A piston geometry and nozzle spray angle investigation in a DI diesel engine by quantifying the air-fuel mixture

Pavlos Dimitriou¹, Weiji Wang¹ and Zhijun Peng²
¹School of Engineering & Informatics, University of Sussex, Brighton, BN1 9OT, UK
²School of Engineering & Design, University of Hertfordshire, Hatfield, AL10 9AB, UK

(Submission date: May 23, 2014; Revised Submission date: August 11, 2014; Accepted date: August 16, 2014)

ABSTRACT
Low temperature diesel combustion has been widely investigated over the last few years for reducing in-cylinder emissions of Direct Injection (DI) diesel engines without sacrificing efficiency and fuel consumption. The spatial distribution of the fuel within the combustion chamber and the air-fuel mixing quality are the key factors affecting temperature generation within the cylinder. Avoiding fuel rich areas within the cylinder can significantly reduce the local high temperatures resulting in low NOx formation. This paper investigates the effects of the combustion chamber geometry and spray angle on the air-fuel mixing and emissions formation of a DI diesel engine. A new quantitative factor measuring the air-fuel mixing quality has been adopted in order to analyze and compare air-fuel mixing quality for different piston geometries. The results have shown that pistons with a narrow entrance and a deep combustion re-entrant chamber benefit from increased air-fuel mixtures due to the significantly higher swirl generated within the cylinder. However, the improved air-fuel mixing does not consequently lead to a reduced NOx generation, which is highly affected by the combustion efficiency of the engine.

1. INTRODUCTION
The phenomena in DI internal combustion engines are very complicated with many chemical and physical processes occurring and interacting with each other. The complex geometries of modern Internal Combustion (IC) piston geometries aggravate the analysis of these processes. In direct injection diesel engines, piston geometry is usually characterized by a re-entrant bowl piston with a protuberance in the middle to improve air-fuel mixing. The air-fuel mixing quality is of great importance to the combustion process, which as a result affects the engine’s performance and emissions formation.

*Corresponding author email: P. Dimitriou@sussex.ac.uk
Extensive research has been conducted into the effects of piston geometry characteristics on the air-fuel mixing and combustion process. Jaichandar and Annamalai [1] investigated the effect of injection timing and bowl geometry on the combustion and performance characteristics of a bio-diesel fueled diesel engine. The authors introduced three different bowl geometry configurations, namely Hemispherical Combustion Chamber (HCC), Toroidal Combustion Chamber (TCC), and Shallow depth Combustion Chamber (SCC). They found that there was a higher brake thermal efficiency and a significant decrease of carbon monoxide (CO), unburned hydrocarbons (UHC) and particulate matter (PM) emissions for the TCC piston geometry compared to the HCC and SCC, although a slight increase in the NOx formation was observed.

Li et al. [2] used a similar piston configuration to simulate the effects of piston bowl geometry on combustion and emissions of a diesel engine under medium load conditions. They observed that a narrow entrance of the re-entrant combustion bowl could generate a strong squish, hence enhancing the air-fuel mixing. They concluded that the performance of the pistons varies with the engine speed, and none of the piston geometries had optimum performance for all the engine speed loads tested. The authors used velocity vector fields and temperature contours to present the effects of piston geometry on the combustion process.

Harshavardhan et al. [3] used STAR-CD Es-ice code to carry out a CFD analysis on the in-cylinder fluid flow and air-fuel interaction in a Direct Injection Spark Ignition (DISI) engine with three different piston geometries compared to a flat piston. They found that air-fuel mixture is distributed over all the combustion space in all the piston configurations. The equivalence ratio was slightly rich at the center of the geometries. The authors presented their results using velocity, turbulence kinetic energy (TKE), and equivalence ratio contours. However, the effect of piston geometry variation on the equivalence ratio and hence the air-fuel mixing, could not be clearly observed by the equivalence ratio contours.

Payri et al. [4] performed a CFD analysis to investigate the flow characteristics inside a diesel engine’s cylinder equipped with three different piston configurations. The results showed that piston geometry had little influence on the in-cylinder flow during the intake stroke and the first part of the compression stroke. However, the flow varies within the cylinder for the three piston configurations near Top Dead Center (TDC) and in the early stage of the expansion stroke. They concluded that pistons with a larger bowl diameter generate a significantly smaller swirl around the TDC. On the other hand, the highest swirl occurred in the piston with the smallest entry radius.

More than a decade earlier, Heywood [5] concluded that for a fixed compression ratio, the swirl at the TDC was enhanced when the bowl diameter of the piston was reduced. He found that the higher levels of swirl led to less smoke, but higher NOx and HC emissions.

Tsao and Dong [6] used the KIVA-II code to analyze the influence of the piston’s bowl depth to the in-cylinder air flow. They found that the fuel velocity in the center of the chamber of the pistons with increased bowl depth was higher due to the higher inertia. In addition, the higher clearance improved the squish motion and the fuel distribution within the cylinder.
In 1995 Zhang et al. [7] noted the importance of adopting a re-entrant chamber in diesel engines. They found that within a piston with a re-entrant chamber, the combustion was enhanced during the expansion stroke, preventing the diffusion of the flame in the squish region and giving lower soot levels. They also stressed that the mean combustion velocity increased with the combustion chamber radius and was lower in the case of flat-bottom bowls.

In recent years, several engine modeling research groups [8, 9, 10, 11, 12] have applied genetic algorithms to optimize complex engine design parameters. By coupling genetic algorithms to CFD codes, a large number of engine operating parameters can be simultaneously optimized at a relatively low computational cost. De Risi et al. [8] mentioned in their discussion the importance of including several modes in the piston’s geometry optimization process since the performance of each chamber is strongly affected by the operating conditions. They found that NOx reduction can be achieved by adopting a narrow and deep combustion chamber with a shallow re-entrance and a low protuberance on the cylinder axis. Additionally, they focused on the importance of the fuel spray orientation toward the bowl entrance for optimized air-fuel mixing.

Using genetic algorithms, Genzale et al. [9] detected a general trend in bowl shapes where a deep, small diameter bowl was preferred for NOx reduction and a shallow, wider bowl was preferred for reduced fuel consumption. They also found that increasing the air swirl ratio entering into the cylinder resulted in increased NOx and decreased soot and fuel consumption.

Song et al. [13] performed three-dimensional flow calculations of the in-cylinder flow for a DI diesel engine with different combustion chambers. Their investigation showed that swirl became more homogenous as the cylinder moved upwards during the early phase of compression stroke. The piston geometry had negligible effect during the intake stroke and the early phase of compression stroke.

Siewert [14] explored the effects of varying the nozzle spray angle and rail pressure on emissions and thermal efficiency of a diesel engine. He found that increased rail pressure improves atomization and vaporization of the spray as evidenced by significant reductions in smoke. However, he concluded that for conditions where the spray misses the bowl due to a wide spray angle (158°), increased rail pressure exacerbates the deterioration of emissions and thermal efficiency due to increased spray penetration outside of the bowl. Mobasheri and Peng [15] performed a computational investigation into the effects of the spray angle on a heavy-duty diesel engine. They concluded that a 105° spray cone angle along with an optimized split injection strategy could significantly reduce NOx and soot emissions without much penalty in fuel consumption, as compared to wider spray angles.

The results in literature confirm that it is difficult to adopt an optimum piston geometry that is suitable for all operating conditions. The effect of the piston chamber’s geometry on the engine performance is very complex due to its dependence on the engine’s specifications, flow field, and the air-spray interaction. The piston bowl geometry has a close relationship with the combustion and emission formation processes of a DI diesel engine, as it can strongly affect the air-fuel mixing quality before the start of and during the combustion.
In this paper, the effects of piston geometry and spray angle on the air-fuel mixing quality, emissions formation, and engine performance are investigated using CFD simulations. The results are analyzed with the aid of the Homogeneity Factor in order to indicate the importance of the parameters of the air-fuel mixing quality and combustion process. The purpose of the research is to indicate how the air-fuel mixing phenomena within the cylinder can influence the combustion characteristics and the emission formation. The investigation is performed under various engine loads and speeds with variable injection timings. The paper is divided into the following categories; first the CFD model is validated compared to experimental engine test results. Following this, simulations for nine different piston geometries under various engine conditions are performed. Finally, the piston geometry with relatively low emissions and high performance is further tested to investigate the effects of the injection spray angle with respect to the injection timing on the air-fuel mixing quality within the combustion chamber.

2. NUMERICAL METHOD
2.1. Sub-models
The CFD investigation was performed using AVL FIRE CFD code for Diesel combustion. The submodels employed in the code are presented below. The selection of suitable submodels for DI diesel engine combustion was based on a previous researchers’ work.

The WAVE model [16] was selected for the primary and secondary atomization of the fuel spray. This model is suitable for high pressure fuel injections. The WAVE as well as the Taylor Analogy Break-up (TAB) [17] models do not distinguish between the primary and secondary fuel atomization, facilitating easier setting of the model parameters. However, the TAB model is not suitable for high pressure fuel sprays, therefore was not used in the study. Some other models that could be used for predicting fuel spray atomization, are the ETAB (Enhanced Taylor Analogy Break-up) [18], FIPA (Fractionnement Induit Par Acceleration) [19], or KH-RT (Kelvin Helmholtz-Rayleigh Taylor) [20]. However, these models treat and simulate the primary and secondary regions separately. This fact could incommode the set up of the correct values for the additional set of parameters. The WAVE model assumes that the droplet size and the break up time are related to the fastest-growing Kelvin-Helmholtz instability [21]. The details of the newly-formed droplets are predicted using the wavelength and growth rate of this instability. The parameters of the model have been tuned to match the experimental data.

The Dukowicz evaporation model [22] was selected for the heat-up and evaporation prediction of the diesel fuel droplets. The Dukowicz model used the heat balance to determine the rate of droplet temperature change. The temperature change indicates that the heat transferred from the gas to the droplet for its vaporization. The model tunable constants have been adjusted to match the experimental data.

The k-ζ-f model developed by Hanjalic, Popovac and Hadziabdic (2004) [23] was preferred over the standard (much simpler) two equation k-ε [24] and RNG k-ε [25] models for the evaluation of the turbulence effect in the combustion chamber.
The robustness, accuracy, and suitability of the model for computations involving grids with moving boundaries and highly compressed flows make it widely popular in IC flows. ECFM-3Z (Extended Coherent Flame Model–3 Zones) model [26] was applied for the combustion model of the simulations. The ECFM-3Z separates a computational cell in three zones in order to enable specific treatment for air fuel mixing, auto ignition, combustion, and pollution formation processes. The three different regime’s computation aids in a deeper understanding of the turbulence flow and provides data such as fuel mixture fraction distribution and fuel evaporation rate that is inaccessible with experimental devices.

Finally, Zeldovich [27] and the Kennedy, Hiroyasu, and Magnussen mechanism [28, 29] were implemented in the software for NOx and soot formation respectively. The soot formation implemented is based upon a combination of suitable extended and adapted joint chemical/physical rate expressions for the representation of the processes of particle nucleation, surface growth, and oxidation.

### 2.2. Engine specifications

A light duty single cylinder diesel engine with a compression ratio of 18.3:1 and a swept volume of 499cm³ was used in this study. The engine was fit with a six-hole injector centrally placed to spray the fuel in the combustion chamber. The specifications for the engine and injection system are listed in Table 2 and Table 3.

#### Table 1: Computational submodels

| Break-up model     | WAVE [16]         |
|--------------------|-------------------|
| Evaporation        | Dukowicz [22]     |
| Turbulent model    | $k-	ilde{\omega}$-f model [23] |
| Combustion         | ECFM-3Z model [26]|
| NOx mechanism      | Extended Zeldovich [27] |
| Soot model         | Kennedy, Hiroyasu and Magnussen [28,29] |

The robustness, accuracy, and suitability of the model for computations involving grids with moving boundaries and highly compressed flows make it widely popular in IC flows.

ECFM-3Z (Extended Coherent Flame Model–3 Zones) model [26] was applied for the combustion model of the simulations. The ECFM-3Z separates a computational cell in three zones in order to enable specific treatment for air fuel mixing, auto ignition, combustion, and pollution formation processes. The three different regime’s computation aids in a deeper understanding of the turbulence flow and provides data such as fuel mixture fraction distribution and fuel evaporation rate that is inaccessible with experimental devices.

Finally, Zeldovich [27] and the Kennedy, Hiroyasu, and Magnussen mechanism [28, 29] were implemented in the software for NOx and soot formation respectively. The soot formation implemented is based upon a combination of suitable extended and adapted joint chemical/physical rate expressions for the representation of the processes of particle nucleation, surface growth, and oxidation.

#### Table 2: Engine specifications.

| Specification          | Value                      |
|------------------------|----------------------------|
| Displaced volume       | 499 cc                     |
| Stroke                 | 86 mm                      |
| Bore                   | 86 mm                      |
| Connecting rod         | 143.5 mm                   |
| Compression ratio      | 18.3:1                     |
| Number of valves       | 4                          |
| Inlet Valve Close      | 64° ABDC                   |
| Exhaust Valve Open     | 69° BBDC                   |
| Engine speed and (load)| 1,200 rpm (80%), 1,600 rpm (60%), 2,000 rpm (40%) |
| Piston shape           | Mexican hat style          |
2.3. Computational grid

The piston and the injector geometry parameters have been set in the software using the 2D Sketcher tool. The computational grid was generated and a mesh independence test was performed for the model. The final grid independent model for the baseline piston geometry, shown in Figure 1, consists of 42,052 and 72,052 hexahedral cells at TDC and Bottom Dead Center (BDC) respectively.

The computational grids for the eight developed piston geometries tested in this research work consist of a very close number of hexahedral cells as for the baseline piston geometry.

2.4. Piston geometry

The baseline piston geometry was modified in the direction of finding the optimum piston geometry aiding to an improved air-fuel mixing. The compression ratio of the engine was locked at 18.3:1 for all piston models in order to make sure that results are only affected by the geometry and not the engine conditions.

The outer and inner bowl diameter (Da, Di) along with the re-entrant bowl radius (R4) and the bowl depth (T) were the four main parameters modified and tested for their contribution to the air-fuel mixing behavior within the cylinder. However, it is

| Injection pressure  | 1600 bar |
|---------------------|----------|
| Number of nozzle holes | 6       |
| Nozzle hole diameter  | 0.169 mm |
| Fuel type            | 49.1 CN diesel |

**Figure 1:** Computational grids at TDC.
very difficult to change one variable without affecting other dependent variables. Therefore, some of the changes have occurred in the rest of the piston variables are as shown in Table 4 and Figure 3. The nine pistons geometry configuration tested are shown in the Figure 3.

### 2.5. Test conditions

According to the results in literature, piston geometry has little influence on the in-cylinder flow during the intake stroke and the first part of the compression stroke. Therefore, the calculation starts at the inlet valve closure (IVC) and ends just before the
exhaust valve opening (EVO) for time reduction reasons. The simulation is carried out on a 60° sector for reduced calculation time due to the symmetric location of the six-hole injector at the center of the combustion chamber. The following air and fuel conditions were applied on the tests.

The investigation was divided into two main categories. The first part involves the study of piston geometry influence on air-fuel mixing by the simulation of nine different piston geometries, while the second part of the investigation is concentrated on the effects of the injection spray angle.

3. PARAMETER DEFINITION
In this paper, the mixing quality parameter used is the Homogeneity Factor (HF), which was originally developed by Peng and Liu [30]. The factor has been revised and further developed for more accurate results. In the formula, it needs at first to find

Figure 3: Pistons geometry configuration.
the fuel difference in a calculated cell (e.g., Cell i), compared to the average equivalence ratio:

\[
\frac{\Phi_i}{AFR_{st} + \Phi_i} \delta m_i - \frac{\Phi_0}{AFR_{st} + \Phi_0} \delta m_i = \frac{(\Phi_i - \Phi_0) AFR_{st}}{(AFR_{st} + \Phi_i)(AFR_{st} + \Phi_0)} \delta m_i
\]

where \( AFR_{st} \) is the stoichiometric air-fuel ratio, \( \Phi_i \) is the equivalence ratio in cell \( i \), \( \Phi_0 \) is the average equivalence ratio, and \( \delta m_i \) is the mass of the mixture in computational cell \( i \).

The total fuel amount in the cylinder is,

\[
\frac{\Phi_0}{AFR_{st} + \Phi_0} M
\]

where \( M \) is the total mass of the mixture.

Then, the parameter, Heterogeneity Factor (HeterF), can be expressed as,

\[
HeterF(\theta) = \sum_{i=1}^{N_{c}} \sqrt{\frac{(\Phi_i \cdot \Phi_0)^2}{2 \Phi_0 M \left( 1 + \frac{\Phi_i}{AFR_{st}} \right)}} \delta m_i
\]

As the increased fuel amount in a cell actually comes from the decrease of the fuel amount in other cells, half of the standard deviation is used in the definition to reflect the non-uniformity more accurately.

Based on HeterF (eq.3), the homogeneity factor (HF) can be derived due to having a quantitative demonstration of the mixing quality.

\[
HF(\theta) = (1 - HeterF(\theta))\%
\]

The HF can measure the percentage air-fuel mixing quality. In an ideal world, a value of 100% of homogeneity within the cylinder would represent a perfect air-fuel mixing condition.

---

**Table 5: Initial air & fuel conditions.**

| Parameter               | Value                  |
|-------------------------|------------------------|
| Intake air temperature  | 380 K                  |
| Intake air pressure     | 1 bar                  |
| Fuel temperature        | 350 K                  |
| Fuel injected           | 5.82 – 19.05 mg/cycle  |
| Injection duration      | 6.04˚ to 11.85˚CA      |
| Start of Injection      | 15˚ – 5˚ CA BTDC       |
4. MODEL VALIDATION

A validation has been performed for the baseline piston geometry model using experimental data conducted on the single cylinder research engine with the specifications as listed in Table 2 and Table 3. The in-cylinder pressure of the engine was measured using a Kistler 6056 in-cylinder pressure sensor with a 0–250 bar range. The heat release rate (HRR) was calculated on the basis of the in-cylinder pressure and the in-cylinder volume. A Testo 350 XL portable emission analyzer was used for measuring emissions formation.

The comparison between the predicted and measured in-cylinder pressure and HRR is shown in Figure 4. The result is based on the assumption of a uniform wall temperature of 470 K for the cylinder wall and 570 K for the cylinder head and the piston top.

The CFD simulation trend for the in-cylinder pressure seems to be in reasonable agreement with the experimental measured values. There is only a slight pressure difference after the start of combustion, which might be related to experimental uncertainties in input parameters to the computations, such as the precise injection duration, start of injection, and gas temperature at the IVC. On the other hand, the calculated heat release rate based on the experimental results seems to follow the same trend as in the simulation. However, the calculated HRR is slightly higher than the simulation experiments and seems to have a smoother drop after the end of the combustion.

Figure 5 presents the comparison of NOx and soot emissions formation for single injection cases with different start of injection timings at 2000 rpm. Simulation results follow a close trend and correspond with the measured values. Thus, the model used in this study can provide enough confidence to the following simulation results with regard to the combustion process and emissions.

![Figure 4: Comparison of simulated and measured in-cylinder pressures and heat release rates for single injection.](image)
5. RESULTS AND DISCUSSION

5.1. Piston geometry

The re-entrant piston geometry is crucial in DI diesel engines for enhancing fuel distribution within the combustion chamber. As previous research has shown, there is no optimum piston geometry for the entire engine operating conditions. On the contrary, the air flow within the chamber is highly affected by the engine speed and load.

Figure 6 presents air-fuel mixing quality within the combustion chamber at TDC, a point just before the start of combustion. In this figure, HF is plotted against the piston internal diameter (Di), bowl radius (R4) and bowl depth (T). The tests were performed in three different operating conditions. The engine speed varied from 1,200 rpm for 80% load to 1,600 rpm for 60% load and 2,000 rpm for 40% load.

The HF is reduced while the piston internal diameter is increased. This agrees with results in literature that show a smaller diameter piston bowl will generate higher swirl around the TDC and consequently will increase the air-fuel mixing quality. It is clear that HF is relatively high for pistons with the lowest piston internal diameter. However, it results demonstrate that pistons 3 and 4 have relatively lower HF due to the small difference between the outer (Da) and inner (Di) piston diameters. On the other hand, pistons with progressive diameter increments lead to higher swirl ratios and enhanced air-fuel mixing.

It can be concluded by reviewing Figure 6 that a narrow entrance with deep combustion re-entrant chambers would significantly increase the air-fuel mixing quality within the chamber. This happens due to the increased air speed that enters the piston bowl, forming a strong squish. On the other side, larger piston bowl diameters with a shallow depth have significantly less swirl within the cylinder leading to a poorer air-fuel mixing.

Figure 5: Comparison of simulated and measured NOx and soot emissions for single injection.
Figure 6: HF against the piston the internal diameter (Di), bowl radius (R4) and bowl depth (T) at TDC.
In Figure 6, HF is shown to be higher for simulations performed at 1,200 rpm compared to 1,600 rpm and 2,000 rpm. This is due to the following two reasons. First, the engine load and speed difference mean that the injection quantity and timing varies for each case. The injection strategy variation leads to a difference in the availability of time for the air and fuel to mix. Moreover, for full load test conditions (1,200 rpm case), the start of combustion has already taken place a couple degrees before the TDC leading to a more homogenous mixture at the TDC. The start of combustion for medium load cases (1,600 rpm) starts roughly a couple degrees after the TDC and for the low load conditions strategy (2,000 rpm), the start of combustion occurs five to six degrees after the TDC. In addition, the HF is higher in low speed cases as there is more available time (slower engine rotation) for the air-fuel mixing process to take place. This can also be confirmed by the subsequently lower influence of the piston geometry on the HF.

In Figure 7, the NOx and soot formation for all the piston geometries at 1,600 rpm over the homogeneity levels, at a point just before the combustion starts, at the TDC, and during the combustion process, 20° ATDC are presented.

Figure 7: Average NOx and soot formation over the HF levels at TDC and 20° ATDC for 1,600 rpm case.

In Figure 6, HF is shown to be higher for simulations performed at 1,200 rpm compared to 1,600 rpm and 2,000 rpm. This is due to the following two reasons. First, the engine load and speed difference mean that the injection quantity and timing varies for each case. The injection strategy variation leads to a difference in the availability of time for the air and fuel to mix. Moreover, for full load test conditions (1,200 rpm case), the start of combustion has already taken place a couple degrees before the TDC leading to a more homogenous mixture at the TDC. The start of combustion for medium load cases (1,600 rpm) starts roughly a couple degrees after the TDC and for the low load conditions strategy (2,000 rpm), the start of combustion occurs five to six degrees after the TDC. In addition, the HF is higher in low speed cases as there is more available time (slower engine rotation) for the air-fuel mixing process to take place. This can also be confirmed by the subsequently lower influence of the piston geometry on the HF.

In Figure 7 the NOx and soot formation for all the piston geometries at 1,600 rpm over the homogeneity levels, at a point just before the combustion starts, at the TDC, and during the combustion process, 20° ATDC are presented.

It can be seen that for the pistons with lower HF, NOx formation is reduced while the soot formation is increased. Pistons with higher HF have an increased NOx formation and reduced soot levels. However, piston geometries with the highest HF levels show a lower NOx formation than geometries with medium levels of homogeneity. Although air-fuel mixing quality is important for the performance and
emissions formation characteristics of a DI diesel engine, generalizations about air-fuel mixing homogeneity and emissions formation cannot be made.

In Figure 8, the NOx over the soot formation for all the piston geometries under the three operating conditions is presented. It is obvious for the first two operating

![Figure 8](image-url)
conditions that none of the pistons can simultaneously combine low levels of soot and NOx formation. For the 2000 rpm case, piston geometry 8 (in Figure 8) has very low levels of soot and NOx at the same time. However as shown in Figure 9, the low IMEP

![Graph 1](image1.png)

![Graph 2](image2.png)

![Graph 3](image3.png)

**Figure 9:** IMEP over BSFC for all piston geometries.
and high BSFC indicate that this is due to an incomplete combustion taking place within the piston at 2000 rpm engine speed.

Piston geometries 3 and 4 have the highest IMEP and lowest BSFC values for all of the operating conditions as shown in Figure 9. However, as it is expected, the higher IMEP levels caused by a more complete combustion within the cylinder, lead to very high NOx formation and low soot levels as shown in Figure 8.

Piston geometry 5 benefits from relatively low levels of NOx and soot at low load conditions with high IMEP and low BSFC levels. In the medium and high load conditions, the soot levels are relatively low while the NOx formation is slightly increased compared to the other geometries for the 1,600 rpm case.

Figure 8 and Figure 9 demonstrate that the performance of each of the piston geometries varies with the difference of the engine conditions. For instance, piston geometry 1 has relatively low NOx and increased soot levels at 2,000 rpm while it has very high NOx and medium soot levels at 1,200 rpm. However, the conclusion can be drawn that pistons with reduced bowl diameter and increased bowl depth result in lower BSFC levels, higher NOx formation, and IMEP.

In Figure 10, the air flow velocity and direction for three piston geometries are presented. The three pistons were selected in order to represent the difference of the air flow between shallow, wide geometries and deep, narrow pistons. Piston geometry 1

---

**Figure 10:** Air velocity contours and flow streamlines at 10° CA BTDC for piston geometries 1, 3 and 8 at 1,200 rpm and 2,000 rpm.


represents combustion chambers with a narrow entrance and wide protuberance. Piston geometry 3 is a narrow entrance combustion chamber with a deep re-entrant protuberance. Finally, geometry 8 is a wide and shallow piston.

The overall velocity within the narrow entrance pistons is greater than the velocity in the wider piston geometry at 10° BTDC in both engine speed conditions. The magnitude of the air speed is higher in high rpm conditions compared to the low rpm conditions due to the increased piston speed. The squish velocity in pistons 1 and 3 are almost twice that in piston 8. The narrow entrance into a deep combustion chamber significantly increases the speed of the air entering the piston bowl, developing strong squish flow. The squish flow in the narrow pistons geometries is roughly similar due to the very close internal diameter widths of the two pistons. However, the flow velocity changes at the center and bottom of the combustion chambers. The swirl flow in piston geometry 1 is higher than geometry 3 due to the smaller bowl area, which increases the angular velocity. Although the higher swirl velocity occurs within chamber 1, the magnitude of the flow is higher at the center of piston 3. This is caused by better interaction between the squish and swirl flow within cylinder 3.

As expected, piston geometry significantly impacts the air flow within the combustion chamber. The stronger squish-swirl interaction can develop toroidal vortices within the chamber, improving the fuel distribution and enhancing the evaporation process. The equivalence ratio and HF of the three pistons examined at 20° ATDC is shown in Figure 11. It is obvious that equivalence ratio contours cannot be easily compared for pistons with different geometries. Although piston 1 has an improved air-fuel mixing within the bowl of the combustion chamber at 1,200 rpm, the fuel is not finely distributed at the squish area for the 1,200 rpm engine speed. This is due to the low squish flow formed during the expansion stroke of the piston as a result of the bowl shape. The HF trend shows that the air-fuel mixing quality is slightly better for piston 3 compared to geometry 1. For piston geometry 8, the lack of swirl flow within the cylinder caused a low quality air-fuel mixture, which can be clearly observed by both the contour and the HF value.

The equivalence ratio contour of piston geometry 1 at 2,000 rpm shows a more balanced fuel distribution at the bowl and squish area of the piston compared to geometry 3. The HF for piston 1 rises at 67.38% at 20° CA ATDC, while for geometry 1 is 63.16%. The large HF difference between the two geometries has a high impact on the emissions and performance of the engine as shown in Figure 8 and Figure 9. The low air-fuel mixing quality within piston geometry 3 resulted in highly increased NOx formation due to fuel-rich bulky regions at the squish area as shown in Figure 12. The increased temperature resulted in higher IMEP values, lower BSFC, and lower soot formation but rapidly increased NOx formation. On the other hand, piston geometry 1 benefits from lower temperature combustion, with very low NOx and increased particulate matter.

As shown in the figures above, the air-fuel mixing quality improvement can have either negative or positive effects on the emissions formation of the engine. It seems that an improved air-fuel mixing can lead to a more complete combustion, which subsequently means higher NOx and less soot formation. However, an improved air-
fuel mixing can also aid less fuel-rich regions within the combustion chamber and therefore reduce high temperature points and NOx formation in the combustion chamber.

5.2. Injection spray angle
Piston geometry 5, which benefits from medium levels of air-fuel homogeneity and relatively low NOx and soot formation at all operating conditions, was selected to analyze the effects of the injection spray angle on the air-fuel mixing within the combustion chamber. Three spray angles were tested under different start of injection (SOI) timings. The three spray angles were targeting the main fuel injection quantity at three different locations of the piston bowl. The 130° injection spray angle points at the lower bottom of the bowl, the 145° injection spray angle points at the center of the bowl, while the 145° spray angle injects the fuel toward the upper area of the piston bowl. Figure 13 illustrates the spray cloud and HF readings at 10° CA ATDC for piston geometry 5 at 1,600 rpm engine speed under three different starts of injection timings.

The piston geometry is a key factor for picking the optimum injection angle. Figure 13 shows that an injection angle of 135° will result in some of the fuel to wet the
Figure 12: Temperature contours of pistons 1 and 3 at 2000 rpm, 20° ATDC.

Figure 13: Spray clouds and HF of piston 5 for 3 different injection angles at 1,600 rpm.
surface of the piston as it moves upwards. This results in a poorer fuel distribution within the cylinder as shown by the HF comparison in Figure 13. The injection timing variation influences the air-fuel mixing quality. An early injection results in a high degree of homogeneity within the cylinder at an early point. On the other hand, a late start of injection reduces the time available for the air-fuel mixing to occur and therefore reduces the homogeneity. However, the HF variation among the different injection angles is not highly influenced by the start of injection timings.

Figure 13 demonstrates that although the 145° spray angle points at the center of the re-entrant bowl, the air-fuel mixing quality levels are lower than in the cases with 160° spray angles. This can be justified by looking at Figure 14, which presents the equivalence ratio contours and the HF levels for piston geometry 5 at three different pistons positions. As shown in the figure, the fuel flow for the 130° spray angle follows the air flow vectors at the air swirl region of the piston bowl. This results in a high amount of fuel vapor concentrated at the bowl edges and the squish area. The 145° spray angle hits at the center of the swirl region while the fuel flow for the 160° spray angle case has an opposite direction than the air flow within the piston bowl. The opposite air and fuel flow directions create small vortices within the bowl aiding in an enhanced mixing quality. The equivalence ratio contour for the 160° spray angle shows that the fuel concentration in the squish region has been reduced while the fuel vapor distribution has been enhanced at the center of the piston bowl.

The effects of the air-fuel mixing quality on the in-cylinder temperatures are presented in Figure 15. The fuel rich squish regions due to the poor air-fuel mixing in the 130° and 145° spray angle cases have resulted in in-cylinder high local temperatures, which increased the NOx formation. On the other hand, the 160° spray angle case exhibits relatively low in-cylinder temperatures at the squish region and the sides of the piston bowl. The NOx emissions levels are 8.6% reduced for the 160°
spray angle case compared to the 145° case. On the other hand, the soot formation is increased by 28% for the 160° case compared to the 130° spray angle case. Increased air fuel homogeneity aids in avoiding fuel rich bulky points and reducing NOx formation. However, the increased homogeneity leads to an increase in particulate matter within the cylinder.

6. CONCLUSIONS
The HF has been used for investigating the effects of piston geometry and fuel injection angles on the air-fuel mixing, emissions, and performance of a DI single cylinder diesel engine. The research work carried out investigates the importance of the HF to be used as a parameter for analyzing results. The aim of this work is not to generalize diesel engine’s performance based on the parameter. The main findings of this work, within the ranges tested, can be summarized as follows:

**Figure 15:** Temperature contours at 15° CA ATDC for three different injection angles, NOx and soot emissions levels at 1,600 rpm.
A large piston bowl diameter with a shallow depth has significantly less swirl flow within the cylinder leading to a poorer air-fuel mixing. On the other hand, a narrow entrance with a deep combustion re-entrant chamber would significantly increase the air-fuel mixing quality within the chamber. This happens due to increased air speed that enters the piston bowl, forming strong squish. Moreover, pistons with progressive diameter increments benefit from high swirl and enhanced air-fuel mixing.

- The piston geometry effects on the air-fuel mixing is minimized for the low engine speed cases as there is more available time (slower engine rotation) for the air-fuel mixing process to take place.

- The air-fuel mixing quality is important for the performance and emissions formation characteristics of a DI diesel engine, however, generalizations about air-fuel homogeneity and emissions formation cannot be made. Results show that an improved air-fuel mixing can lead to a more complete combustion, which subsequently means higher NOx and less soot formation. However, an improved air-fuel mixing can also aid in less fuel-rich regions within the combustion chamber and therefore reduce high temperature points and NOx formation in the combustion chamber.

- Piston geometries with increased swirl – squish interaction flows benefit from better air-fuel mixing at the center of the cylinder.

- The opposite air and fuel flow directions create small vortices within the bowl aiding in an enhanced mixing quality. The equivalence ratio contour for the 160° spray angle shows that the fuel concentration in the squish region has been reduced while the fuel vapor distribution has been enhanced at the center of the piston bowl.

- It was confirmed that there is no piston geometry that can benefit from low emissions and high engine performance for all engine operating conditions.

The findings of this paper demonstrated that HF is a very useful indicator for studying and understanding in-cylinder air-fuel mixing and emissions formation behavior. The HF is an important parameter that can be employed for analyzing results. The research work carried out in this paper focuses on the effects of the piston geometry and spray angle on the air-fuel mixing quality, emissions formation, and engine performance. However, there are many other factors, which can highly affect the air-fuel mixing quality of a DI diesel engine. Further research is required in order to adopt HF as a combustion analysis tool that could be used in the future for forecasting the emissions formation.

ACKNOWLEDGEMENTS
This work was produced in the framework of SCODECE (Smart Control and Diagnosis for Economic and Clean Engine), a European territorial cooperation part-funded by the European Regional Development Fund (ERDF) through the INTERREG IV A 2 Seas Programme.

The authors acknowledge the AVL Company to provide computational resources for this research.
REFERENCES

[1] S. Jaichandar, K. Annamalai, Effects of open combustion chamber geometries on the performance of pongamia biodiesel in a DI diesel engine. Fuel, 2012, 98, 272–279.

[2] J. Li, W.M. Yang, H. An, A. Maghbouli, S.K. Chou, Effects of piston bowl geometry on combustion and emission characteristics of biodiesel fueled diesel engines., Fuel, 2014, 120, 66–73.

[3] B. Harshavardhan, J.M. Mallikarjuna, CFD Analysis of in-Cylinder Flow and Air-Fuel Interaction on Different Combustion Chamber Geometry in DISI Engine, International Journal on Theoretical and Applied Research in Mechanical Engineering (IJTARME), 2013, 2, 3, 104–108.

[4] F. Payri, J. Benajes, X. Margot, A. Gil, CFD modeling of the in-cylinder flow in direct-injection Diesel engines, Computer & Fluid, 2004, 33, 995–1021.

[5] J.B. Heywood, Internal Combustion Engine Fundamentals, Mc Graw-Hill New York, 1988.

[6] K.C. Tsao, Y. Dong, Y. Xu, Investigation of Flow Field and Fuel Spray in a Direct-Injection Diesel Engine via Kiva-II Program, SAE Paper 901616, 1990.

[7] L. Zhang, T. Ueda, T. Takatsuki, K. Yokota, A study of the Effect of Chamber Geometries on Flame Behavior in a DI Diesel Engine, SAE Paper 952515, 1995.

[8] A. de Risi, T. Donateo, D. Laforgia, Optimization of the Combustion Chamber of Direct Injection Diesel Engines, SAE Paper 2003-01-1064, 2003.

[9] C. Genzale, D. Wickman, R.D. Reitz, An advanced optimization methodology for understanding the effects of piston bowl design in late injection low temperature diesel combustion, In: Proceedings of THIESEL 2006 conference on “thermo and fluid dynamic processes in diesel engines”, Valencia-Spain, 2006.

[10] Y. Liu, R.D. Reitz, Optimizing HSDI Diesel Combustion and Emissions Using Multiple Injection Strategies, SAE Paper 2005-01-0212, 2005.

[11] T. Hiroyasu, M. Miki, M. Kim, S. Watanabe, H. Hiroyasu, H. Miao, Reduction of Heavy Duty Diesel Engine Emission and Fuel Economy with Multi-Objective Genetic Algorithm and Phenomenological Model, SAE Paper 2004-01-0531, 2004.

[12] P.K. Senecal, E. Pomraning, K.J. Richards, Multi-Mode Genetic Algorithm Optimization of Combustion Chamber Geometry for Low Emissions, SAE Paper 2002-01-0958, 2002.

[13] J. Song, Y. Chunde, Y. Liu, Z. Jiang, INVESTIGATION ON FLOW FIELD IN SIMPLIFIED PISTON BOWLS FOR DI DIESEL ENGINE, Engineering Applications of Computational Fluid Mechanics, 2008, 2, 3, 354–365.

[14] R. Siewert, Spray Angle and Rail Pressure Study for Low NOx Diesel Combustion, SAE Paper 2007-01-0122, 2007.

[15] R. Mobasher and Z. Peng, A Computational Investigation into the Effects of Included Spray Angle on Heavy-Duty Diesel Engine Operating Parameters, SAE Paper 2012-01-1714, 2012.
[16] R.D. Reitz, **Modeling Atomization processes in High-Pressure Vaporizing Sprays**, Atomization and Spray Technology, 1987, 3, 309–337.

[17] P.J. O’Rourke, A.A. Amsden, **The TAB Method for Numerical Calculation of Spray Droplet Breakup**, SAE Paper 872089, 1987.

[18] F.X. Tanner, **Liquid Jet Atomization and Droplet Breakup Modeling of Non-Evaporating Diesel Fuel Sprays**, SAE Paper 970050, 1997.

[19] C. Habchi, D. Verhoeven, C. Huynh, L. Lambert, J.L. Vanhemelryck, T. Baritaud, **Modeling atomization and breakup in high-pressure diesel sprays**, SAE Paper 970881, 1997.

[20] M.A. Patterson, R.D. Reitz, **Modeling Spray Atomization with the Kelvin-Helmholtz-Rayleigh-Taylor Hybrid Models**, Atomization and Sprays, 1999, 9, 623–650.

[21] L. Kelvin, W. Thomson, **Hydrokinetic solution and observations**, Philosophical Magazine, 1871, 42, 362–377.

[22] J.K. Dukowicz, **Quasi-steady droplet phase-change in the presence of convection**, Technical Report LA-7997-MS, Los Alamos Scientific Laboratory, 1979.

[23] K. Hanjalic, M. Popovac, M. Hadziabdic, **A robust near-wall elliptic-relaxation Eddy-viscosity turbulence model for CFD**, Int. J. Heat and Fluid Flow, 2004, 25, 6, 1047–1051.

[24] W.P. Jones, B.E. Launder, **The prediction of laminarization with a two-equation model of turbulence**, Int. J. Heat Mass Transfer, 1972, 15, 2, 301–314.

[25] V. Yakhot, L.M. Smith, **The renormalization group, the ε-expansion and derivation of turbulence models**, J. Sci. Comp., 1992, 7, 35–61.

[26] O. Colin, A. Benkenida, **The –zones extended coherent flame model (ECFM3Z) for computing premixed/diffusion combustion**, Oil & Gas Science and Technology, 2004, Rev. IFP. 159, 6, 593–609.

[27] Y.B. Zeldovich, P.Y. Sadovnikov, D.A. Kamenetskii, **Oxidation of Nitrogen in Combustion**, Academy of Sciences of USSR, Institute of Chemical Physics, 1947, Moscow-Leningrad.

[28] B.F. Magnussen, B.H. Hjertager, **On mathematical modeling of turbulent combustion with special emphasis on soot formation and combustion**, Sixteenth International Symposium on Combustion, 1977.

[29] H. Hiroyasu, K. Nishida, **Simplified Three Dimensional Modeling of Mixture Formation and Combustion in a DI Diesel Engine**, SAE Paper 890269, 1989.

[30] Z. Peng, B. Liu, L. Tian, L. Lu, **Analysis of Homogeneity Factor for Diesel PCCI Combustion Control**, SAE Paper 2011-01-1832, 2011.