

CFD Analysis of VVT/VVA on the Gas Exchange and Fuel-Air Mixing in a Diesel Engine

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ABSTRACT

A three-dimensional simulation was carried out for investigating effects of negative valve overlap (NVO) on gas exchange and fuel-air mixing processes in a diesel homogeneous charge compression ignition (HCCI) engine with early fuel injection. It was found that the case with longer NVO produced a stronger swirl motion and a more significant vortex below the intake valve due to the high annular jet flow through the valve curtain area during the intake stroke. However, there was not much difference in the values of swirl ratio, tumble ratio and turbulence intensity between different NVOs at the end of compression stroke. It was also seen that enlarged NVO not just increased in-cylinder temperature but also improved the temperature homogeneity. With increased NVO, there is a bigger spray shape and more droplets exist in gaps of sprays. This demonstrates that stronger turbulence intensity and higher temperature homogeneity with higher NVO improve fuel vaporization and air-fuel mixing.

INTRODUCTION

With the developing of global economy, the problems of energy crisis and environment pollution become more serious. Automotive engines as one of the main source of petroleum consumption and emissions to atmosphere have received increased attentions. Homogeneous charge compression ignition (HCCI) combustion is one of the most attractive modes for automotive engines to achieve both high efficiency and low NO_x and particulates emissions. In the past decade, HCCI has been widely investigated around the world.

However, several problems must be solved before the HCCI engine can be commercialized, including high CO and HC emissions, limited range of speed and power output, high heat release rate and difficulty in controlling ignition timing. Unlike combustion in spark-ignition or diesel engines, the ignition and combustion of HCCI engines are mainly controlled by chemical kinetics [1, 2],

so controlling the combustion phase and rate becomes more difficult.

Various control strategies, such as variable compression ratio [3], variable exhaust gas percentage [4, 5] and variable valve timing/actuation (VVT/VVA), have been investigated by many researchers. Adjusting the residual gas fraction by VVT/VVA is considered to be the most promising and feasible way for achieving HCCI combustion control [6]. There is a multitude of VVT/VVA systems under development, such as exhaust reinduction using late intake valve opening (IVO) and late exhaust valve closing (EVO) [7], exhaust rebreathing using a second exhaust valve event during the intake stroke [8, 9], in-cylinder air outflowing by late intake valve closing (IVC) [9, 10] and exhaust retention using early EVO and late IVO (also referred as negative valve overlap (NVO)) [11].

Murata et al. [10] demonstrated that ignition timing control of premixed diesel combustion can be realized by optimizing compressed gas temperature by adopting late IVC. Helmantel and Denbratt [9] reinducted the exhausts gas from the exhaust port by using a second exhaust valve opening during the intake stroke, and found that it was an efficient way to serve as exhaust gas recirculation (EGR) for controlling the combustion phase. However, the ringing intensity was found to be increased, which is thought to be result of temperature stratification in the trapped cylinder charge. Caton et al. [12] compared different valve profiles by which exhaust residual may be used to achieve HCCI, and concluded that reinduction strategies by late IVO and late EVC had significant higher efficiency and reduced NO emissions than exhaust retention strategies by NVO, although the fore strategy suffered from higher HC emissions. They also indicated that the best combination of load range, efficiency, and emissions may be achieved using a reinduction strategy with variable intake lift instead of variable valve timing (VVT).

For diesel HCCI engines, the required compression ratio is typical too low for satisfactory performance in cold

start and low load situations, where the temperature at the end of compression is critical and the preparation of homogeneous charge is very hard. One possible way is changing the compression ratio by using IVC. Shi et al. [11] studied the diesel HCCI combustion by directly injecting the diesel fuel into the in-cylinder residual gas during NVO interval, and found that a large amount of residual with a high temperature benefited fuel vaporization and mixing. It was concluded that increasing the NVO benefited the combustion stability and fuel vaporization at low loads.

However, the influence of VVT/VVA on the in-cylinder air motion and the interaction of air motion with the fuel spray are not fully understood yet. It is useful to study VVT/VVA and its effects on the behavior of gases during the induction and exhaust strokes, which will help to understand gas dynamics effects on the fuel-air mixing, the engine volumetric efficiency, emissions and performance [13]. In this study, a three-dimensional simulation was employed to simulate the gas exchange process and compression process up to top dead center (TDC) in a diesel HCCI engine with NVO and early in-cylinder injection. The effect of NVO on charge preparation was discussed in more detail.

NUMERICAL MODEL

METHODOLOGY

The computational fluid dynamics (CFD) code is a version of the KIVA-3V [14] which has been improved by introducing several submodels including Han and Reitz heat transfer model [15] and a piston-ring crevice flow model [16]. The turbulent flows within the combustion chamber are modeled using the RNG turbulence model, modified for variable-density engine flows [17]. Spray models are included in the code to account for spray atomization, breakup, collision, vaporization and spray/wall interaction [14]. The spray is represented by a particle method. The interaction of the spray and the gas flow is accounted for through coupling source terms in the governing equations. Diesel fuel model DF2 ($C_{12}H_{26}$ Cummins model) was used in this study, which was assembled by T. L. McKinley of Cummins Engine Company. A moving-grid method is used to trace piston and valves movement.

COMPUTATIONAL GRID

The computational domain includes intake ports and exhaust ports, the cylinder and the piston bowl, as shown in Fig. 1. Hexahedral cells have been preferred for the grid generation. The number of cells varies from around 45,000 cells at top dead center (TDC) of compression, to 150,000 cells at bottom dead center (BDC). It has a typical grid size of 2~3 mm. Kim et al. [18] indicated that KIVA-3V results with the coarser mesh (the typical size of $2.2 \times 2.2 \times 3.0 \text{ mm}^3$) are shown in some confidence of accuracy after comparing the results from two different mesh density. It was observed that a resolution of 2 mm in the axial direction did not affect the

results and the difference was less than 5% in spray vaporization simulations by Juneja et al. [19]. So for the present study, the solution obtained with the reference grids was considered sufficiently accurate.

INITIAL AND BOUNDARY CONDITIONS

The computations were conducted from BDC at the end of expansion stroke (-180°CA , 0°CA relative to intake TDC) to TDC at the end of compression stroke (360°CA). Initial thermodynamic and turbulence quantities were assumed to be uniform in the ports and the cylinder. In order to have a reference for comparisons, the same initial and boundary conditions were used for all the runs. Constant pressure boundary conditions were assigned at inlets and outlets, so the dynamic effects were neglected. The details of the initial conditions for in-cylinder, intake port, and exhaust port regions are shown in Table 1. Those values were estimated based on the low load and cold start conditions.

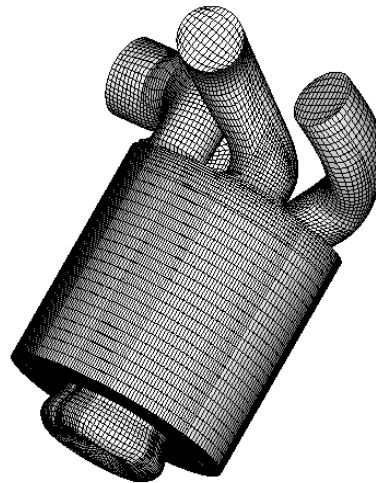


Fig. 1 Computational grids at BDC

Table 1 Initial Conditions

	Temperature (K)	Pressure (atm)
In-Cylinder	460	2.0
Intake Port	300	1.0
Exhaust Port	380	1.0

TEST ENGINE

The test engine simulated in this paper is derived from a high-speed direct-injection (HSDI) diesel engine with a displacement of 0.5 liters/cylinder. A major modification to the original engine is decreasing the compression ratio from 18.2 to 14.3 by increasing the squish height in order to suit for HCCI combustion. The engine specifications are listed in Table 2. The engine is

designed with two direct intake ports whose inlet is tangential to the wall of the cylinder, as shown in Fig. 2.

Table 2 Engine Specifications

Displacement (single-cylinder)	0.5 L
Bore	86.0 mm
Stroke	86.0 mm
Connection Rod Length	160.0 mm
Squish Height	1.81 mm
Geometry Compression Ratio	14.3:1
Swirl Ratio	1.4
Speed	2000 rpm
Combustion Chamber	In-piston Mexican Hat
Fuel	Diesel
Wall Temperature	400 K
Firedeck Temperature	400 K
Piston Temperature	400 K

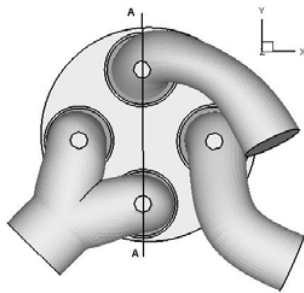


Fig. 2 Layout of the intake and outtake ports

Table 3 Injection Parameters

Injector Protrusion	1.3 mm
Injector Nozzle Numbers	6
Hole Injector Diameter	0.159 mm
Injector Cone Angle	60°
Injection Timing	300 °CA (60 °CA BTDC)
Injection Duration	30 °CA

The injector was located in the center of the cylinder head. According to the research results from Lechner et al. [20] and Kim et al. [21], a narrow cone angle nozzle with spray cone angles of 60° in conjunction with early injection timing was used in this study to reduce fuel deposition on the cylinder wall. The injection parameters are listed in Table 3.

RESULTS AND DISCUSSIONS

Four different valve profiles by varying the NVO have been investigated in this study. The value of NVO is defined as the crank angle difference between EVC and IVO. In all examined cases, EVO and IVC were fixed, and NVO was realized by varying early EVC coupled to a symmetrically vary late IVO. The maximum valve lifts were changed with NVO respectively. Four cases were examined, for which the EVC were 0, -30, -60, -90 °CA (NVO of 0 to 180 °CA), as shown in Fig. 3. These cases are referred as NVO=0°, NVO= 60°, NVO=120° and NVO=180° in the following sections.

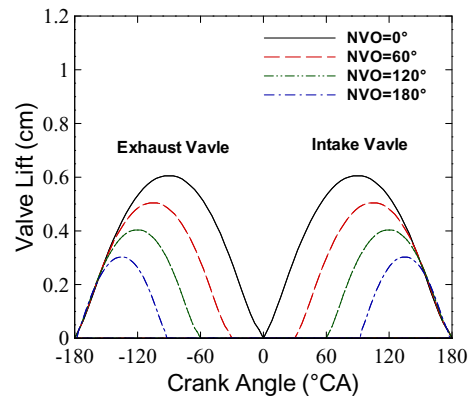


Fig. 3 Valve profiles used for simulation

Effect on Macro Properties

The graphs depicted in Fig. 4-6 show the in-cylinder mass, temperature and pressure with these four NVOs, respectively, and provide an understanding of the processes occurring during the gas exchange and compression strokes. In order to fully understand those results, the figures are discussed jointly.

It can be seen from Fig. 4 that much hot residual gas is trapped in the cylinder for longer NVO cases with early EVC during the exhaust stroke. Because the piston still moves up, the residual is compressed until TDC, which results in significant increments of temperature and pressure for the cases of NVO= 60°, NVO=120° and NVO=180° as shown in Fig. 5 and Fig. 6. It has been found that injecting the fuel into the hot residual gas in the cylinder during the NVO interval can reform the gasoline and improve ignitability [22] and benefit the vaporization of diesel fuel [11].

As the piston moves down and intake valve opened, the combustion product in the chamber is pushed into intake port because the pressure in the chamber is still higher than that in intake ports as shown in Fig. 5. As a result, the in-cylinder mass of residual gas decreases for all NVO cases, shown in Fig. 4. After this period, it comes into the true air induction process, and the fresh air is drawn into the combustion chamber.

From Fig. 6, it can be seen that the temperatures at 180 °CA (at IVC) are 326 K, 334K, 339 K and 342 K for NVO= 0°, NVO= 60°, NVO=120° and NVO=180° respectively. However, it should be noted that it is not always that the longer NVO will leads higher in-cylinder temperature at IVC. As the initial in-cylinder temperature

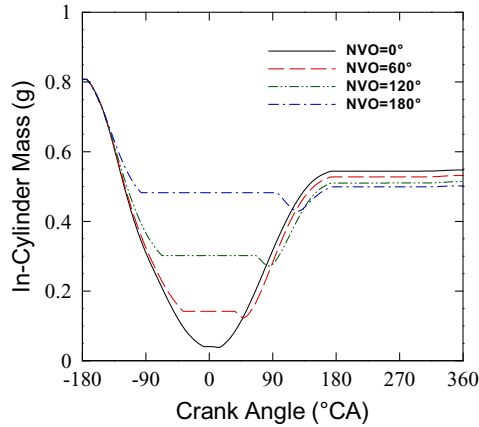


Fig. 4 In-cylinder mass as function of crank angle

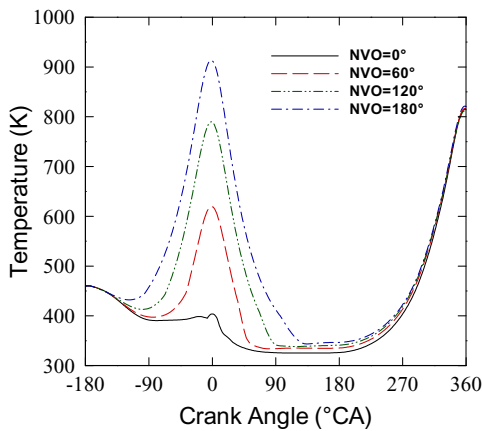


Fig. 6 Temperature as function of crank angle

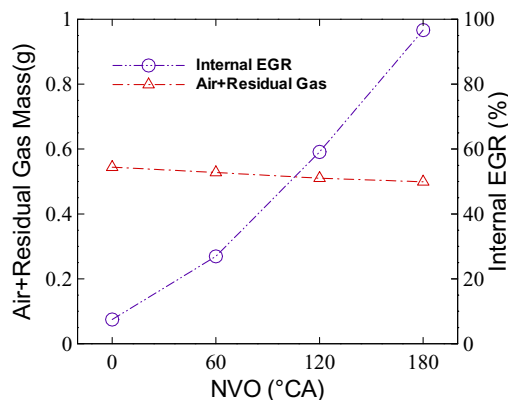


Fig. 8 Effects of NVO on the trapped residual gas and total in-cylinder gas at IVC

At last, the amount of internal EGR for NVO=0° is 8%, while it is as high as 93% for NVO=180°, and the total in-cylinder mass decreases slightly with longer NVO, shown

at -180 °CA is decreased to 380 K, it is interesting to notice that temperature at IVC becomes lower for the longer NVO case due to later EVO as shown in Fig. 7. Therefore, the temperature at IVC is influenced by all the factors including engine speed, in-cylinder temperature and pressure at IVO, valve timing and maximum lift etc.

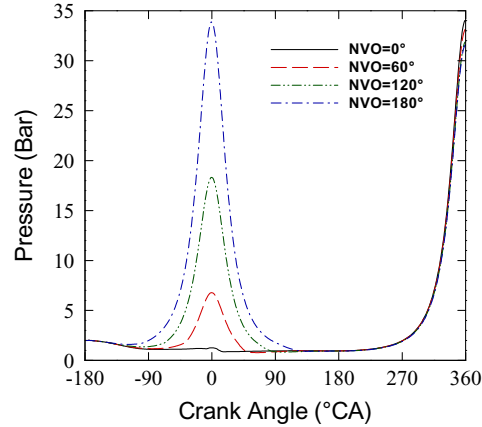


Fig. 5 Pressure as function of crank angle

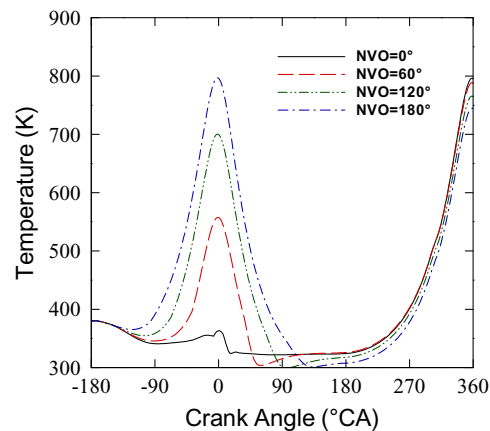


Fig. 7 Temperature as function of crank angle with the initial in-cylinder temperature of 380 K

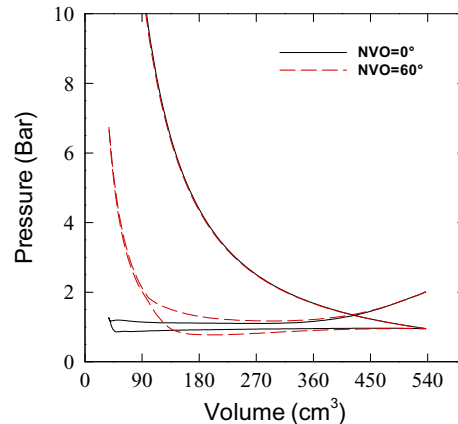


Fig. 9 P-V diagram for NVO=0° and NVO= 60°

in Fig. 8. Because combustion products such as CO₂ and H₂O have higher specific heat than air and tend to reduce compression temperature, the temperature difference at