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CFD Modeling and Experimental Validation of Different Piston Crown Designs in an HCCI Engine Fuelled with ISO-Octane

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Abstract: HCCI combustion incorporates the advantages of both Spark-Ignition (SI) engine and Compression Ignition (CI) engines. The homogeneous mixture is inducted into the cylinder without throttling losses and compressed until the mixture reaches the auto-ignition point then combustion occurs spontaneously without discernable flame propagation. This feature helps to improve engine performance while producing a relatively high thermal efficiency. In the present study, three-dimensional CFD calculation was used where mesh creation and specific zone name with different topologies of each zone has been meshed separately using ANSYS. Fluent was used to model combustion phenomenon in HCCI engine. The validation and simulation were conducted based an HCCI single-cylinder engine fuelled with gasoline in a 4-stroke engine at an engine speed of 1500 rpm with compression ratio of 11.7:1 evaluated using three split injection. Combustion parameters such as cylinder pressure, temperature and heat release rate were obtained from the validation work. The CFD Model yields good results for experiment and CFD simulation. The study focused on how different piston crown designs affect the performance of the HCCI engines. Three different designs have been created and evaluated through CFD analysis where all other engine-operating parameters were kept the same as the experimental work. The pistons names A, B and C for simplicity. Then, the study will analyze the in-cylinder pressure, in-cylinder temperature, heat release rate, turbulent kinetic energy, indicated mean effective pressure and power output of different piston designs and evaluate the most suitable piston to be used in HCCI engines in order to improve the engine performance. The results demonstrate the capability of improved piston crown design in HCCI engine to reduce the levels of gas emissions from engines. All pistons in the investigation reached a peak pressure and temperature above the experiment, pistons A had the highest peak pressure and temperature followed by pistons B and C, respectively. Compared to other piston crown designs, the piston A has the highest power output caused by high peak pressure towards the end of combustion that leads to passable diffusion combustion. Piston A's design could be used in an HCCI engine configuration to improve engine performance.

Key words: HCCI engine, piston crown designs, numerical simulation, performance, pressure, Malaysia

INTRODUCTION

A Homogeneous Charge Compression Ignition (HCCI) engine is a relatively new mode of combustion process which has in the recent times generated great interest for use in a vehicle and the generation of stationary power (Khaliq *et al*., 2014). The HCCI integrates attributes of the conventional diesel and gasoline engines where the former produces a high efficiency engine while the latter incorporates low emissions levels (Zhang *et al*., 2014). HCCI is the auto-ignition process by which the combustion occurs instantaneously when the fuel-air mixture has enough chemical activation energy at the end of compression stroke. Thus, both high efficiency and low emissions can be obtained (Najafabadi *et al*., 2013). For HCCI engine, gasoline fuel can operate cleaner than diesel fuel to attain higher thermal efficiency (Hairuddin *et al*., 2014).

A study conducted by Hunicz *et al*. (2015) on combustion injection timing established that different strategies for forming mixtures such as port fuel injection, early and late direct injection and multiple injection, help to overcome the problems associated with injection time in HCCI engines. The study realized that certain control techniques including compression ratio, recirculation of exhaust gas as well as variable injection angle help in attaining HCCI combustion. This technique for controlling the rate of fuel injection depends on the infusing little

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amount of fuel toward the start of the injection with a specific end goal to confine the initial rate of heat released and subsequently to keep the Nitric Oxide (NO_x) generation at bare minimum. At the instance of injection, the pressure of injection is expanded to improve blending and diminish emission of soot (Das *et al*., 2015). Previous studies on multiple injection of the engine is a parameter used to minimize soot emission and the amount of NO_x emitted into the environment (Cao *et al*., 2007; Contino *et al*., 2013; Coskun *et al*., 2014; Das *et al*., 2015; Hunicz *et al*., 2015; Gugulothu and Reddy, 2015; Rezaei *et al*., 2013; Zheng *et al*., 2015b) indicated that there is a separate fuel injector for every cylinder that shoots fuel into the combustion chamber. The study noted that the increased benefit of combining different injection systems improves emissions due to better control processes of combustion. Multiple injection has been shown to be a vital tool for immediate decrease of NO_x emissions for the engines when the injection timing is put into consideration. Thus, investigating the multiple injections on ignition procedure is worth investigation for HCCI engines (Li *et al*., 2013). Their efficiencies can be maximized by increasing the compression ratio and adopting a faster combustion rate (Canakci, 2012).

With this regard, piston shapes and designs can assist in establishing an optimum performance with less emission and fuel consumption. Generally, there are three commonly known types of pistons designs in HCCI gasoline fueled engines the Flat piston, the bowl piston and the dome piston. The choice of the design depends on the desire of the manufacturer (Shaver *et al*., 2009). There is therefore, need to understand in-cylinder fluid dynamics, since, it has been noted to be unsteady, three-dimensional and turbulent hence challenging to study. High-quality mesh and the use of an appropriate valve lift profile were some aspects associated with a predictable flow structure.

The flat-top piston has several merits including less surface area, so, it is lighter with a shorter, faster heat path to the cylinder wall. In addition, the piston crown is in tension under load, the valves close fast, the opening isn't masked by chamfer, the piston shape does not affect the entry and exit angles of the valves and the combustion chamber is a true hemisphere (Zheng *et al*., 2015a). The piston bowl is commonly utilized in gasoline engines. The HCCI engines do not have ignition phase, so that, the piston crowns may form the combustion chamber (Zheng *et al*., 2015a). Such engines use pistons with different shaped crowns as much as the direct injection is becoming popular, gasoline engines also use the same type of pistons. The shapes of the piston bowl usually manage the movement of the air and fuel as pistons moves up for the compression stroke. The fuel and the air swirl into a vortex before combustion take place thus, creating a better mixture (Kakaee *et al*., 2016). By influencing the fuel or air mixture, one can achieve better and more efficient combustion. Therefore, it is important to emphasis that bowl pistons do have different shapes which are commonly designed to reduce fuel consumption.

Model description: The modified model presented in this study is based on a model presented in the past by Roychowdhury *et al*. (2002). It describes the processes of combustion in HCCI engines during the part of the engine cycle on the effects fuel injection strategies and mass fuel shares between the injections. Fuel injection at an early stage of "Negative Valve Overlap" NVO, during exhaust compression, enables exhaust-fuel reactions. These processes produce some quantities of auto-ignition promoting species, established that for multiple injection, cases with various injection timing such as -60, 20 and 300 Crank Angle Degrees (CAD). In this study, simulations were carried out for a single-cylinder, 4-stroke gasoline engine, the rotational speed of 1500 rev/min that run in HCCI Mode (Hunicz *et al*., 2015). The specifications of the engine are shown in Table 1.

The engine was equipped with a mechanical variable valvetrain which enabled control of valve lifts and timings as shown in Table 2 and Fig. 1 show the general profile of a valve lift used for HCCI engine model. The Intake Valve Opening (IVO) and Exhaust Valve Closing (EVC) locations are varied at different rotation angle (Hunicz *et al*., 2015).

These valves help the flow of fuel and air together and also allow burned gases to be exhausted from the engine. These valves give swirl and squish to fuel-air mixture. Swirl is meant by the movement with no apparent pattern while squish is used where the fuel-air mixture is squished under the circumstances where the pressure is high due to the rise in temperature.

Table 1: Engine specifications

Engine types	4-stroke
Number of cylinder	1
Engine speed (rpm)	1500
Number of valves	2
Displaced volume (cm^3)	498.5
Bore (mm)	84
Stroke (mm)	90
Connecting rod length (mm)	201.5
Compression ratio	11.7
Intake pressure (MPa)	0.13
Intake temperature (K)	323
Injection strategy	Split (1/4, 1/4, 0, 1/2)

Table 2: Valvetrain settings

Variables	Values
Intake Valve Opening (IVO)	82°CA
Intake Valve Closing (IVC)	212°CA
Intake valve lift	3.6 mm
Exhaust Valve Opening (EVO)	521°CA
Exhaust Valve Closing (EVC)	640°CA
Exhaust valve lift	2.9 mm

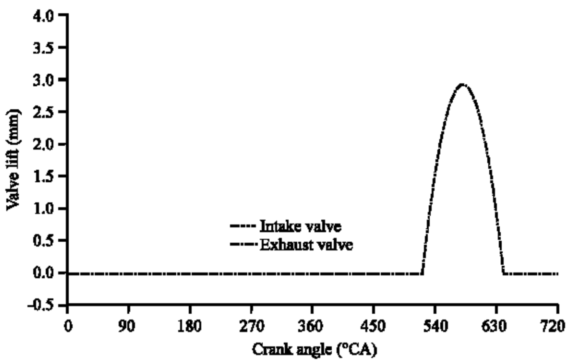


Fig. 1: General valves profile

Geometric model: In this study, the geometry of the HCCI engine's combustion chamber was modeled by CATIA Software and exported into fluent. A three-dimensional CFD calculation was used where mesh creation and specific zone name with different topologies of each zone has been meshed separately using ANSYS. Fluent needs one to provide an initial volume mesh with a corresponding mesh topology that the following topologies in this section can conveniently be utilized to appraise the dynamic mesh automatically. Despite these remarkable exertions, fluent does not need one to set up all in-cylinder problems expanding the mesh topology. When one develops the mesh for the in-cylinder model using any mesh generation tools, one needs to take care of regions that move, deforms or stationary and come up with these regions with appropriate cell shapes. The piston surface was extending on its perimeter and approached the cylinder head close to TDC that generated the amount of squish on the basis of an experiment (Hunicz *et al*., 2015). Figure 2 present the computational domain of the 3D combustion chamber geometry with exhaust and inlet valves. This modeling gives us a clearer idea of how an HCCI engine should look like. The geometry was established at zero crank angle position at TDC. Figure 3 present bowl shape piston of the experiment from (Hunicz *et al*., 2015) was utilized for validation.

Mesh generation: Mesh generation is one of the crucial steps for the CFD process. It was necessary to use the model mathematical accuracy, since, it could apprehend

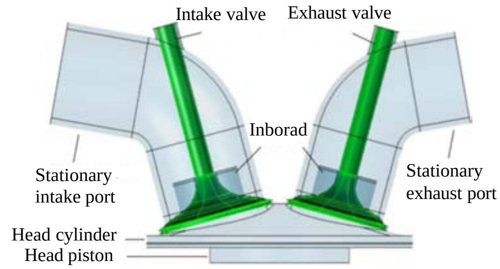


Fig. 2: Geometry of 4-stroke gasoline engine at TDC position modeled by CATIA Software

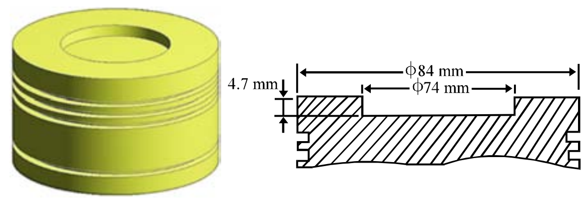


Fig. 3: Piston shape used for combustion model validation

for the quality of the mesh developed. As such, ANSYS Software was used to prepare the mesh, since, it could produce both structured and unstructured types. To determine the piston performance with varied crowns, it was necessary to rely on quadrilateral and triangle structure mesh. As the pistons travels downwards from TDC to BDC position, structure mesh (quadrilaterals and triangles) was the best option for generation of mesh. These mesh domain were fully structured to contain around 177,987 cells at TDC and 347,385 at BDC position as shown in Fig. 4, required to extend the re-meshing area to accommodate the valves if they are extended. For this to done, one needs to specify the dynamic layering region adjacent to the piston surface, so that, it can move alongside the piston until there is some specified distance from the TDC position. Going beyond this cutoff distance would mean that the layering zone should be stopped and that the piston wall should be allowed to continue to the BDC position. Based on the fact that there will be relative motion between the non-moving dynamic layering and the piston head surface, cell layers will be further be added given that the ideal layer height criteria desecrated.

Boundary and initial conditions: The turbocharge produces very little pressure initially at low temperature, a simple turbine will be used to increase the intake pressure of the HCCI engine. The initial values obtained from the conducted experiment that was measured and set to 0.13 MPa absolute. The system has been fitted with heaters heating is used to keep the emission of HC and

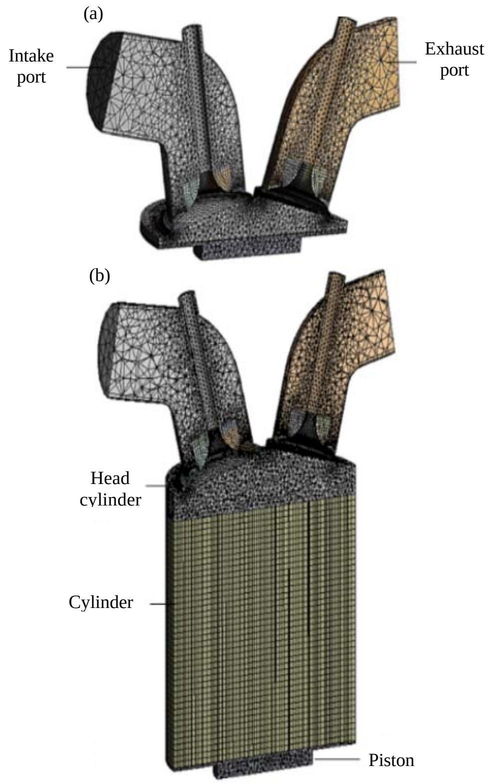


Fig. 4: Section meshed domain geometry model piston: a) at TDC and b) at BDC

CO to a low-level (Hunicz *et al*., 2015). Intake charge temperatures was projected to be around 323 °K at a minimum temperature for ready engine working conditions. The initial turbulence intensity was set at 4% of the mean flow which is quite sufficient for fully turbulent fluid flow, hydraulic diameter of 32 mm. The piston's top surface and valves are defined as the moving wall and others are anon-moving wall. All wall temperatures are set at 298°K (Roychowdhury *et al*., 2002). The fuel type used was a multiple-ignition-stage fuel gasoline in a single-cylinder engine setup with four strokes, direct injection and fuel mass of 18 mg per engine cycle (Hunicz *et al*., 2015).

Turbulence model: In this study, the Re-Normalisation Group (RNG) theory of turbulent model is utilized for examining the physical singularities involved in the alteration of kinetics energy. The k-ε Model was selected as swirl dominated flow and compressibility effects model, since, frequently recommended in the literature (Coskun *et al*., 2014; Harshavardhan and Mallikarjuna, 2015; Jia and Xie, 2006; Wang *et al*., 2015). It was equivalent for k-ε Model but keeping the advantages to include the effects of the swirl that is important for engine analysis. It is further subjected to the motion of suitable boundary condition for the piston, cylinder, fluid and walls. The transport equations that were solved for the turbulent kinetic energy, k and its dissipation rate, ε are given by:

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_i}\left(\alpha_k \mu_{eff}\frac{\partial k}{\partial x_i}\right)+ G_k+G_b-\rho\varepsilon-Y_M \quad (1)$$

And:

$$\frac{\partial}{\partial t}(\rho \varepsilon)+\frac{\partial}{\partial x_i}(\rho \varepsilon u_i) = \frac{\partial}{\partial x_i}\left(\alpha_\varepsilon \mu_{eff}\frac{\partial \varepsilon}{\partial x_i}\right)+ C_{1\varepsilon}(G_k+C_{3\varepsilon}G_b)-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k}-R_\varepsilon \quad (2)$$

Where:

G_k = The generation of turbulence kinetic energy due to the mean velocity gradients
G_b = The generation of turbulence kinetic energy due to buoyancy
Y_M = The contribution of the fluctuating dilatation in compressible turbulence to the overall dissipation rate

The quantities α_k and α_ε are the inverse effective prandtl numbers for k and ε, respectively. $C_{1\varepsilon}$, $C_{2\varepsilon}$ and $C_{3\varepsilon}$ are constants.

MATERIALS AND METHODS

Discretization methods: According to Neshat and Saray (2014), discretization is definite as the process of transferring continuous models and functions into discrete counterparts. This process is often done to enable evaluation as well as implementation of discretization method. Fluent allows the user to choose the discretization scheme for the convection terms of each governing equation. When the pressure-based solver is used, all equations are by default, solved using the second-order upwind differencing scheme as the spatial discretization is used for the momentum, energy and turbulence equations (Wendy *et al*., 2008). Pressure-Implicit with Splitting of Operators (PISO) algorithm was used in the study (Coskun *et al*., 2014). The conservation equation is discretized with the segregated solver are used for the computations which employs a cell-centered finite volume approach on an arbitrary quadrilaterals and triangles mesh relating the arbitrary Lagrangian-Eulerian approach. The spray module which is used for early spray injection case is based on the discrete model of Dukowiez alongside the Lagrangian computation particles in the model display a spray of particles in a uniform motion. The mixture of spray and the fluid is analyzed by number of different models which include the methods of aerodynamic drag, dispersion, evaporation, secondary break-up. A standard Arrhenius-based combustion model with single step mechanism to was used as first step for simulation of HCCI engine combustion. Fluent uses a control volume-based system to reform the major equations to equations of algebra that are solved numerically (Hunicz, 2014). While the energy equation utilized boundary conditions for combustion analysis (Li *et al*., 2014).

Fuel injection timing: The engine was fitted with a direct-injection gasoline fuel system. This system used high-pressure fuel circuits operating at 10 MPa. In the experiments, multiple fuel injection was applied where the mass of fuel per engine cycle was divided into a maximum of 4 doses. Four characteristic Start-of-Injection (SOI) timings were selected based on the experimental data (Hunicz *et al*., 2015; Hunicz and Kordos, 2011). In this study, the mass of fuel injection was 18 mg and fuel injection strategies of 1/4, 1/4, 0, 1/2 are chosen. It means a quarter of the amount of fuel injected in the first stage and the second stage in addition to the injection half of fuel in the fourth stage of the engine cycle. The progress of the main event processes for different injection strategies at a constant intake pressure of 0.13 MPa absolute and a fixed fuel mass of 18 mg.

The ISO-octane (C_8H_{18}) reaction mechanism used as fuel to simulate the ignition and combustion chemistry of high-octane gasoline and used the generalized laminar finite rate model to analyze the iso-octane - air combustion system.

Spray breakup model: The spray simulation involves two-stage flow phenomena and requires the numerical result of conservation equation for the liquid as well as the gas stage simultaneously. The spray module was based on a statistical approach referred as the discrete droplet approach. The evaporation model was founded on the characteristics of one component fuel like ISO-octane. Fluent proposes two spray breakup models, the Taylor Analogy Breakup (TAB) and the wave model. TAB Model is a classic method for calculating droplet breakup, rely on Taylor's analogy between the distorting droplet and oscillating globules and the spring mass system while wave models are usually based on liquid jet stability, it depends on density-viscosity and velocity. The present research TAB Model was used with the value of breakup time constant, since, it was based on the analogy between the oscillating as well as distorting droplets are a result of the spring-mass system (O'Rourke and Amsden, 1987; Reitz, 1987; Tanner, 1997). The dynamic drag low was utilized for tracking the variations in the droplet shape (Liu *et al*., 1993).

Validation of the CFD model: Validation is important to ensure that the simulation results are in agreement with the experiment results. The validation of the CFD Model approach provides a vital insight into the investigation of internal combustion engines. A 3D Model of the engine was built and the outcomes are validated with experimental results to confirm the accuracy of the results obtained through the application of CFD.

The simulation was validated using a mass of 18 mg of injected fuel and (1/4, 1/4, 0, 1/2) as the injection strategies, fuel injection timing of -60, 20 and 300°CA, taken from experiment (Hunicz *et al*., 2015). The validation will investigate the pressures, temperatures and Heat Release Rate (HRR) that can be reached through simulation using the CFD Model. The CFD provides the opportunity to undertake a repetitive parameter analysis in the presence of clearly defined boundary conditions to investigate different configurations of desired parameters for combustion.

In-cylinder pressure: In-cylinder pressure is a very significant aspect in engine combustion diagnosis or analysis. It is imperative to note that in-cylinder pressure is very crucial for engine monitoring systems. The main purpose of the engine is to generate sufficient in-cylinder pressure which is converted to energy and used to do work. High in-cylinder pressure is desirable as compared to low pressure. Moreover, high in-cylinder pressure is associated with high engine performance while low in-cylinder pressure is related to poor engine performance. Consequently, the in-cylinder pressure is very imperative in determining and evaluating the performance of HCCI engines.

Figure 5 shows a plot of in-cylinder pressure against crank angle using the CFD Model and compared against experiment. When the contours of pressure at different crank angles are visualized, the simulated pressure was found to increase with an increased crank angle, even though the simulation plot result. Then, when the pressure reaches 357°CA, combustion begins which indicates the Start of Combustion (SOC) point. The simulated pressure increased to a maximum at 5.21 MPa then started to drop at 365°CA. The maximum pressure rise depends on upon the quantity of fuel injected, inlet pressure and temperature and other operating conditions. Note that simulated peak pressure is 5.21 MPa at 365°CA

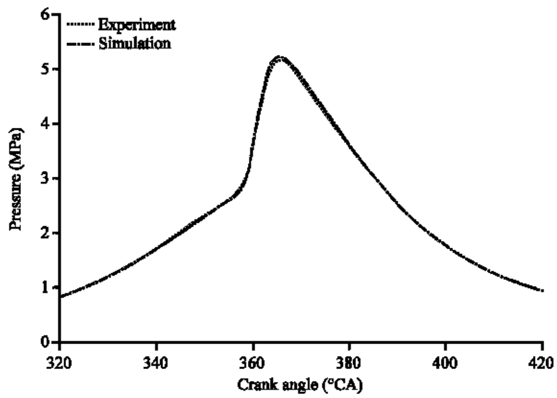


Fig. 5: Comparisons between simulation and experimental pressure results

and experimental peak pressure is 5.13 MPa at 365°CA. The result shows that the in-cylinder pressure is in good agreement with the experimental data where the plots of the simulation and experiment follow similar trends and slopes.

In-cylinder temperature: In-cylinder temperature is linearly correlated with pressure. During the simulation process, the start of combustion is at temperature of approximately 912°K at 357°CA. The simulation results showed that as the crank angle increases, the temperature increases but at a slower rate until 357°CA the temperature increases to a peak. After the peak temperature at 367°CA, any increase in the crank angle results in a small drop in the combustion temperature. Note that simulated peak temperature is 1740°K at 367°CA and experimental peak temperature is 1715°K at 367°CA. The higher temperatures indicate that the HCCI engines have a high combustion temperature inside the chamber to burn the fuel completely. The result of simulated in-cylinder temperature is compared against experiment and as shown in Fig. 6 where the CFD result is in good agreement with the experimental result.

Heat Release Rate (HRR): Heat release rate is the amount of energy released in the form of heat when fuel-air mixture is burnt within the combustion chamber. Heat release rate is important in determining the power output, temperature and in-cylinder pressure in the engine. Figure 7 shows the Heat Release Rate (HRR) graph plotted against crank angle. It can be observed, the HRR remained constant before the ignition and combustion stage of the engine. The heat release rate rapidly shoots to 357°CA when the fuel is ignited and combustion

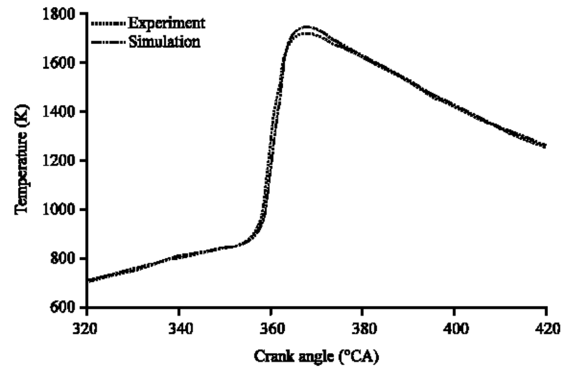


Fig. 6: Comparisons between simulation and experimental temperature results

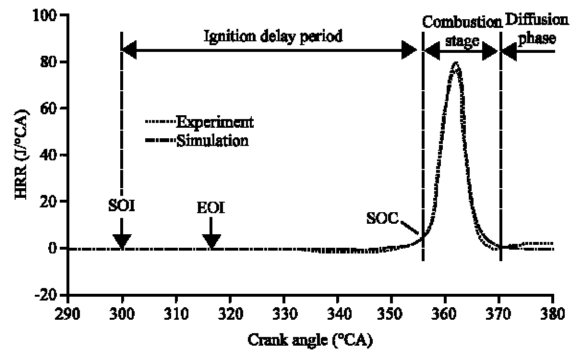


Fig. 7: Comparisons between simulation and experimental heat release rate results

occurs for both simulation and experimental then reaches a peak at 362°CA. After this point, the piston moves down the cylinder due to the pressure generated by the combustion, the heat release rate rapidly drops. The drop in heat release rate is maintained at 370°CA. These results showed that the combustion requires chemical energy to reach its peak. The first stage due to premixed combustion strongly depends on the amount of fuel that prepared for combustion. The second stage due to diffusion combustion is controlled by the fuel-air mixing rate (Sharma *et al*., 2016). Note that the simulation results are higher than the experimental results at peak level. The peak simulation heatrelease rate is 79.47 J/°CA whereas experimental peak heat-release rate is 75.69 J/°CA.

From the combustion analysis, it was clear that the difference between simulation and experimental in heat release depends on fuel characteristics as the effect of chemical fuel components. The fuel was simulated such as iso-octane while in experimental was gasoline, ISO-octane was found to have an only small difference in peak heat release with gasoline (Canakci, 2012). This demonstrates that the CFD Model is capable of accurately predicting the combustion behaviour in the combustion chamber.

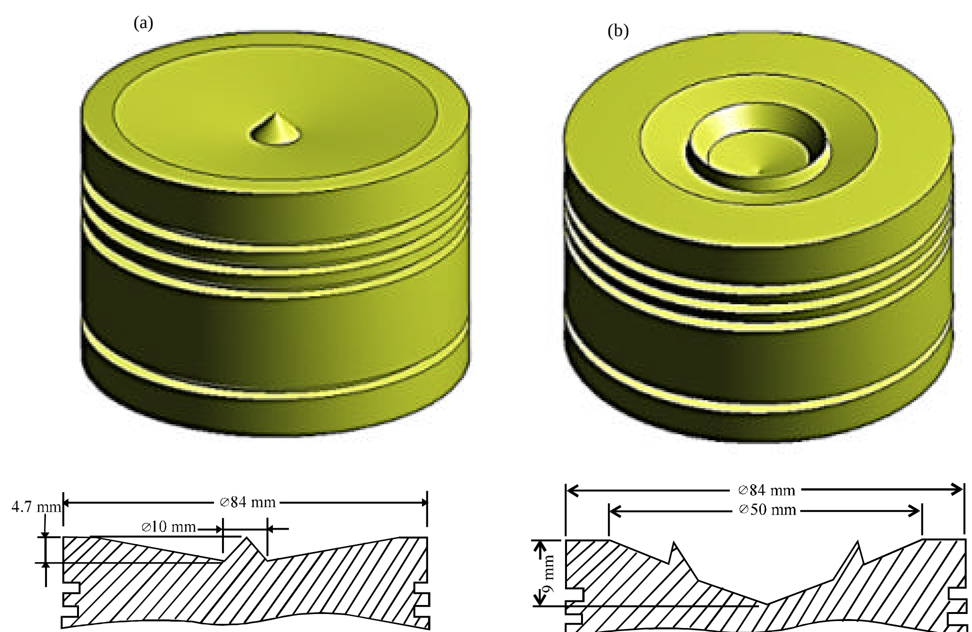
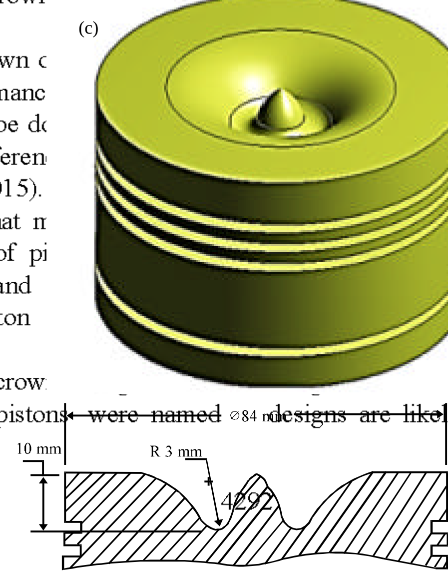


Fig. 8: Comparison of three piston crowr



Piston crown designs: Pistons crown c important part of the engine performanc be either, convex or concave may be dc head in order to remove interferen (Harshavardhan and Mallikarjuna, 2015). engine is a cylindrical member that m cylindrical liner. The movement of pi homogeneous mixture of fuel and which is compressed as the piston (Genzale and Wickman, 2006).

In this study, three piston crow investigated, for simplicity, the pistons were named

nodels are developed in ANSYS for The mesh diagram of each of the d in the sections follow. The three op contours with similar compression Fig. 8. Numerous studies have proved surface will be beneficial for mixture nd Gupta, 2015; Harshavardhan and Indrodia *et al*., 2014; Gugulothu and al., 2014; Raj *et al*., 2013; Wu *et al*., 015a). Therefore, different shapes are p surfaces of three pistons. These designs are likely to overcome many issues associated with the performance of HCCI engine. For piston A when its head is concave shaped with a small conical with a height of 4.7 mm and base of diameter 10 mm. The conical at the center of the piston crown encourages swirl motion of fuel and air mixture. For piston B when its head is a stepped conical crown. The outer cone ringed two smaller cones at the center, the inner two separated from the outer by a wall of different inner and outer slant angles. For piston C when its head is similar to Mexican hat.

RESULTS AND DISCUSSION

Three piston geometries of A, B and C were designed and considered for investigation because some piston designs were used in previous studies (Benajes *et al*., 2015) while others are designed to get higher performance. All the three models were developed in CATIA Software. All the three types of top piston contours had a similar compression ratio of 11.7. Different piston crown designs have a different impact in engines. The piston motion is not only controlled by the crank shaft but also determined by the interaction of forces in the combustion chamber. The role of the piston in engines is to increase the generation of power. The piston crown design allows the air to experience tumbling and swirling effects inside the combustion chamber, thus, helps the fuel to mix homogeneously with air. The use of improved piston design in HCCI engines is expected to improve the combustion rate of the engine.

In-cylinder pressure: Figure 9 shows the effects of different piston designs on the in-cylinder pressure. The experiment plot is the graph obtained from the piston used in the experiment. The simulated pistons showed that all pistons used in the investigations reached a peak pressure above the experiment which can be used to obtain higher work by producing higher peak torque. Therefore, high in-cylinder pressures are desirable.

All piston designs had their pressures increasing at a constant rate. Each piston had different peak pressure. Significant differences appear during the start of combustion process after 357°CA when the pressure raised to the peak. Piston A has the highest peak pressure followed by piston B. Piston C has the lowest peak pressure as shown in Fig. 9. Due to the HRR and TKE for piston A, the in-cylinder is higher than piston C due to higher turbulence kinetic energy which produces better between air and fuel. The result shows that a high HRR causes a rapid rise in the combustion pressure.

Using pistons of different crowns implies that the combustion rate will vary accordingly, thus, affecting the

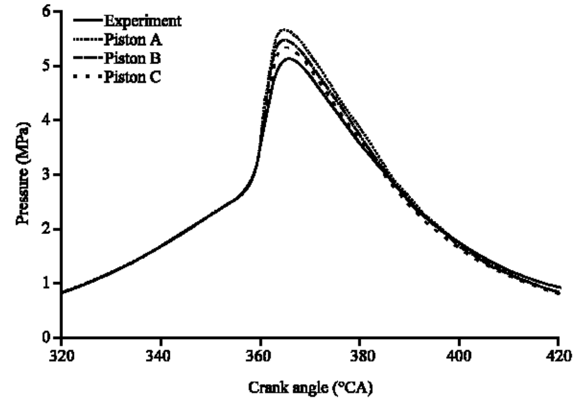


Fig. 9: Comparison between experiment and different piston top designs in-cylinder pressure

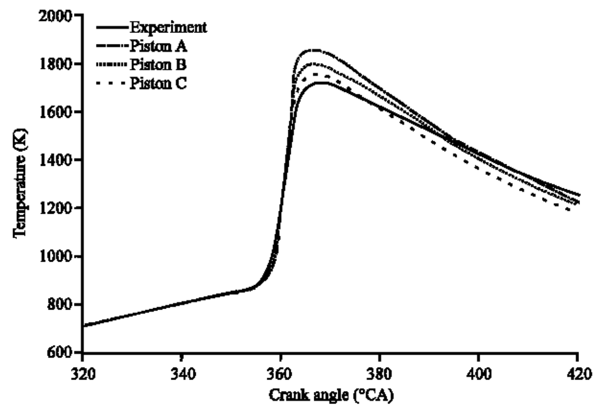


Fig. 10: Comparisons between experiment and different piston in-cylinder temperature

efficiency of HCCI engine. In-cylinder pressure affects the fuel combustion rate and works done which the piston moves up and down the cylinder thus affecting the amount of torque transferred to the shaft by the piston. Therefore, the CFD Model can be used to determine the peak pressure at which combustion of fuel reaches its optimum.

In-cylinder temperature: Different pistons crowns lead to a difference in temperature values. In Fig. 10 shows the combustion temperature in both simulation pistons and the experiment. Pistons start at higher temperatures, thus, increasing the fuel combustion rate and increases with an increase in the crank angle. The temperature moves upwards with an increase in crank angle until about 357°CA when combustion occurs and the temperature increased to peak. It noted that there existed a sudden change of the slope of the curve according to the type of piston.

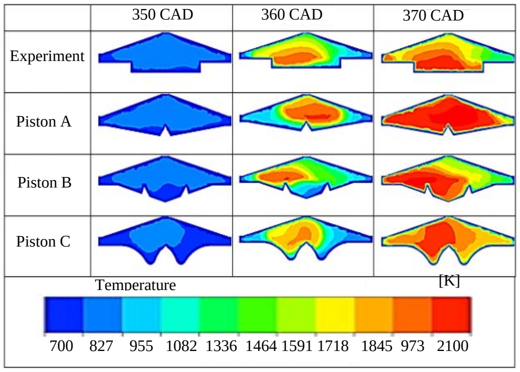


Fig. 11: Contour views of the temperature distributions at 350, 360 and 370°CA for experiment and different piston top designs

Each piston had different peak temperature is correlated linearly with pressure as combustion increases, a higher peak temperature indicates better performance because the piston can perform well at high temperatures. Piston A was the highest in terms of peak in-cylinder temperature, followed closely by B and C. While all piston crown designs exhibit the almost the same trend and level of temperature increase with the increase in crank angle, piston A exhibited the highest peak temperature. Additionally, the crown design for piston A allows for maximum air circulation within the cylinder, thus has high turbulence kinetic energy. High turbulence kinetic energy directly affects air-fuel mixing thus improves the combustion process.

Figure 11 compares the contour for different pistons of temperature and its effect in combustion process during the compression and expansion strokes at 350, 360 and 370°CA. It is clear that the high temperature region in piston A has better combustion process that is more homogeneous compared to other pistons and generated more heat and allows high temperature gas to reach the liner and oxidize much of the fuel. Piston A indicates that combustion starts at high temperatures and continues to increase gradually to the peak then drops as the combustion continues. The entire combustion process is dominated by high temperatures. In all cases, the pistons show that combustion begins at a higher temperature but reduces with an increase in crank angle. Therefore, piston A is more efficient than the other pistons.

Heat release rate: The heat release controls the ignition delay, the burn of the premix and the diffusion burn. In Fig. 12, the initial heat release at the beginning of the engine cycle was stable until a crank angle of 357°CA then rose to a peak level at 362°CA. Piston A produces the highest heat release rate, followed by piston B

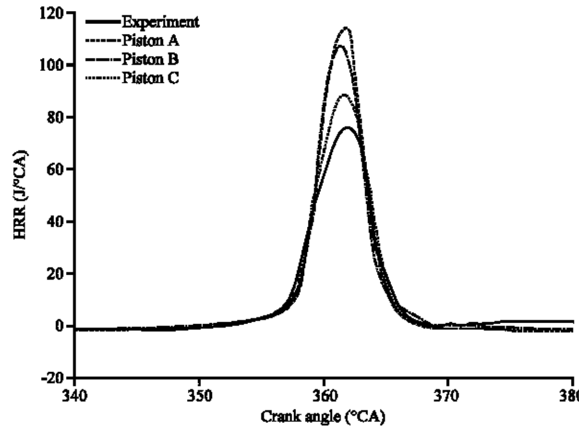


Fig. 12: Comparisons between experiment and different piston crown designs in-cylinder heat release rate

and C. The heat release rate affects the pressure and temperature in the cylinder. The shape of the piston crown highly affects both pressure and temperature as indicated earlier. Piston A has a higher heat release rate than the other pistons, since, it exhibits higher pressure and temperature as compared to the other pistons. The mixing process between air and fuel plays a significant role in determining tumble and swirls motions, created a turbulence within the cylinder that enhanced the air-fuel mixing and the fuel vaporization (Catapano *et al.*, 2016). In this case, it is worth noting that in cylinder pressure and temperature highly influence the HRR and thus a higher HRR means improved combustion efficiency (Caton, 2015). After reaching the peak heat release rate, the heat release rate dropped to about 0 beyond a crank angle of 370°CA.

As can be seen, at piston A has higher peak HRR (112.48 J/°CA) appears around 362°CA. The effect of piston A, accelerating heat release is better than all the pistons, leading to an increase in the pressure. A theory explaining this phenomenon is that the heat losses are higher in piston A during the stages of combustion since the TKE is high. This means that there is high energy left in the turbulence and flow when the compression from combustion pushes the unburned gases into the squish. Squish is the name given to the radially internal or transverse air movement that happens toward the end of the compression stroke when a part of crown piston face and cylinder head approach each other nearly. Therefore, HRR inside the combustion chamber plays a significant role in the HCCI combustion mode.

Turbulent kinetic energy: From the viewpoint of the fundamental turbulence physics, the major parameter

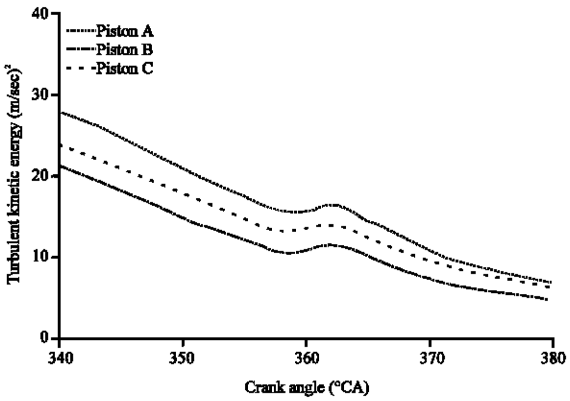


Fig. 13: Turbulent kinetic energy at different piston crown designs

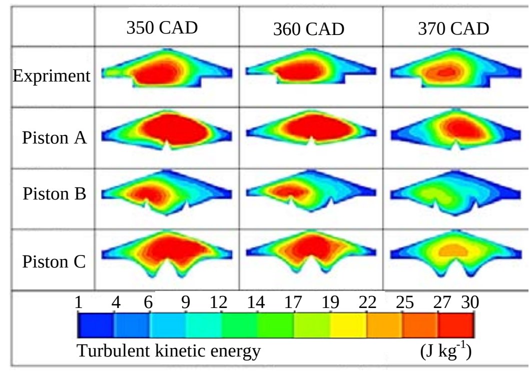


Fig. 14: Turbulent kinetic energy contours at 350, 360 and 370°CA for different piston crown designs

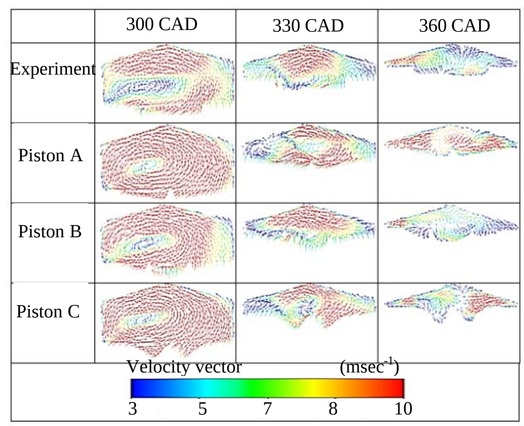


Fig. 15: Velocity vectors at 300, 330 and 360°CA for experiment and different piston crown designs

required to characterize the turbulent flow characteristic is the Turbulence Kinetic Energy (TKE). The turbulent kinetic energy is related to the turbulence intensity which is generally generated by the angular momentum of the in-cylinder flow around each of the three orthogonal axes. It depends on the piston movement in cylinders with different engine speeds and on the reaction of gases inside the combustion chamber (Wendy *et al*., 2008). It is a fact that fluid flow can either be laminar or turbulent. Most of the flows experienced in the HCCI are turbulent (Ghorbanpour and Rasekhi, 2013). Figure 13 shows turbulent kinetic energy level during the combustion of fuel into the HCCI engine. The calculation started at a crank angle of 0°CA and ended at 420°CA but the important region in comparing pistons are 340-380°CA. In this region, the turbulent kinetic energy and dissipation rate start at a higher level andthen decrease to a lower level when combustion starts at 357°CA. Here, it rises slightly with an increase in the crank angle untilthe peak turbulent kinetic energy and dissipation rate at 362°CA. Then, as the pistons move down, the turbulent kinetic energy and dissipation rate fall again as the piston descends on the power stroke with an increase in the crank angle. Past this crank angle, the turbulent kinetic energy and dissipation rate drop to a minimum until the end of the combustion of fuel.

The results are shown in Fig. 14, piston A has a higher turbulence level in the cylinder as compared to the other piston crown designs because of the higher heat release which improves the mixing rate and wall heat transfer. Turbulent kinetic energy causes faster breakup and evaporation of liquid droplets resulting in improved air-fuel mixing and better combustion and strong enough the ignition will be advanced (Wendy *et al*., 2008). This increases the kinetic energy in the piston and translates to increased temperature.

Piston A reaches the peak turbulent kinetic energy fast and remains higher than the other pistons and experiment this may be due to concave shape with a small upward-pointing conical at the center which gives better guidance for the mixture entering the cylinder. It was noted that the configuration of the piston affects the turbulent of the fuel inside the cylinder. These results showed that increasing the turbulent in an HCCI engine, piston A would offer a more favorable option as it increases the power output. At piston A, turbulence in the center region can be seen and the turbulence level is much higher compared to piston B at 350°CA. It leads to a large burnt fuel mass fraction.

Velocity vector analysis: The geometry of combustion chamber could have a significant impact on the formation of squish and consequently, affect the evaporation and mixing processes. Figure 15 shows the velocity vector field in cylinder near TDC at 300, 330 and 360°CA. Both

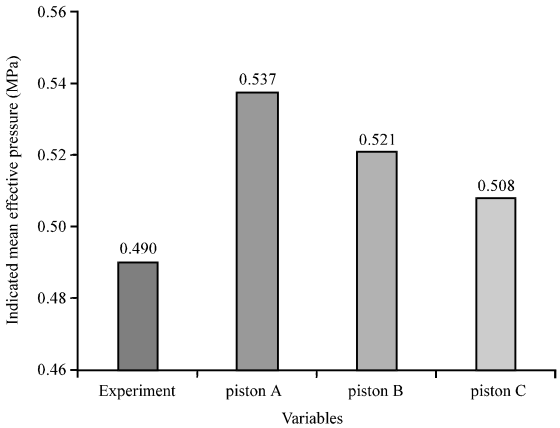


Fig. 16: IMEP at different piston crown designs

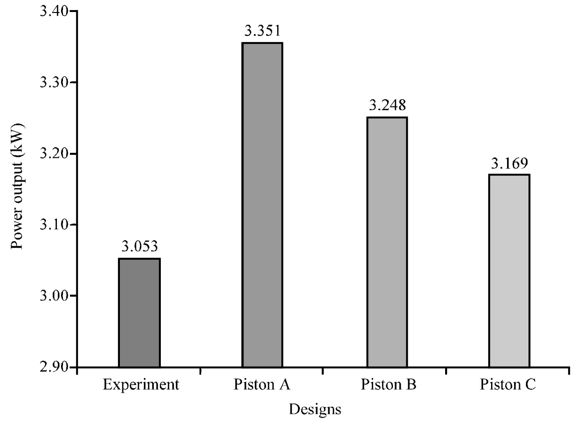


Fig. 17: Power output at different piston crown designs

magnitude and direction of the velocity vector could affect the evaporation and mixing processes. In addition, the direction of the velocity is related to the shape of the piston geometry which can further enhance the air-fuel mixing process, subsequently improving the combustion process. Figure 15 demonstrates the in-cylinder flow comparison between the experimental piston and the other piston crown shape's velocity plots for all three piston configurations and show similar patterns in air movement and the dominant clockwise vortex. This is because pistons A and C both have a vortex generated into the centre combustion chamber compared to other pistons which would greatly increase the speed of air entering the top piston as a result of the angular velocity and the formation of a strong squish. Moreover, an opposite air flow is formed against the fuel spray in pistons A and C design which can further enhance the air-fuel mixing process, later to improve the combustion process.

Indicated Mean Effective Pressure (IMEP): Engine load can be expressed as torque or Indicated Mean Effective Pressure (IMEP). For comparing engines of different sizes, IMEP is preferred. The IMEP increases with advanced combustion phasing, mainly due to improved combustion efficiency. This may be attributed to better combustion due to better air-fuel mixing as a result of improved swirl ratio and Turbulent Kinetic Energy (TKE) as determined in a simulation study (Fig. 16). Piston A has the highest IMEP result followed by B and C.

Power output: Figure 17 shows the highest and lowest magnitudes of power output for the pistons. Piston A has the highest power output due to high peak pressure towards the end of combustion leading to adequate diffusion combustion. On another hand, the lowest magnitude of power output is encountered in piston C compared to other pistons. Piston A has a higher heat release rate, thus produces more power and IMEP than piston B and C.

In this study, simulations were carried out for a singlecylinder, 4-stroke engine, rotational speed of 1500 rev/min that run in HCCI mode by using iso-octane as fuel. The bowl piston combustion geometry was used for model construction.

Validation of results was carried out by using in-cylinder pressure, in-cylinder temperature and heat release rate measurements based on experimental results of Hunicz *et al.* (2015). The results showed good agreement between numerical and experimental.

The results reported in this study illustrate that the numerical simulation can be one of the most powerful and beneficial tool to compute the essential features of combustion parameters for ICE development, optimization and performance analysis. The investigations also show that shapes of crown piston have a significant influence on various engine performance factors such as in-cylinder temperature, in-pressure, turbulent dissipation rate, turbulent kinetic energy, power output and Indicated Mean Effective Pressure (IMEP) at different crank angles. Thus, CFD Model is best suited to study the effects of different piston crown designs in an HCCI engine fueled with ISO-octane.

After the analysis the following results can be summarized as follows: the in-cylinder peak pressure of the experiment was found to be 5.13 MPa in comparison with piston A of 5.64 MPa. Thus, an improvement of 10% of pressure is achieved. The generated in-cylinder pressure is converted to torque and used to do work which means that a high in-cylinder pressure is directly proportional to the increased engine performance.

The Indicated Mean Effective Pressure (IMEP) is a way of expressing engine load and thus, the IMEP is the

Table 3: The individual and total weighted scores of pistons A, B and C against various engine performance determining factors

Engine performance factors	Piston A	Piston B	Piston C
In-cylinder temperature	3	2	1
In-cylinder pressure	3	2	1
Heat Release Rate (HRR)	3	2	1
Turbulence kinetic energy	3	1	2
Velocity vector analysis	3	1	2
Indicated mean effective pressure	3	2	1
Power output	3	2	1
Total	21	12	9

average pressure that acts on a piston in the different parts of its cycle. In piston A there is an increase of 9.6% in IMEP compared with the experiment result. A higher IMEP value mean better fuel combustion due to improved air-fuel mixing, high swirl, high TKE and thus in overall, improved combustion efficiency and engine performance.

The power output magnitudes for the different pistons were weighed. In piston A there is an increase of 9.76% in power output compared with the experiment result. Piston A had the highest power output followed by B and C. In summary, since, three pistons were used in the simulation, the performance of each piston are weighted on a rating of 1-3 where a weight measure of 3 will signify best performance result while a rating of 1 will indicate lowest performance in each factor as shown in Table 3.

CONCLUSION

Based on the summation of the weighted factors according to the performance of each piston in each metric, piston A scores highest on overall meaning that this piston's design is the most appropriate to use for improving the performance of HCCI engines fueled with gasoline. In this case, piston A is evaluated as the best piston to be used in an HCCI engine fuelled with gasoline. Engine designers and automotive designers might have to make some trade-offs when it comes to designing high-performance.

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