

Numerical Study of the Effect of Piston Shapes and Fuel Injection Strategies on In-Cylinder Conditions in a PFI/GDI Gasoline Engine

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ABSTRACT

SI-CAI hybrid combustion, also known as spark-assisted compression ignition (SACI), is a promising concept to extend the operating range of CAI (Controlled Auto-Ignition) and achieve the smooth transition between spark ignition (SI) and CAI in the gasoline engine. In order to stabilize the hybrid combustion process, the port fuel injection (PFI) combined with gasoline direct injection (GDI) strategy is proposed in this study to form the in-cylinder fuel stratification to enhance the early flame propagation process and control the auto-ignition combustion process. The effect of bowl piston shapes and fuel injection strategies on the fuel stratification characteristics is investigated in detail using three-dimensional computational fluid dynamics (3-D CFD) simulations. Three bowl piston shapes with different bowl diameters and depths were designed and analyzed as well as the original flat piston in a single cylinder PFI/GDI gasoline engine. An engine operating load of IMEP=3.6 bar was selected to evaluate the effect of piston shapes on the in-cylinder conditions, including flow conditions, fuel stratification patterns, thermal conditions and fuel evaporation ratios. The GDI ratios and direct injection timings were also varied to find the most appropriate injection strategy to achieve optimal in-cylinder condition. The simulation results presented in this study clarified the effect of bowl piston shapes, GDI ratios and direct injection timings on the in-cylinder fuel stratification in a PFI/GDI gasoline engine. They would provide the fundamental knowledge and effective guidance on selecting the optimal bowl piston shape and fuel injection strategy to stabilize the SI-CAI hybrid combustion in the future experimental investigations.

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INTRODUCTION

Controlled Auto-ignition (CAI) combustion has been shown to effectively reduce the NO_x emissions and increase fuel efficiency [1] and it has been subject to extensive studies in the past few decades. However, the lack of direct control on the auto-ignition process and relatively narrow operation range inhibit the practical application of this low temperature high efficient combustion mode [2]. In order to enhance the control of auto-ignition and extend the operation range of CAI combustion, spark ignition (SI) was used to assist CAI combustion. In this case, the early flame propagation induced by spark discharge could effectively control the later auto-ignition [3, 4]. In addition, this hybrid combustion mode could be used to bridge pure SI mode and CAI mode during mode transition when applying to the full load operations in the practical engine [2, 5, 6, 7, 8]. However, the recent study on the SI-CAI hybrid combustion indicated the existence of significant cycle-to-cycle variation (CCV) during mode transition [8, 9, 10, 11].

In the SI-CAI hybrid combustion, the flame propagation is mainly controlled by the transport of heat and active species in the flame front and distorted by the in-cylinder turbulence [12]. The variation of the early flame propagation would in turn affect the later auto-ignition process [13]. The premise of controlling and stabilizing SI-CAI hybrid combustion is to effectively manage the early flame propagation process. The local fuel/air equivalence ratio around the spark plug significantly affects the spark kernel formation and flame propagation process in spark ignition engine. Persson et al. [14] used high speed fuel Planar Laser-Induced Fluorescence (PLIF) to study the effect of fuel stratification on spark assisted compression ignition (SACI) with ethanol as fuel. They found the lower fuel concentration in the vicinity of the spark plug decreased the flame expansion speed with the overall lean mixture (λ 1.4). Middleton et al. [15] investigated the propagation of premixed laminar reaction fronts with different fuel/air equivalence ratios using transient one-dimensional flame simulations and indicated the increased flame burning velocity with the lower fuel/air equivalence ratio increasing from 0.5 to 1.0. The experimental

study performed by Benajes et al. [16] shows that the increase in the directly injected fuel mass has a positive effect on the assistance of the spark in the combustion process for both combustion stability and cycle to cycle control. Williams et al. [17] studied the in-cylinder fuel-air mixing distributions in a firing gasoline-direct-injection engine using PLIF imaging and indicated the reduced mean equivalence ratio around spark plug would increase the 0-10% mass fraction burned (MFB) duration. The previous studies demonstrated the significant effect of equivalence ratio on flame propagation process.

In order to form a desired fuel stratification, the appropriate direct injection strategy in combination with a suitable bowl piston is required [18, 19, 20]. Mismatch between the piston bowl and injection timing would cause inappropriate fuel air mixing and lead to unstable combustion and increased emissions. Xu et al. [21] applied CFD modeling method to optimize piston shapes to form the rich fuel cloud around the spark plug in a GDI engine. The numerical simulations performed by Joh et al. [22] also demonstrated that piston shape had a strong influence on air-fuel interaction and the charge distribution around spark plug. The invention reported in [23] designed a combustion chamber including a bowl piston for a gasoline direct-injection controlled auto-ignition combustion engine. In addition to the piston shapes, the injection strategy also plays an important role on controlling in-cylinder fuel stratification patterns. Earlier injection timing increases fuel surface wetting, resulting in excessive engine out smoke, while later injection timing decreases available time for air-fuel mixing prior to the time of ignition [24]. The results reported in [25] showed that misfire occurred when the charge was either over-mixed and too lean to ignite in the spark plug region or over-penetrated and located away from the spark plug to the intake side of the chamber. The split injection strategies also showed significant influence on the fuel/air mixture cloud formation [26]. Persson et al. [14] proposed the combined injection strategy with both port fuel injection and direct injection to effectively form overall homogeneous mixture with appropriate fuel stratification around spark plug. The simultaneous injection using PFI injectors and DI injectors presented in [20] were able to improve the fuel/air mixture formation and resulted in reduced fuel consumption and torque fluctuations.

In the application of SI-CAI hybrid combustion, the relatively rich mixture around the spark plug is preferred to enhance the robustness of the early flame propagation. On the other hand, the diluted mixture in the outer region is preferred to control the heat release of auto-ignition. In order to effectively manage the fuel stratification, the combination of port fuel injection (PFI) and gasoline direct injection (GDI) is adopted in this study. The overall stoichiometric fuel/air mixture was applied to enable the emission reduction with the conventional three-way catalysts. In order to take advantage of the benefits provided by both PFI and GDI technology, the best matching between the spray and piston shapes is necessary to create appropriate fuel stratification pattern while providing adequate fuel around the spark plug to

attain reliable ignition. The effect of bowl piston shapes and direct fuel injection strategies on the in-cylinder conditions, including flow conditions, fuel stratification patterns, thermal conditions and fuel evaporation ratios, is investigated in detail using three-dimensional computational fluid dynamics (3-D CFD) simulations. Three bowl piston shapes with different bowl depths, diameters and curvatures were designed and modelled in a single cylinder engine. The GDI ratios and direct injection timings were varied to investigate the effect of different injection strategies. A multi-hole injector was selected to perform the numerical study because of its superior performance and low cost [27, 28]. An engine operating load of IMEP=3.6 bar was selected to perform the simulations. The simulation results presented in this study clarified the effect of bowl piston shapes, PFI/GDI fuel ratios and direct injection timings on the in-cylinder fuel stratification in a PFI/GDI gasoline engine. They would provide the fundamental knowledge and effective guidance on selecting the optimal bowl piston shape and fuel injection strategy to stabilize the SI-CAI hybrid combustion in the future experimental investigations.

METHODOLOGY

Numerical Models And Validation

In this study, the commercial CFD software STAR-CD™ was adopted to perform the simulations. A set of models, including MPI2 nozzle model [29], Reitz-Diwakar atomization and breakup model [30], were employed to simulate the fuel spray process. The O'Rourke model and Bai model [31] were adopted to consider the inter-droplet collision and wall impingement. Reynolds-Averaged Navier Stokes (RANS) approach was applied with RNG k- ϵ turbulence model in the simulations. Pressure-Implicit with Splitting of Operators (PISO) algorithm was used to solve the equations. The heat transfer was implemented through the general form of the enthalpy conservation equation for the fluid mixture [32]. The Angelberger wall function [33] was used for the simulation of the wall heat transfer.

A multi-hole injector was selected in this study because of its superior performance and low cost [27, 28]. The optical visualizations of the spray plume in a constant volume vessel were performed to validate the spray models. The optical measurements were carried out using Schlieren method [34]. The injection pressure and fuel temperature were 10 MPa and 293 K. The background pressure and temperature were 0.1 MPa and 293 K. The detailed procedure of the measurement could be found in [34].

In the simulation, a fine mesh with 1, 140, 000 grids with 1.5 mm grid size was adopted. The time step in the simulation was 0.005 ms. The parameters of the adopted models were adjusted to reproduce the spray measurements. Figure 1 compares the liquid droplets image of the spray plume from optical measurements and simulations. The spray liquid tip penetration is shown in Figure 2. The results show reasonable agreement between the modelled spray and the measurements.

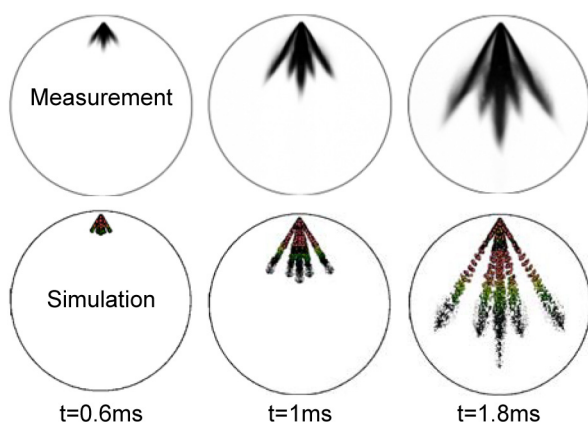


Figure 1. The spray plume from optical measurement and simulations.

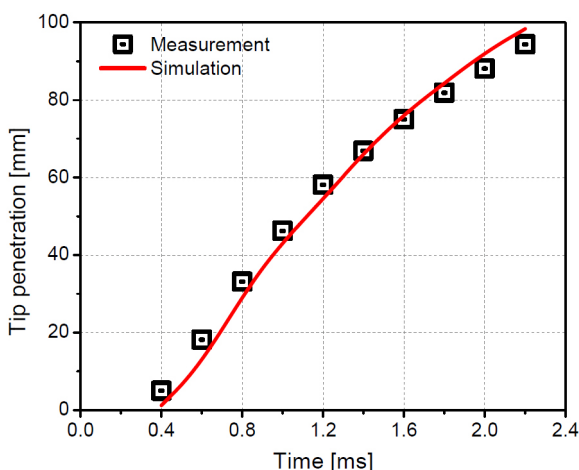


Figure 2. The spray liquid tip penetration from measurement and simulation.

Piston Shapes Design

In this study, different piston shapes were designed from the origin flat piston in a single cylinder engine [2]. The engine specifications are given in Table 1. The engine comprises a Ricardo Hydra single cylinder block and a specially designed cylinder head, which is equipped with a 4-variable valve actuation system (4VVAS) with BMW's Vanos and Valvetronics on both the intake and exhaust camshafts. The engine enables both port fuel injection (PFI) and gasoline direct injection (GDI) with one injector mounted in the intake port and another direct injector mounted at the intake side.

Table 1. Engine specifications.

Bore	86 mm
Stroke	86 mm
Displacement	0.5 L
Compression ratio	10.66
Combustion chamber	Pent roof / 4 valves
Fuel injection	PFI/GDI
Fuel	Gasoline 93 RON
Intake pressure	Naturally aspirated
Throttle	WOT

Figure 3 shows the newly designed bowl pistons. In order to maintain the compression ratio, boss-shaped bodies were added on the flat piston for bowl pistons and different piston bowl shapes were then cut from the boss-shaped bodies. Compared to Piston A, Piston B has larger bowl diameter. Piston A and B have been manufactured and ready for engine tests. Basing on Piston A and B, a deeper piston bowl was designed in Piston C.



Figure 3. Newly designed bowl pistons.

Case Setup

The moving meshes for flat piston, Pistons A, B and C were generated in ES-ICE using the mapping method. The engine mesh with Piston A is shown in Figure 4 as an example. All the four meshes have similar grid size (around 0.8 mm). In order to systematically evaluate the effect of piston shapes and fuel injection strategies on the in-cylinder conditions, an engine operating load of IMEP=3.6 bar was selected to perform the simulations. Table 2 shows the specific operation conditions. The stable SI-CAI hybrid combustion was achieved by utilizing both external exhaust gas recirculation (EGR) and internal EGR achieved by the negative valve overlapping (NVO) strategy. It should be noted that the residual gas fraction (RGF) before the combustion shown in Table 2 includes both internal and external exhaust gases and was estimated by the corresponding one-dimensional simulations based on GT-power. A large amount of residual gas was used to dilute the combustion process and lower the combustion temperature to achieve high thermal efficiency and low NOx emission. The other experimental detail could be found in reference [8] and [13].

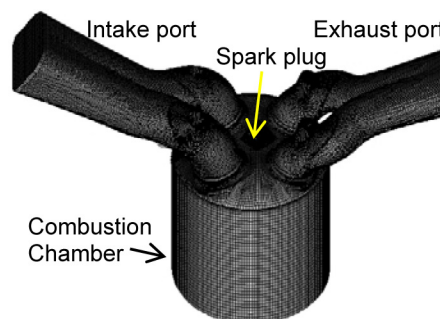


Figure 4. Engine mesh with Piston A.

Table 2. Operation conditions.

Indicated mean effective pressure (IMEP)	3.6 bar
Exhaust valve open (EVO)	177 °CA ATDC
Exhaust valve close (EVC)	254 °CA ATDC
Exhaust valve lift (EL)	1.9 mm
Intake valve open (IVO)	494 °CA ATDC
Intake valve close (IVC)	603 °CA ATDC
Intake valve lift (IL)	5.0 mm
Spark Timing	685 °CA ATDC
Fueling Rate	13.4 mg/cycle
Lambda	1
Residual gas fraction (RGF)	40 %
Engine Speed	1500 r/min

Table 3. Simulation cases and computational conditions.

Cases	Piston Type	GDI ratio	GDI SOI [°CA ATDC]
F-0.35-680	Flat Piston	0.35	680
F-0.35-660	Flat Piston	0.35	660
F-0.35-640	Flat Piston	0.35	640
F-0.2-680	Flat Piston	0.2	680
F-0.2-660	Flat Piston	0.2	660
F-0.2-640	Flat Piston	0.2	640
A-0.35-680	Piston A	0.35	680
A-0.35-660	Piston A	0.35	660
A-0.35-640	Piston A	0.35	640
A-0.2-680	Piston A	0.2	680
A-0.2-660	Piston A	0.2	660
A-0.2-640	Piston A	0.2	640
B-0.35-680	Piston B	0.35	680
B-0.35-660	Piston B	0.35	660
B-0.35-640	Piston B	0.35	640
B-0.2-680	Piston B	0.2	680
B-0.2-660	Piston B	0.2	660
B-0.2-640	Piston B	0.2	640
C-0.35-680	Piston C	0.35	680
C-0.35-660	Piston C	0.35	660
C-0.35-640	Piston C	0.35	640
C-0.2-680	Piston C	0.2	680
C-0.2-660	Piston C	0.2	660
C-0.2-640	Piston C	0.2	640

In this study, the effects of piston bowl shapes and injection strategies on in-cylinder conditions were studied in motored operations. In the simulations, the inlet temperature and pressure were fixed at 355 K and 0.99 bar. The initial in-cylinder temperature and pressure were 571 K and 0.49 bar, respectively. These conditions were estimated by the corresponding one-dimensional simulations using GT-power. The wall temperature for cylinder head, piston head and cylinder liner were 400K, 442K and 371K respectively. The CFD simulations were carried out from IVO to top dead centre

(TDC), including the direct injection process. The intake mixture prepared by the port fuel injection was set as the homogeneous fuel/air mixture. It should be noted that the overall in-cylinder equivalence ratio after direct injection for all cases was kept stoichiometric. Table 3 shows all the simulation cases and the corresponding computational conditions. The case name of "F-0.35-680" refers to the case with Flat Piston, a GDI ratio of 0.35 and a start of injection (SOI) of 680 °CA. The compression ratio was kept at 10.66 and the total in-cylinder mass after the direct injection was kept constant for all the cases studied.

RESULTS AND DISCUSSION

In-Cylinder Flow Conditions

Figure 5 shows evolution of the in-cylinder swirl ratio (SR), tumble ratio (TR) and cross tumble ratio (CTR) before injection. The equation used to calculate the swirl ratio is given as following [35]:

$$SR(\theta) = \frac{\sum_i^m v_i(\theta) r_i(\theta) V_i(\theta) \rho_i(\theta)}{\frac{2\pi n}{60} \sum_i^m r_i(\theta)^2 V_i(\theta) \rho_i(\theta)} \quad (1)$$

where n is engine speed, θ the crank angle, i the cell number, $v_i(\theta)$ the cell volume, $\rho_i(\theta)$ the cell density, $v_i(\theta)$ and $r_i(\theta)$ are the tangential velocity and radius respectively in the cylindrical coordinate with z axis as the swirl axis. With the same method, the swirl axis along with the cylindrical coordinate system in Equation (1) is then replaced as the tumble/cross tumble axis to evaluate TR/CTR. The tumble/cross tumble axis is parallel to y/x axis and crosses the central point between maximum and minimum z value of the combustion chamber.

For simplicity, Figure 5 only compares the cases with GDI ratio=0.35. As expected, the newly designed piston shapes have little effect on the intake process, indicating any modification of the in-cylinder conditions would only come from the interactions between spray jets and piston shapes.

In order to reveal spatially resolved in-cylinder conditions the cylinder volume is divided into seven cylindrical zones as shown in Figure 6. In addition, a Spark Zone is defined as a spherical volume of 20 mm in diameter around spark plug.

It has been found that the high mean flow velocities and turbulence levels can shorten combustion duration in the lean mixtures [36]. Figures 7 and 8 show the mean flow velocity magnitudes (V_m) of the whole combustion chamber and Spark Zone respectively. Overall, large GDI ratio increases the in-cylinder flow velocity at all injection timings. As the injection timing is advanced, the effect of piston shape is weakened. The mean velocity magnitudes are similar for different piston shapes at SOI=660 °CA and 640 °CA. In addition, the relatively