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# Flow and Mixture Optimization for a Fuel Stratification Engine Using PIV and PLIF Techniques

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**Abstract.** This paper describes an application of PIV (particle image velocimetry) and two-tracer PLIF (planar laser-induced fluorescence) techniques to optimize the in-cylinder flow and to visualize two fuels distribution simultaneously for developing a fuel stratification engine. This research was carried out on a twin-spark four-valve SI engine. The PIV measurement results shows that a strong tumbling flow was produced in the cylinder as the intake valves were shrouded. The flow exhibited a symmetrical distribution in the plane perpendicular to the cylinder axis from the early stage of intake until the late stage of compression. This flow pattern helps to stratify the two fuels introduced from separate ports into two regions laterally. The stratification of fuels was observed visually by the two-tracer PLIF technique. During the PLIF measurement, two tracers, 3-pentanone and N,N-dimethylaniline (DMA), were doped into two fuels, hexane and iso-octane, respectively. Their fluorescence emissions were separated by two optical band-pass filters and recorded by a single ICCD camera simultaneously via an image doubling system. The PLIF measurement result shows that two fuels were well stratified.

**Keywords:** SI engine, Fuel Stratification, In-cylinder flow, PIV, PLIF

## 1. Introduction

It has been well known that charge stratification lean-burn is an effective means to reduce fuel consumption, NO<sub>x</sub> and CO<sub>2</sub> emissions in internal combustion engines<sup>[1,2]</sup>. However, this sort of mixture pattern can only be used in part load operations. At high loads the engine efficiency is still limited by knocking combustion.

Recently, a new concept of fuel stratification has been proposed and studied by the authors. This concept requires that two fractions, heavy and light, of standard gasoline or two different fuels be introduced into the cylinder separately through two separate intake ports. Two fuel fractions or fuels will then be stratified into two regions laterally in the cylinder by a strong tumbling flow. Each region has a spark plug to control the ignition. This fuel stratification concept can be used in traditional charge stratification lean-burn mode at part load conditions as long as only one fuel is introduced. At high loads, if the low octane-number (ON) fuel fraction or fuel is ignited first, the high ON fuel fraction or fuel will be left in the end gas region to resist knocking, so that the maximum allowable compression ratio can be increased to improve the fuel economy. In addition, with the high compression ratio the low ON fuel or fraction may easily lead to auto-ignition at part loads. Furthermore, the auto-ignition tendency of the low ON fuel may be enhanced by compression of the propagating flame if the combustible mixture with high ON is ignited first. This spark-assisted controlled auto-ignition combustion mode will dramatically reduce NO<sub>x</sub> and CO<sub>2</sub> emissions<sup>[3]</sup>. As a result, the fuel stratification can give a high efficiency not only at part loads but also at high load operations.

Good charge stratification is crucial to realize the proposed stratified fuel combustion operation. In order to achieve fuel stratification, the intake system of the research engine had to be modified to produce a strong tumble flow. The flow was measured and optimized by a digital cross-correlation particle image velocimetry (PIV) system. The effectiveness of fuel stratification in the presence of strong tumble flow was investigated by a two-tracer PLIF (planar laser induced fluorescence) system developed for the purpose. In this paper development and application of these two laser diagnostic techniques are described in detail. Measurement results are also discussed.

## 2. Experiment Procedures

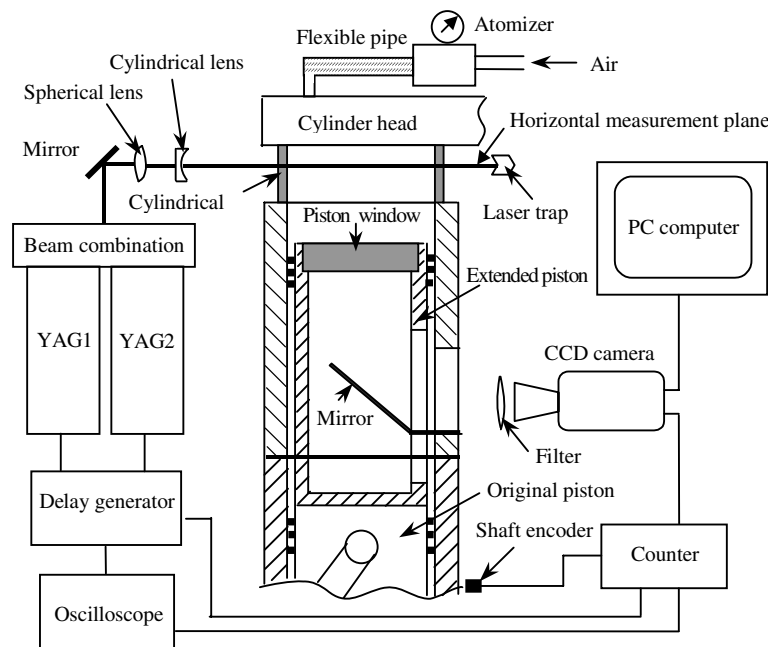
### 2.1 Optical Engine

The research was conducted on a Ricardo Hydra single-cylinder optical engine with an extended piston and an extended cylinder block designed for optical purpose. The engine has a bore of 80mm and a stroke of 89mm. The compression ratio is 11:1. A four-cylinder prototype cylinder head with twin spark plugs and three valves in each cylinder was mounted on the extended cylinder block.

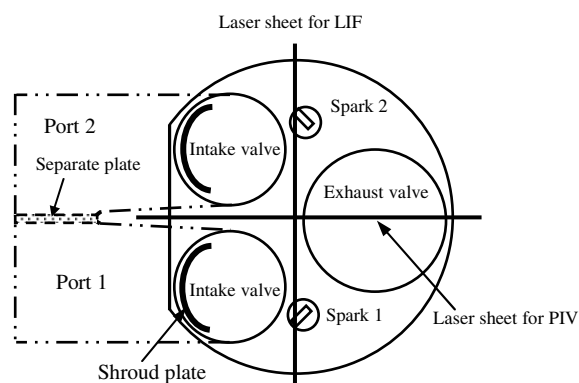
The engine has optical accesses (as shown in figure 1 and 3) via a fused silica window (55 mm in diameter) on the top of the piston and a transparent upper liner section (30mm in highness) which was installed between the cylinder head and the extended cylinder block. A UV (ultraviolet) mirror was placed in an angle of 45 degree in the extended piston to alter the direction of the field of view.

### 2.2 PIV Measurement

Figure 1 shows the schematic of PIV measurement system. Two Nd:YAG lasers with second harmonic output (532nm) were used in the PIV system. The beams were combined into a collinear beam and then were steered into the cylinder via a series of mirrors. A laser sheet with a thickness of  $425\mu\text{m}$  at its waist was formed by a spherical lens ( $f = 1000\text{mm}$ ) and a concave cylindrical lens ( $f = -76\text{mm}$ ). The measurement planes were a horizontal plane 3mm below the cylinder head (Fig.1) and a vertical plane passing through the symmetric plane of the combustion chamber shown as in figure 2.



**Figure 1.** schematic of the PIV measurement system



**Figure 2.** schematic of intake system, combustion chamber and measurement planes

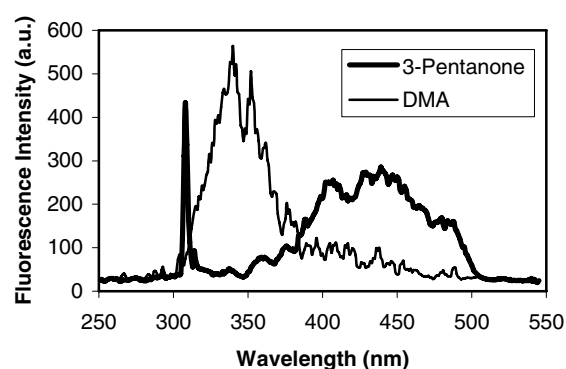
A Kodak CCD camera with a resolution of  $1008 \times 1018$  pixels was employed. A double exposure mode was applied to give two exposure pulses in an interval of 5ns and a width of 33ms. To obtain a frame-straddled pair of images, signal sequences were set in such a way that the first laser pulse was fired at the last part of the first image exposure period and the second laser pulse at the beginning range of the second exposure. A pulse generator was used to control the timing and energy output of the lasers. A fine mist of silicone oil droplets was generated by a jet atomizer and then seeded into the intake port of the engine. A PC computer and a TSI Insight 2.0 software package were used to acquire particle images, perform on-line post processing and display velocity vectors. Ensemble averaged flow parameters were calculated and analysed in order to examine the bulk flow behaviours and reduce the effect of cycle-to-cycle variation of flow.

During measuring, the engine was motored to 1200rpm at wide throttle opening. The separation time between two laser pulses was set to  $10\mu\text{s}$  during the intake stroke and to  $30\mu\text{s}$  during the compression stroke in order to account for the change of in-cylinder flow velocities. The combustion chamber was painted black to eliminate the influence of background light resulted from the surface reflection. For each measurement, 80-120 realizations were acquired from consecutive cycles.

### 2.3 Two-tracer PLIF Measurement

The present study requires measuring two fuel distributions in the cylinder simultaneously. Thus, each fuel or fuel fraction needs to be doped with a tracer whose fluorescence spectrum should be in different spectral region from the tracer doped into the other fuel. This is referred to as two-tracer PLIF technique. The fluorescence signals obtained should be separated and detected simultaneously via a specially designed optical system.

After summarizing the tracers that have been applied in PLIF measurements of engine mixture, it is found that those tracers fall into three chemical families: aldehyde, ketone and aromatic<sup>[4,5]</sup>. Aldehydic and ketonic



**Figure 3.** Fluorescence spectra of 3-pentanone and DMA

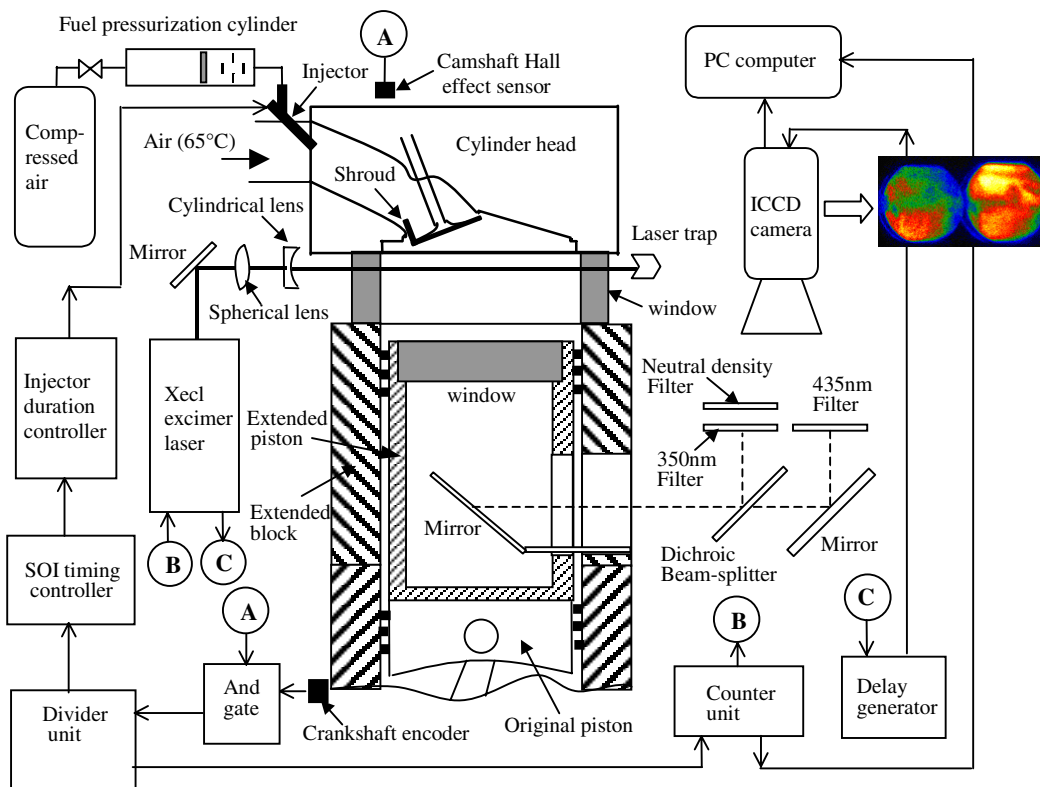
compounds have similar fluorescence emission bands from 350nm to 600nm while the aromatic compounds have fluorescence emission bands from 260nm to 430nm. Therefore, the suitable candidates of tracers for the two-tracer PLIF technique would be aldehydes/ketones and aromatics as their emissions are in significantly different spectral regions when excited by the same wavelength. In aldehydic/ketonic

compounds, 3-pentanone (boiling point 102°C) is the one that has been frequently used in gasoline engines because its boiling point is near the middle of the gasoline distillation curve<sup>[6-11]</sup>. In aromatics, N,N-dimethylaniline (DMA, 193°C) is commonly used<sup>[12,13]</sup>. A spectroscopic measurement of the two tracers was carried out in a constant volume chamber<sup>[5]</sup> and the result (shown in figure 3) revealed that the fluorescent spectrum of DMA is in a range of 310nm-440nm with a peak intensity of 340nm. While 3-pentanone has an emission band from 370nm to 500nm with a peak at 440nm. The difference in the two peak spectra was large enough to be separated by two narrow band optical filters, 350nm  $\pm$  5nm (FWHM) and 435nm  $\pm$  12.5nm.

Hexane (69°C) and iso-octane (98°C) were used as fuels, in which are doped by 3-pentanone and DMA respectively. Hexane was injected into an intake port at 70°CA BTDC in the exhaust stroke and iso-octane into the other port at 140°CA BTDC. The tracer concentration in fuels depends closely on the engine design and operation conditions. For the current study, it was found by trial and error that 25% (in weight) 3-pentanone in hexane and 10% DMA in iso-octane could give not only sufficient fluorescence intensity but also a fair intensity balance between two fluorescence images.

Figure 4 presents a schematic diagram of the two-tracer PLIF measurement system. The excitation source was a XeCl excimer laser with a wavelength of 308nm. An approximate 2mm thick laser sheet was formed using the identical optics as the PIV measurements. The measurement planes were a vertical plane passed through two spark plugs (in figure 2) and a horizontal plane of 5mm below the cylinder head (in figure 4). In order to take the two fluorescence images simultaneously by a single ICCD camera, an image doubler was set up, which consisted of a dichroic beam splitter and an UV mirror. The dichroic beam splitter reflected most of the fluorescence signal in the UV range to the camera and allowed fluorescence emission in the visible light range to pass through, which was then directed to the ICCD camera by the mirror behind it. This produced a pair of images side by side on the camera (in figure 4). Because the DMA fluorescence signal was stronger than the 3-pentanone during the intake and early compression strokes, a series of neutral density filters with different darkness were added onto the 350nm band filter to attenuate the DMA fluorescence signal.

The ICCD camera has a resolution of 578  $\times$  385 pixels. A Nikkor UV lens with a focal length of 105mm and a maximum aperture of f/4.5 was mounted in the front of the ICCD. As the image capturing system has only a frequency of around 0.2Hz, the fuel injection was then controlled by a divider to take action only in those cycles when images were to be captured. Synchronization of the engine, laser, camera and PC computer was performed by a delay generator and a counter unit, both of which were controlled by the crank angle and TDC signals from a shaft encoder. A 300-500ns exposure time, much longer than the life time of the fluorescence, was set in order to account for timing variations of laser firing when receiving an external trigger signal.



**Figure 4.** Schematic of two-tracer PLIF measurement set-up

The PLIF measurement was conducted under motored conditions with a speed of 1200rpm and a wide throttle opening. The intake air was heated to around 65°C to accelerate the vaporisation of fuels and tracers. PLIF measurement was taken at every 30°CA crank angle. For each crank angle measured, 20 fluorescence images were recorded and then followed by 5 background images which were taken under the same conditions but without fuel injection.

The images were processed in the such steps: The background images were first averaged in order to reduce the effect of shot-to-shot variation of background variations. The averaged background image was then subtracted from each PLIF image. These individual background-removed PLIF images were finally ensemble-averaged to obtain a mean fuel distribution.

### 3. In-Cylinder Flow Optimisation

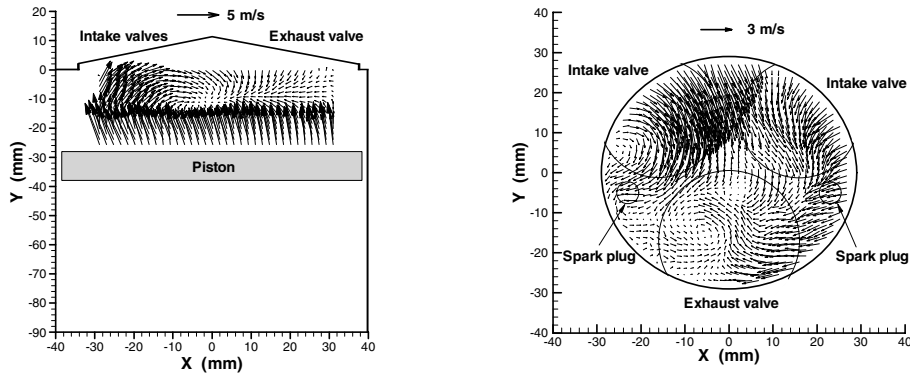
#### 3.1 In-Cylinder Flow of Original Cylinder Head

The air flow in the cylinder of the original cylinder head was measured by PIV. The mean velocity map at 60°CA BTDC of the compression are shown in figure 5. It is clearly seen that there was not large-scale tumbling flow in the cylinder. Furthermore, velocities on the horizontal measurement plane were irregularly distributed. Velocity component in the direction along two spark plugs was significant in magnitude. This flow pattern could not help to stratify the charge laterally along the direction of two spark plugs. In addition, the in-cylinder flow velocity off the piston top is only 3 m/s or less, too weak to let the fuel vapour or droplets follow with. The intake system must be modified to produce a strong tumbling flow.

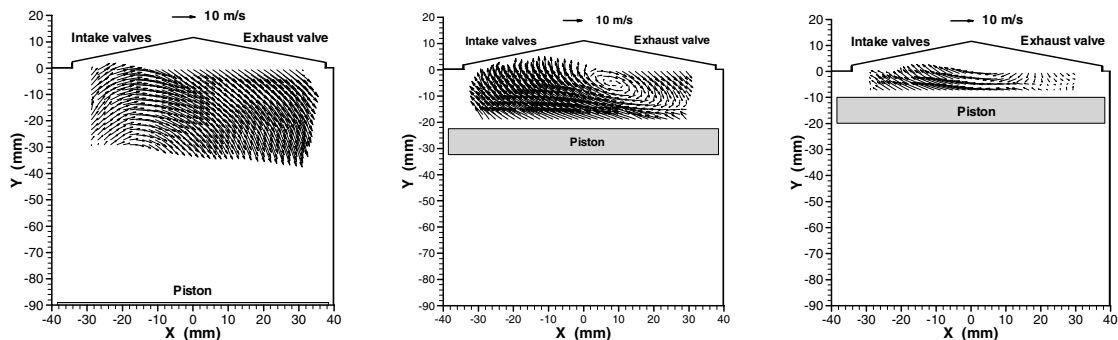
#### 3.2 Intake System Modification

Two modifications were taken for the original intake system. Firstly, the ‘Y’ shaped siamese intake port was separated into two independent intake ports by a steel plate having a thickness of 2mm (figure 2).

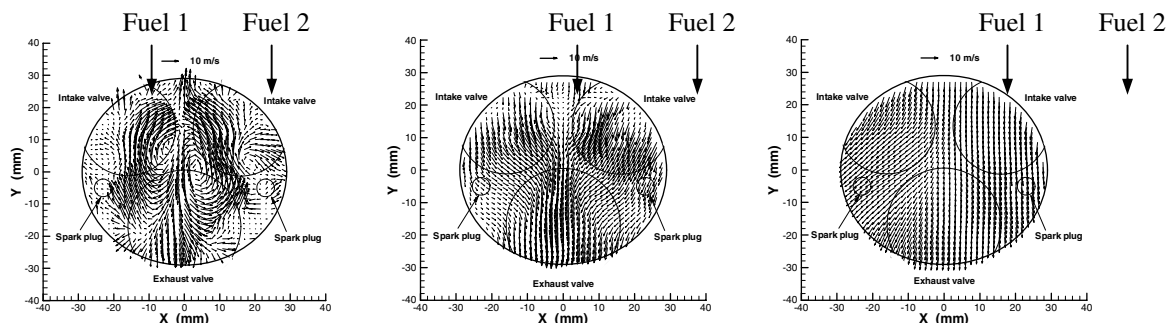
Then the lower part of the perimeter of both intake valves was shrouded by  $120^\circ$  arc plates (figure 2 and 4). To avoid the inlet valves from rotation during the engine operation, a notch-to-tooth like structure was fitted at the end of the valve stem. Steady flow rig test results<sup>[14]</sup> showed that the tumble intensity factor increased from 0.8 to 2.6 after the addition of the shrouding plates to the intake valves. The in-cylinder flow characteristics are analysed in the following.



(a) In the vertical measurement plane (b) In the horizontal measurement plane  
**Figure 5.** Mean velocity maps at  $60^\circ\text{CA BTDC}$  of the compression (1200rpm)



(a)  $180^\circ\text{CA ATDC (BDC)}$  (b)  $50^\circ\text{CA BTDC}$  (c)  $30^\circ\text{CA BTDC}$   
**Figure 6.** Mean velocity distributions in the vertical measurement plane during the compression stroke



(a)  $90^\circ\text{CA ATDC (intake)}$  (b)  $180^\circ\text{CA ATDC (BDC)}$  (c)  $60^\circ\text{CA BTDC (compression)}$   
**Figure 7.** Mean velocity distributions in the horizontal plane during the intake and compression strokes

### 3.3 Flow Characteristics after Intake Modification

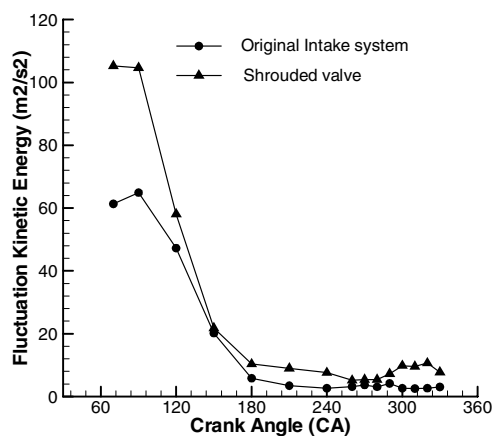
The in-cylinder velocity distribution during the intake and compression strokes after optimisation of intake system is presented in Fig.6 and 7. Following flow behaviours can be observed:

(1) A strong tumble was formed in the cylinder at the end of the intake stroke (figure 6(a)). It developed in the compression stroke (figure 6(b)) and sustained even towards the end of the compression (figure 6(c)). With the strong tumble vortex, the intake flow kinetic energy can be well kept through the intake and compression strokes until its breakdown into turbulence towards the end of the compression, leading to a significant increase in turbulent kinetic energy. This can be seen in figure 8. The turbulence kinetic energy was increased considerably in the late stage of the compression stroke comparing with the original cylinder head. This would speed up the combustion and help to extend the lean burn limit.

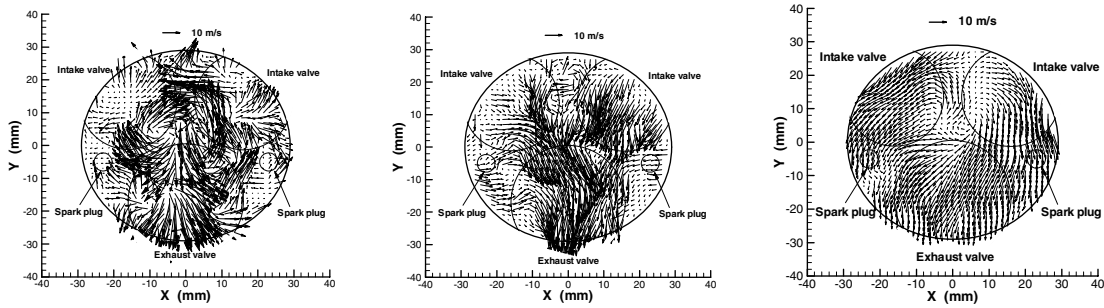
(2) In contrast to figure 5(b), figure 7 shows a good symmetrically-distributed velocity map in the horizontal measurement plane. During the intake stroke (figure 7(a) and (b)), the velocity around the symmetric plane of the combustion chamber (the line  $X=0$  in figure 7) was much greater than that in the other areas as the flow from two intake ports merged around the symmetric plane, where there were few flow components traversing the symmetric plane from one side of the cylinder into the other. This is benefit to keep apart the fuels inducted from two intake ports, hence leading to two mixture zones laterally. This means that the two fuels have been stratified laterally since inducting into the cylinder. In the compression stroke (figure 7(c)), the higher velocity values around the symmetric plane were considerably reduced. The flow field was dominated by the flow from the intake side to the exhaust side. This is beneficial to maintain the stratified pattern towards the end of the compression.

(3) The velocity in the cylinder was nearly doubled in magnitude comparing with the flow field before the intake modification (figure 5). This allows the fuel vapour or droplet to easily follow with the flow to form a stratification pattern. In addition, the flow kinetic energy was greatly increased (in figure 8).

Although the above mean velocity field presents an ideal flow pattern for the fuel stratification, each instantaneous velocity distribution does not show such ideal patterns owing to the presence of cycle-to-cycle variation of bulk flow. Figure 9 shows a few instantaneous velocities at the same measurement plane and same crank angles as shown in figure 7. In comparison to the two figures, a cyclic variation of bulk flow was apparent. Each instantaneous velocity exhibited a rather different distribution not only from the other cycles but also from its ensemble-averaged mean velocity. With the presence of cyclic variation, there were lots of velocity components passing across the symmetric plane during the intake and compression processes. This would force two fuels within two stratified regions to transfer across the symmetric plane and mix each other, undermining the fuel stratification.



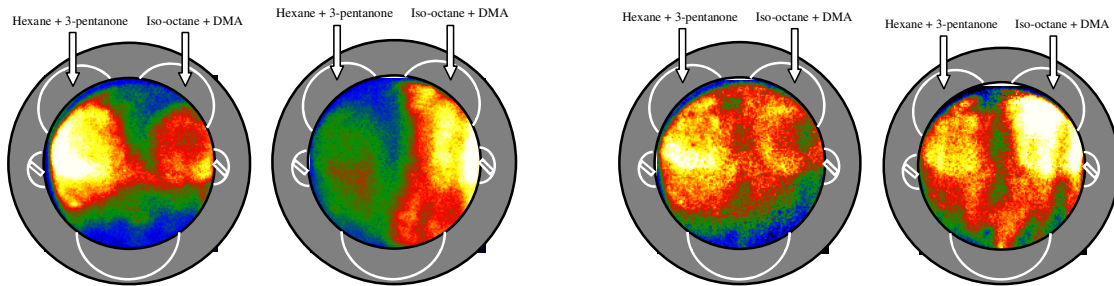
**Figure 8.** Velocity fluctuation kinetic energy versus the crank angle in the vertical measurement plane



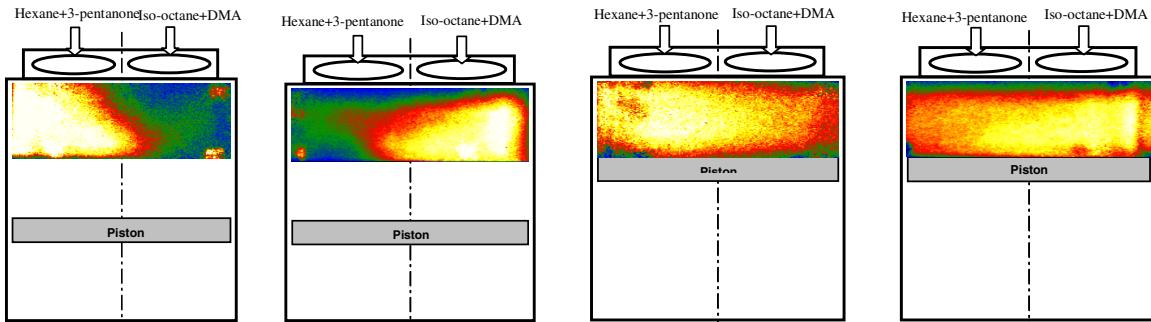
(a) 90°CA ATDC (intake)                      (b) 180°CA ATDC (BDC)                      (c) 60°CA BTDC (compression)  
**Fig.9.** Instantaneous velocity distribution in the horizontal plane during the intake and compression strokes

**4. Visualization Of Fuel Stratification**

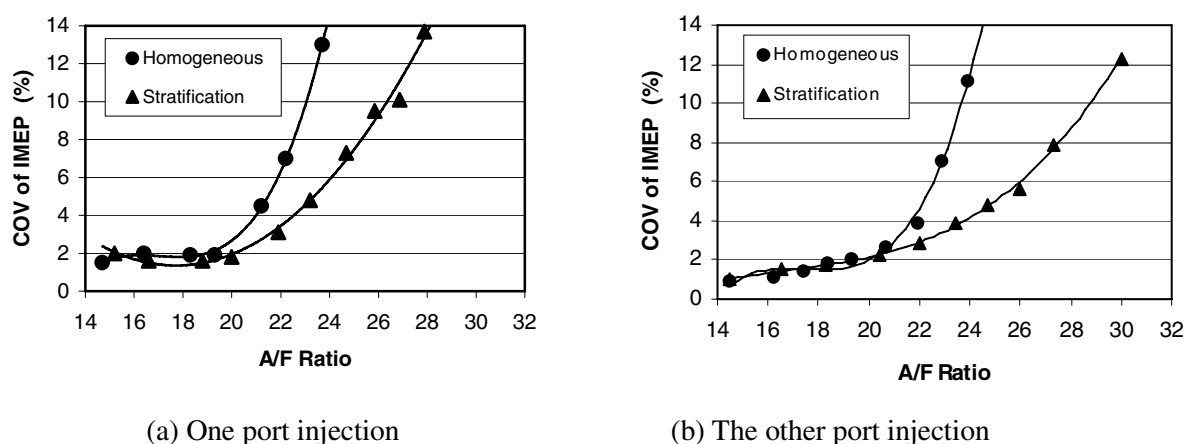
When two fuels were inducted into the cylinder through two inlet ports, the fuel distributions in the horizontal and vertical measurement planes are shown in figure 10 and 11 respectively. In these figures, Hexane doped with 3-pentanone was introduced through the left intake port, and iso-octane with DMA was injected into the right port. It is observed that two fuels were distributed dominantly in their entry side during the intake stroke. This means a formation of fuel stratification. The stratification along the direction of two spark plugs was also present during the compression stroke. However, the stratification was much clearer in intake stroke (90°CA ATDC) than in the late stage of the compression stroke (60°CA BTDC) due to velocity components along the direction of two spark plugs in each instantaneous velocity field (figure 9). This cyclic variation of bulk flow affects the fuel stratification especially when the piston approaches the top dead center during the compression stroke.



(a) 90°CA ATDC (intake)                      (b) 60°CA BTDC (compression)  
**Figure 10.** Fuel stratification in the horizontal plane at different crank angles



(a) 90°CA ATDC (intake)                      (b) 60°CA BTDC (compression)  
**Figure 11.** Fuel stratification in the vertical plane at different crank angles



**Figure 12.** COV of IMEP versus A/F ratio for fuel stratification and homogeneous combustion (IMEP=0.46Mpa, start of injection at 150°CA BTDC in the exhaust stroke, 1500 rpm)

In order to further verify the fuel stratification, a lean-burn limit test was carried out. To create a stratified mixture in the cylinder, the gasoline was injected into one intake port and the other port just allows fresh air going through. To generate a homogeneous mixture condition, the gasoline was equally injected into two intake ports. The ignition was commenced by the spark plug located in the fuel entry side. Figure 12 shows coefficient of variation (COV) in indicated mean effective pressure (IMEP) versus air/fuel ratio at the part load operation interpreted in the caption of the figure. It is clear that the lean-burn limit of charge stratification was considerably extended for each port injection. If 10% of COV of IMEP was taken as a limit of lean burn, the lean-burn limit was extended from about A/F=23 for homogeneous combustion to A/F=27 - 29 for stratification combustion. This result further supports the conclusion that good fuel stratification is formed in the present research engine.

## 5. Conclusions

This paper describes the application of PIV and two-tracer PLIF to optimise the in-cylinder flow and to visualise two fuels distribution simultaneously for developing a fuel stratification engine. The following conclusions have been made:

1. The PIV and two-tracer PLIF techniques have shown to be effective tools in optimisation of the in-cylinder flow and simultaneous visualisation of two fuels distribution.
2. The PIV measurement results show that a strong tumble is formed in the cylinder after the intake valves are shrouded. The flow field is characterized by a symmetric distribution of mean velocity in the plane perpendicular to the cylinder axis, leading to the stratification of two fuels introduced from two separate intake ports.
3. The cyclic variation of instantaneous velocity undermines the fuel stratification.
4. The fuel stratification has been clearly viewed by two-tracers PLIF measurement. It also further confirmed by a lean-burn limit test.

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