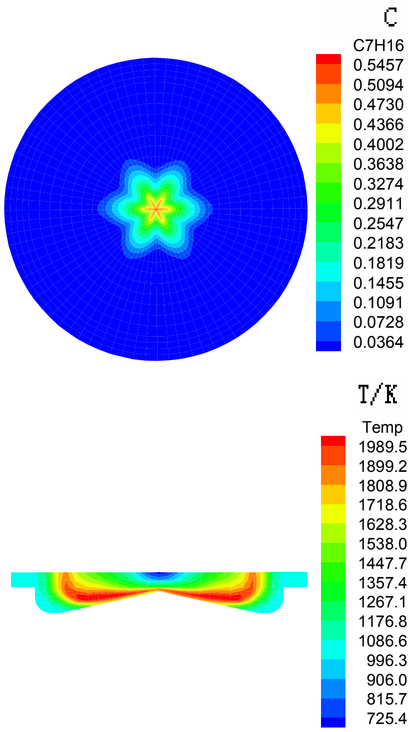
4.1 Formation of Mixture in Normal Combustion Mode

This paper simulates the density and temperature fields in the process of mixture formation with the normal combustion mode in a Diesel engine, the results are shown in Fig.3.



(a) 19^0CA BTDC





(b) 11^0CA BTDC

Fig.3 In-cylinder Fuel Concentration and Temperature Fields in the Mixture Formation Process under the Normal Combustion Mode

The unit of fuel density is mole fraction, and the unit of temperature is K. Injection starts at 35^0CA BTDC, and terminates at 10^0CA BTDC. The injection duration is 25^0CA.

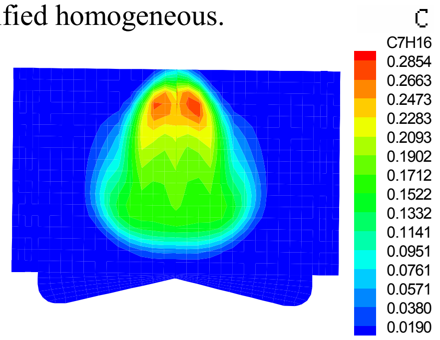
Concentration and temperature fields at 16^0CA after the start of injection are shown in Fig.3 (a). There are rich mixtures near the nozzle, due to fuel just leaving the nozzle. The fuel of this local part is unburned, where its temperature is about 750K. In the middle and front parts of the fuel injection, by air entrainment, the mole density of the fuel is below 0.25. The temperature field shows that the maximum temperature is around 1300K in the middle and front of the injection. That also denotes why combustion is very strong in this phase, which matches with the piston shape. Its mole fraction is 0.03 ~ 0.25, and the air equivalence ratio less than 0.6. It is obvious that the diffusion combustion is the main combustion mode of normal Diesel engine, namely "fire contains fuel" form. In other parts of the combustion chamber, the temperature is about 1000K.

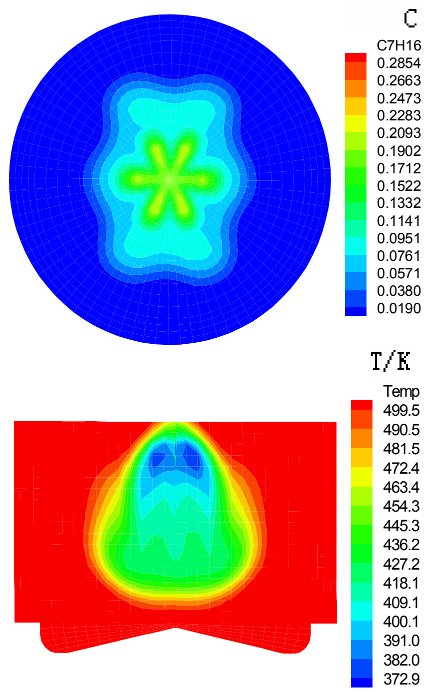
Concentration and temperature fields at 1^0CA before the end of injection are shown in Fig. 3(b). As the injection has almost completed, the injection distance is short. The region of high temperature is enlarging, and the maximum temperature near 2000K. The high temperature in the expanded region promotes the NO_X and smoke formation. Therefore, there are high NO_X and smoke emissions in the normal temperature combustion mode. As shown in Fig. 3(b), the in-cylinder average temperature is a high value of 1100K at 11^0CA BTDC. With piston moving up and fuel combustion, the temperature is higher, so the CO and HC emissions are low.

4.2 Formation of Mixture in HCCI Combustion Mode

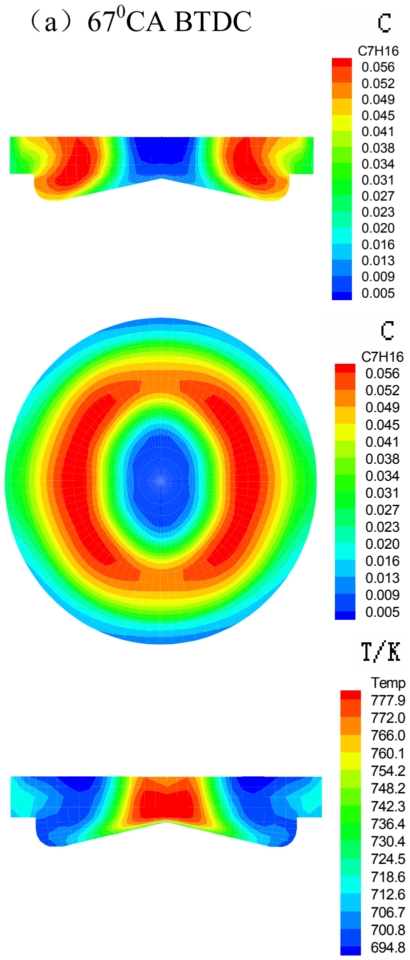
The simulation results of in-cylinder fuel concentration and temperature fields in the process of mixture formation in HCCI mode are shown in Fig.4. Injection begins from 82^0CA BTDC with its duration of 25^0CA. At 15^0CA after the start of injection, the fuel injection develops and already near the piston top side. In the middle and front parts of the fuel injection, the fuel mole fraction is about 0.15 and its temperature is near 450K, as Fig.4(a) shown. In addition, ten nozzles are divided into two types: the diameter of upper 6 nozzles is 0.25mm, while the below 4 nozzles is 0.18mm. The below 4-sprays are close to the upper 6-sprays. Therefore, the shape of the middle concentration field is rectangle, as Fig.4 shown. The in-cylinder fuel concentration and temperature fields at 25^0CA BTDC are shown in Fig. 4(b), when injection has already finished with the rich mixture forming a zone, and the fuel continues mixing. The temperature of the zone is about 700K, where its mole fraction is about 0.05. The first phase reaction takes place slowly near the zone, where its temperature is about 750K, which is determined by its concentration and temperature. Most of the space in the combustion chamber is full of lean mixture, namely, the air equivalence ratio is over 3.5. Furthermore, as the temperature field shown, due to endothermic fuel vaporization, the temperature of rich homogenous mixtures is lower than the lean homogenous mixture.

According to (b), the in-cylinder mixtures are divided into three parts: the air/fuel equivalence ratio at the circular zone is about 0.4, the air/fuel equivalence ratio in the lateral zone is about 1, and the air/fuel equivalence ratio in the center zone is about 3.5. It is obvious that these mixtures are stratified homogeneous.





(a) 67^0CA BTDC



(b) 25^0CA BTDC

Fig.4 In-cylinder Fuel Concentration and Temperature Fields in the Mixture Formation Process under the HCCI Mode

As the density and temperature field shown, the rich mixtures accumulate in the ω-type combustion chamber. The mixtures near the cylinder wall are somewhat rich and impinging wall while the mixtures in the center of cylinder are too lean to ignite. These result in high unburned HC emissions and low thermal efficiency. Its experimental value is 18.66%. With the piston moving up, the mixtures are more homogeneous and multipoint burning (this is HCCI combustion). The experiment shows that the value of NO_X and smoke emissions is zero. The combustion systems must be changed in order to avoid fuel impinging wall.

5 Conclusions

(1) The normal combustion mode of Diesel engine is the diffusion combustion, namely "Fire contains fuel" form. This combustion mode results in the formation of NO_X and smoke.

(2) The normal combustion mode results in the in-cylinder high average temperature, and low CO and HC emissions.

(3) Diesel HCCI combustion is a premixed combustion mode by homogeneous mixture before combustion, which reduces the emissions of NO_X and smoke.

(4) Diesel HCCI combustion results in high CO emissions.

(5) HCCI combustion is actualized by early-injection, which results in high unburned fuel emissions and low thermal efficiency if the combustion system is unchanged. Therefore, the combustion system must be machined.

References:

[1] Zhou Longbao, Liu Xunjun, Gao Zongying, *The Iinternal Engine Principle*, Machinery Industry Press, 1999 (in Chinese)

[2] Rudolf H Stanglmaier, Charles E. Roberts, *Homogeneous Charge Compression Ignition (HCCI): Benefits, Compromises, and Future Engine Applications*, SAE Paper, 1999-01-3682, 1999

[3] Lin Tiejian, Su Wanhua, Pei Yiqiang, *Study of Auto-ignition and Burning Rate Control on the Process of Diesel Engine HCCI Combustion*, Natural Science Progress, Vol. 13, No.5, 2003, pp. 518—522.

[4] Li Degang, Huang Zhen, Qiao Xinqi, et al, *Effects of Compression Ratio and CO_2 on the Process of HCCI Combustion Fueled with DME*, Journal of Shanghai Jiaotong University, Vol.39, No.5, 2005, pp.169-172. (in Chinese)

[5] Golovitchev, V.I., Nordin, N., Jarnicki, R., et al, *3-D Diesel Spray Simulations Using a New Detailed Chemistry Turbulent Combustion Model,* SAE Paper, 2000-01-1891, 2000

[6] Zheng Zhaolei, Yao Mingfa, *Numerical Study on the Chemical Reaction Kinetics of N-heptane for HCCI Combustion Process*, Transactions of CSICE, Vol.22, No.3, pp.227-234. (in Chinese)