

Miller-PCCI Combustion in an HSDI Diesel Engine with VVT

Yutaka Murata, Jin Kusaka and Yasuhiro Daisho
Waseda University

Daisuke Kawano, Hisakazu Suzuki, Hajime Ishii and Yuichi Goto
National Traffic Safety and Environment Laboratory (NTSEL)

Copyright © 2008 SAE International

ABSTRACT

A variable valve timing (VVT) mechanism has been applied in a high-speed direct injection (HSDI) diesel engine. The effective compression ratio (ϵ_{eff}) was lowered by means of late intake valve closing (LIVC), while keeping the expansion ratio constant. Premixed charge compression ignition (PCCI) combustion, adopting the Miller-cycle, was experimentally realized and numerically analyzed. Significant improvements of NO_x and soot emissions were achieved for a wide range of engine speeds and loads, frequently used in a transient mode test. The operating range of the Miller-PCCI combustion has been expanded up to an IMEP of 1.30 MPa.

INTRODUCTION

Diesel engines have an advantage in terms of fuel economy performance. However, diesel emissions, especially nitrogen oxides (NO_x) and particulate matter (PM), are some major causes of environmental pollution. Therefore, more stringent diesel engine emissions regulations are scheduled in the very near future. The improvements of in-cylinder combustion, fuel technology and exhaust gas after-treatment systems have been combined to develop low-emission diesel engines. From the viewpoint of reducing the burden on the after-treatment systems, application of clean diesel combustion is strongly expected for future environmentally friendly diesel engines.

In recent years, premixed charge compression ignition (PCCI) combustion has been considered as a promising combustion strategy, which has potential to drastically reduce NO_x and soot simultaneously. There have been many PCCI combustion studies indicating combustion controls for diesel emissions reduction [1-6].

NO_x and soot formation regions have been clarified on the universal equivalence ratio (ϕ) - temperature (T) maps [7-9]. In diesel combustion processes, NO_x is generated in local high temperature and near stoichiometric regions. Soot is formed in a certain confined temperature range

and over-rich regions. Therefore, achievement of PCCI combustion is equivalent to the avoidance of these two regions simultaneously during the combustion process. It is a difficult challenge to reveal how to use the remaining clean region on the ϕ -T map.

In this study, a method utilizing the flexibility of a variable valve timing (VVT) system is experimentally proposed to control PCCI combustion over a wide operating range. At the same time, ϕ -T map analyses [10-12] are combined with 3D-CFD to investigate the emissions reduction mechanisms of PCCI combustion.

EXPERIMENTAL ENGINE

Table 1 shows the principal specifications of a single cylinder light duty high-speed direct injection (HSDI) diesel engine used in this study. The engine is a water-cooled four-stroke-cycle diesel engine with four valves. The basic specifications of the engine, the mechanical compression ratio, swirl ratio, combustion chamber geometry and fuel injection nozzle design are those

Table 1 Specifications of HSDI diesel engine

Type of engine	Direct injection, single cylinder, water-cooled, 4-stroke-cycle, 4-valves
Bore x Stroke mm	ϕ 85 x 90
Swept volume cm ³	510.7
Compression ratio	15.4 (Base IVC= -150, ϵ_{eff} =14.6)
Injection system	Common-rail (Max: 160 MPa)
Nozzle design	ϕ 0.153 mm x 5 holes (Spray angle: 145 deg.)
Cavity design	Reentrant (ϕ 40.9 mm)
Swirl ratio	1.86
EGR control	Cooled-EGR
Boost control	Supercharged with intercooler
Valve actuation	with VVT (Variable Valve Timing)
Intake valve (Cam)	-370(open), -150(close) deg.ATDC
Exh. valve (Cam)	-590(open), 355(close) deg.ATDC

Table 2 Test fuel properties

Item	Value	Item	Value
Cetane number	58.3	Sulfur mass ppm	3
Cetane index (JIS K2280)	61.1	CHON mass%	C
Density (15 deg.C) g/cm ³	0.8217		H
Viscosity (30 deg.C) mm ² /s	3.355		O
Flash point deg.C	64.0		N
Lower heating value kJ/kg	43092	HPLC vol.%	Saturation
Distillation deg.C	IBP		80.3
	10%		Olefin
	50%		0.0
	90%		Aromatic
	EP		19.7
			1 ring
			2 rings
			+3 rings
			0.3

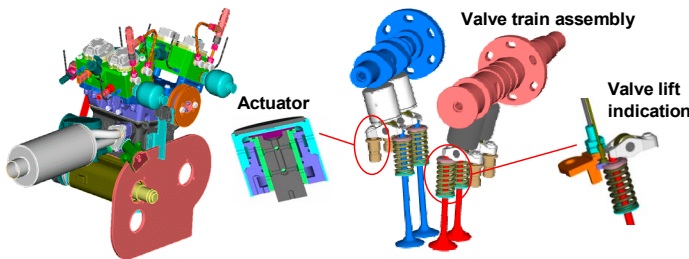


Fig.1 HSDI diesel engine with variable valve timing (VVT) mechanism

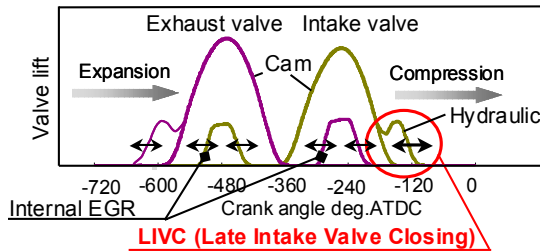


Fig.2 Electro-hydraulic valve actuation (EHVA)

applied in conventional diesel engines. The test fuel used in this study was conventional diesel fuel having a cetane number of 58. The properties of the fuel are listed in Table 2.

The engine is equipped with a common-rail fuel injection system, a low-pressure cooled EGR line, an external supercharging system with an intercooler and variable valve timing (VVT) mechanism. Fig.1 shows the external view and internal components of the VVT system. Lift sensors were installed to monitor each valve lift. As shown in Fig.2, additional valve opening operation can be realized by using electro-hydraulic valve actuation (EHVA). The valve lift curves are determined by a combination of the conventional mechanical valve train and the EHVA. The maximum valve lift operated by the EHVA is limited to 3.5 mm. By means of changing the intake valve closing (IVC) timing, the effective compression ratio $\epsilon_{\text{eff}} (=V_{\text{IVC}}/V_{\text{TDC}})$ can be controlled. In this

study, we focused on using late intake valve closing (LIVC) during the compression stroke to lower the ϵ_{eff} .

3D-CFD CALCULATION METHOD

RANS type 3D-CFD calculations are performed using the KIVA3V code [13] developed by LosAlamos National Laboratory to understand the in-cylinder phenomena occurring in the engine experiments. In order to improve the prediction accuracy of diesel spray and combustion, sub-models of KIVA3V should be modified. The sub-models used in this study are listed in Table 3. In particular, the Kelvin-Helmholtz and Rayleigh-Taylor (KH-RT) sub-models for the fuel droplet break-up developed by University of Wisconsin [14-16] were applied to reproduce the fuel spray atomization processes. Furthermore, the interaction between turbulent mixing and gas phase chemical kinetics for the diesel combustion processes was considered by adopting the partially stirred reactor (PaSR) turbulent combustion sub-model [17] developed by Chalmers University of Technology. The reaction model of diesel

Table 3 Sub-models used in KIVA3V code

	Sub-model
Turbulence	RNG k- ϵ [19]
Drop breakup and atomization	KH-RT [14-16] (Kelvin-Helmholtz/Rayleigh-Taylor)
Drop drag	Drop distortion [20] [21]
Collision and Coalescence	Nordin model [22]
Velocity interpolation	Applied [22]
Vaporization	Single component [23]
Wall impinge	Wallfilm/Splash [24]
Ignition and combustion	PaSR (Partially Stirred Reactor) [17] Diesel oil surrogate mechanism (C ₁₄ H ₂₈) (71 species, 316 reactions) [18]
Heat transfer	Law-of-the-wall [25]

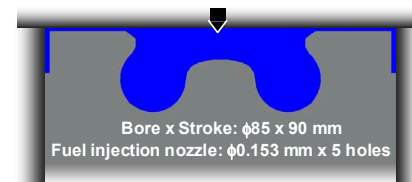


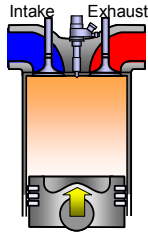
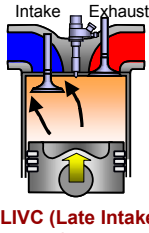
Fig.3 Cavity design and computational mesh

surrogate fuel ($C_{14}H_{26}$) [18] was introduced in the combustion sub-model. The reaction mechanism includes 71 chemical species participating in 316 chemical reactions. The computational mesh was generated by using ICEM-CFD. The combustion chamber geometry and sector grid used in the calculations are shown in Fig.3. The total number of computational cells is about 10,000 at IVC.

COMBUSTION CONCEPT

For suppressing soot emissions, even under a large amount of EGR, it is necessary to improve mixture formation to prevent locally over-rich regions, which cause soot formation. The strategy for realizing PCCI combustion in this study is shown in Table 4. With late intake valve closing (LIVC) by actuating VVT, the effective compression ratio ϵ_{eff} ($=V_{IVC}/V_{TDC}$) is reduced, while keeping the intake valve opening (IVO) timing conventionally constant. Reducing the ϵ_{eff} contributes to lowered compressed gas temperatures near top dead center (TDC), thereby increasing the ignition delay to enhance fuel-air mixing. The exhaust valves are operated by original cam actuation. Accordingly, the expansion ratio is set constant to realize the Miller-cycle. The low effective compression and high expansion ratios are applied to achieve PCCI combustion for low emissions and high thermal efficiency. Conventionally, the IVC timing was set at -150 deg. ATDC during the compression stroke to maximize the intake gas amount. However, in the case of adopting LIVC, part of the intake gas flows back into the intake ports during the valve-opening period, reducing the working gas. Consequently, the overall excess air ratio becomes lower with the same fuel injection quantity per stroke and EGR rate. The possibility of soot reduction from enhanced mixing, even in an overall richer condition, will be determined through the investigation of this combustion concept.

Table 4 Strategy for realizing PCCI combustion

		
	Conventional	LIVC (Late Intake Valve Closing)
Effective compression ratio ϵ_{eff}	ϵ_{eff} (High)	ϵ_{eff} (Low) : Miller-cycle
Expansion ratio	E (High)	
TDC gas temp.	$T_{IVC} \epsilon_{eff}^{k-1}$ (High)	$t_{VC} \epsilon_{eff}^{k-1}$ (Low)
Ignition delay	Shorter	Longer
Fuel-air premixing	X	O (More enhanced)
Intake gas amount	O	X (Reduced)

ϵ_{eff} : Effective compression ratio ($= V_{IVC}/V_{TDC}$)

RESULTS AND DISCUSSIONS

PCCI WITH EARLY FUEL INJECTION AND LIVC

In order to investigate the characteristics of PCCI combustion with early in-cylinder fuel injection, in contrast to conventional diesel combustion, 3D-CFD calculations were conducted for two operating conditions (Table 5). The engine speed was fixed at 1,800 rpm with constant fuel injection quantity per stroke. The load was equivalent to 33 percent, assuming a multi-cylinder engine had a BMEP of 2 MPa at the maximum torque point. In the case of simulating conventional diesel combustion, the fuel injection timing was set at -3.5 deg. ATDC so as to adjust the 50% mass fraction of burned fuel (MFB50%) position at 7 deg. ATDC. For the PCCI case, an earlier fuel injection timing of -40 deg. ATDC with a narrower spray angle was adopted, along with a late IVC timing of -100 deg. ATDC. The LIVC timing lowered the ϵ_{eff} to 10.5 to prevent undesired early ignition of the injected fuel and increase the ignition delay to enhance fuel-air premixing. The expansion ratio, intake and exhaust gas pressures and fuel injection pressure were kept constant as shown in Table 5.

Calculated spatial distributions of in-cylinder combustion temperature and equivalence ratio for the conventional diesel combustion and PCCI combustion are compared in Fig.4. The ignition of the conventional diesel combustion occurs on the outer side of the spray during fuel injection. Note the conventional diesel spatial temperature distribution at 9 deg. ATDC. Typical diffusion-flame combustion was formed with the conventional diesel combustion. Local high temperature and high equivalence ratio regions were found in the combustion chamber.

In the PCCI combustion, the ignition occurred long after fuel was injected. The long ignition delay prepared a well-mixed air-fuel region, resulting in the combustion

Table 5 Calculation conditions for investigating PCCI combustion compared with conventional diesel combustion

	Conventional Diesel	PCCI
	w/o VVT	with VVT
Engine speed rpm	1800	
Fuel quantity mg/st.(Load %)	21.5 (33)	
Effective compression ratio ϵ_{eff} (IVC timing deg.ATDC)	14.6 (-150)	10.5 (-100) : LIVC
Fuel injection timing deg.ATDC	-3.5 (Adjust MFB50% position at 7deg.ATDC)	-40.0
Fuel spray cone angle deg.	145	60
Expansion ratio	15.4 (constant)	
Intake and exhaust gas pressure kPa(gauge)	50	
Fuel injection pressure MPa	160	

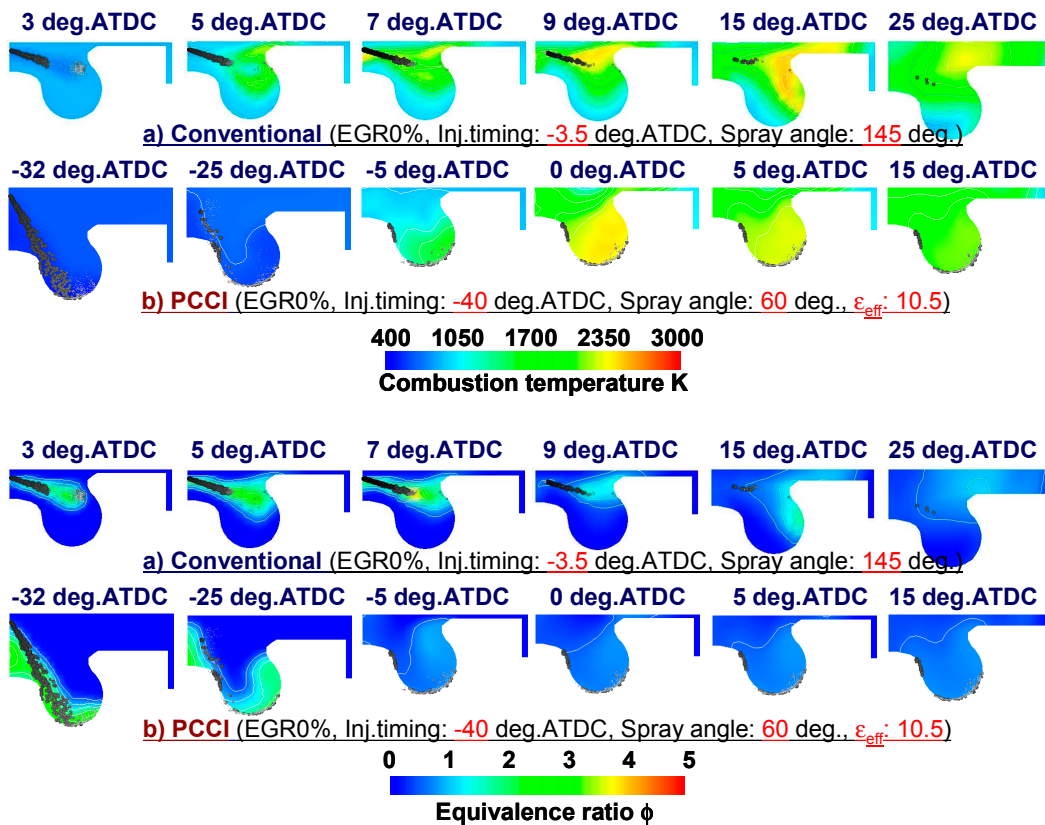


Fig.4 Calculated temporal spatial distributions of in-cylinder combustion temperature and equivalence ratio for conventional diesel combustion and PCCI combustion

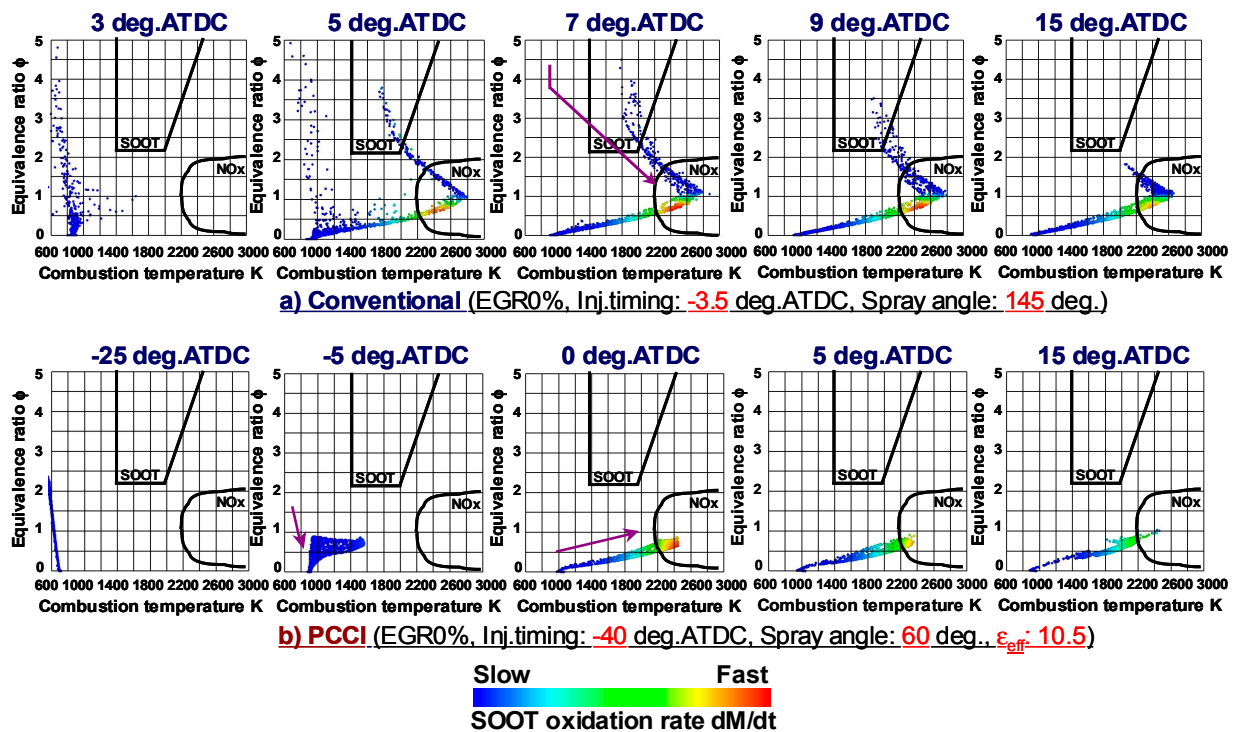


Fig.5 Predicted temporal in-cylinder local equivalence ratio (ϕ) and combustion temperature (T) conditions on the ϕ -T map for conventional diesel combustion and PCCI combustion