An Investigation into In-Cylinder Tumble Flow Characteristics with Variable Valve Lift in a Gasoline Engine

Tianyou Wang¹, Daming Liu¹, Binqian Tan¹, Gangde Wang¹, Zhijun Peng^{1,2*}

1) State Key Laboratory of Engines, Tianjin University, 300072, China

2) School of Engineering and Technology, University of Hertfordshire, AL10 9AB, UK

* Corresponding author: pengzj@tju.edu.cn, +86 135 21292726

ABSTRACT

In this paper, the investigation into in-cylinder tumble flow characteristics with reduced Maximum Valve Lifts (MVL) is presented. The experimental work was conducted in a modified four-valve Spark-Ignition (SI) test engine, with optical accesses for measuring in-cylinder air motion in the vertical direction. Three different MVL of 6.8 mm, 4.0 mm and 1.7 mm were tested and Particle Image Velocimetry (PIV) was employed for those measurements. Measurement results were analysed by examining the tumble flow field, the tumble ratio variation and the fluctuating kinetic energy distribution. Meanwhile, a numerical analysis method for detecting the vortex centre was developed. From results of the vortex centre distribution, the cyclic variation of the in-cylinder flow was explored. The phase-averaged flow fields show that higher MVLs could produce stronger vertical flows which turn more toward to the piston top and finally are possible to form big scale tumble flow structure. Although lower MVLs create a higher tumble ratio when the piston is close to the Bottom Dead Centre (BDC), higher MVLs substantially produce higher tumble ratios when the piston is moving close to the Top Dead Centre (TDC). In terms of kinetic energy, lower MVLs result in higher values including higher total kinetic energy and higher fluctuating energy. Finally, the vortex centres results demonstrate lower MVLs could enhance cycle-to-cycle variation due to the weakened tumble vortex.

Keywords: in-cylinder air motion, tumble flow, Variable Valve Lift (VVL), gasoline engine, Particle Image Velocimetry (PIV)

1 INTRODUCTION

The growing need to reduce CO₂ emissions and fuel consumption has led to an innovation driven increase in the thermal efficiency of gasoline engines, thus decreasing the gap to diesel engines. Although integration technologies such as hybrid powertrain systems and downsizing have played very important roles in improving gasoline engines' fuel efficiency and CO₂ emissions, there are continuing efforts in cutting fuel consumption from the optimizing combustion. In this area, Variable Valve Actuation (VVA) [1-3], Gasoline Direct Injection (GDI) [4, 5], and gasoline Controlled Auto-Ignition (CAI) [6, 7] have made and are continuing to make significant contributions. While recent developments of GDI technology have claimed to improve fuel consumption by up to 20% over conventional Port Fuel Injection (PFI) gasoline engines [8, 9], in actual fact the GDI technology is not working independently, but rather with the aid of other new systems, such as VVA or/and CAI and/or enforced induction and other relevant technologies.

VVA has been widely used on gasoline engines, including PFI and GDI gasoline engines. There are several possible control strategies in using VVA on gasoline engines: it can run as Variable Valve Timing (VVT) and/or Variable Valve Lift (VVL), although some VVA provides further different operations, such as two valve-open pulses etc [10]. As the most familiar technology of VVA, VVT has been applied practically since the last decade for improving volumetric efficiency, reducing the pumping losses and reducing harmful emissions. To achieving those benefits, EIVC/LIVC (Early/Late Intake Valve Closing), LIVO (Late Intake Valve Opening), VVO (Variable Valve Overlap) can be implemented.

The main function of VVL is to replace the throttle for regulating the intake air to achieve lower pumping losses than throttled operations. A VVL system needs to integrate appropriate cam with different profiles and corresponding Maximum Valve Lift (MVL) for the optimal relationship between discharge coefficient and crank angle [11, 12]. Compared to throttled operations, VVL could help to maintain a much higher absolute in-cylinder pressure (or lower vacuum) at the early intake stage. Only at the late intake stage, lower valve lift will reduce the in-cylinder pressure to similar values in throttled operations [13, 14]. As demonstrated by Grimaldi et al [15], the pumping loss of non-throttled VVL configuration could be decreased to approximately 50% of that of throttled condition. In recent years, VVL has been successfully applied on several models of production engines. Mitsubishi Motors Corporation developed a unique multi-mode valve system MIVEC (Mitsubishi Innovative Valve Timing and Lift Electronic Control) and has utilized it on a 4-cylinder in-line engine in early 1990s [16]. Another successful application of VVL on production engines is the new generation of BMW's valvetronic system, the one real full VVL system where the advantages of non-throttle load control can be completely realized [17]. Following those practical utilizations, new research and developments about flexible VVL have been

continuously revealed, such as TMVL (Twin Mechanical Variable Lift) valvetrain technology innovated by Ricardo Engineering Consultant Company [18] and 2-Step VVA developed by Delphi Corporation [19].

While reducing pumping losses is the main reason for employing VVL on gasoline engines, its influence on in-cylinder air motion has been noted [13, 20]. A number of researches have demonstrated that VVL's impact on in-cylinder air motion is beneficial to the combustion and emission reduction. However, under some conditions, it increases the possibility of cycle-to-cycle variation [21-24]. In an initial measurement by authors on steady flow test rig which is based on the principle originally suggested by Ricardo Consulting Engineers [25, 26], results (as shown in Figure 1) have demonstrated that reduced MVL could significantly enhance in-cylinder swirl motion, but simultaneously reduce tumble motion strength.

With those complicated influences of VVL on in-cylinder flow, it will be necessary to explore the detail how air-fuel mixing and combustion would be affected under those conditions. In the present study, the tumble flow fields under reduced MVL would be examined with a PIV system and necessary data processing technology. The aim is to investigate the impact of reduced MVL on in-cylinder air motion under relevant operating conditions. Results about tumble ratio, fluctuating kinetic energy of tumble flows will be analysed for investigating effects of VVL on the air-fuel mixing process and combustion.

2 EXPERIMENT DESCRIPTION

2.1 Optical Engine and Test Conditions

The tested optical engine for those PIV experiments is a modified Cagiva FOX350 four-valve single cylinder engine. Its main specifications can be found in Table 1. A schematic including the optical engine system and the PIV system used for tumble flow measurement can be found in Figure 2. For meeting requirements of the present measurement on tumble flow with PIV, the 27 mm long quartz cylindrical ring (as shown in Figure 3) was used for accessing the laser sheets and PIV images in the vertical planes.

Most main parts of the original cylinder head were kept except that different camshafts were fitted for different valve lifts. The engine has a general pent-roof type combustion chamber. The geometric details of four valve positions and the optical window are shown in Figure 3. The Maximum Valve Lift (MVL) of the original valvetrain was 6.8 mm. For this research, two more camshafts with MVLs of 4.0, and 1.7 mm were prepared and used on the engine. Those different camshafts still have the same valve timings as that of the original valvetrain. The valve profiles are presented in Figure 4, with 0°CA (Crank Angle) referring to the TDC (Top Dead Centre) at the beginning of the induction stroke.

During the current experiment process, the engine's intake and exhaust manifolds were not installed. Instead, simple and exhaust pipes manufactured for lab application were fitted for feeding PIV seeding and ventilating exhaust gas. Those pipes are believed resulting in very low pressure drop and having neglected influence on in-cylinder flow characteristics. The engine speed is motored at 960 rpm with an intake pressure of atmospheric pressure during all the experiments for different camshafts. In-cylinder pressure traces of different MVLS are shown in Figure 5. It can be found in Figure 5 that reduced MVL results in lower in-cylinder pressure, obviously at the peak pressure position. This demonstrates the lower MVL reduces the in-cylinder charge amount, or reduces the volumetric efficiency.

Meanwhile, the discharge coefficient with different MVL was measured for have further understanding to the effects of reduced MVL on volumetric efficiency. The measurement was conducted on a steady flow test rig which is based on the principle originally suggested by Ricardo Consulting Engineers [25, 26]. In Figure 6, it shows the variation of the discharge coefficient as function of MVL. When the MVL has an adequate high range (greater than 6 mm as shown in the figure), the discharge coefficient has almost no reduction with the decrease of MVL. However, with the MVL reduced to less than 5mm, the discharge coefficient has almost a linear reduction with the MVL.

2.2 PIV Measuring System

FlowMaster PIV system developed by LaVison was employed for the present measurements of tumble flows. The system setup can be found in Figure 2. The light source is supplied by two SOLO120 Nd:YAG lasers with their second-harmonic output (532 nm). The laser beams were sent through a spherical lens (1000 mm focal length), followed by a concave cylindrical lens (-20 mm focal length). Finally the laser sheet with a thickness of about 2 mm at the beam waist was formed. The lasers have approximately 3~5 ns pulse width and 120 mJ pulse energy. Laser pulses from two Nd:YAG lasers were controlled by a controller to fire at required timing point and with the separation of 25 μs . A trigger signal was provided to the controller from a shaft encoder which has a 1°CA resolution. A CCD camera (Sony ICX085 CCD sensor, Nikon AF Nikkor 50mm f/1.8 lens) with an image resolution of 1300×1030 and 6.7 μm pixel spacing was used for image acquisition. Seeding was added into the intake air flow by a Laskin sprayer with disoctyle sebacate as solution. 3~5 bar air in accordance with the requirement of seeding density was supplied to the sprayer by an air compressor. The particle mist had an average 1~3 μm diameter.

2.3 PIV Measurements

Synchronization with engine angular information and laser operation frequencies allowed having one

measurement, i.e. one instantaneous velocity field, at each engine cycle. 100 consecutive cycles are performed at each given crank angle (CA). When the in-cylinder tumble flows were investigated, two measuring planes with one as the vertical symmetric plane in the cylinder (Plane 1) and another through axes of same side intake and exhaust valves were selected for locating laser sheets. The arrangement can be found in Figure 7.

For processing PIV images, an interrogation spot size of 64×64 pixels is selected. The average displacement for the particle pairs in each spot is calculated using a cross-correlation algorithm and is overlapped by 50%. The Gaussian algorithm was used to detect the correlation peak in the current study. The validity of the velocity vector was judged by the ratio of the velocity peak intensity and the average intensity, which is the sum of pixel intensities in the correlation area, excluding the zero peak and velocity peak areas, divided by the number of pixels in the correlation area. Only about 5-10 per cent spurious vectors were present during the data process owing to the loss of particle pairs, or low seeding density, particularly in the intake process when greater out-of-plane velocity was present. These spurious vectors were removed using a standard deviation operator and replaced by a mean velocity of the adjacent values.

While the view area is set to $(L_x, L_y) = ([-34 \text{ mm} : 34 \text{ mm}], [-34 \text{ mm} : -2 \text{ mm}])$, 42×17 velocity points are available with spatial resolution for displayed velocity point of 1.91 mm × 1.91 mm in the measurement plane. This arrangement is for meeting the requirement to resolve as detailed as possible flow structure with a limited view area.

Some degree of random measurement error is hidden in the original velocity vectors owing to the combined effects of turbulence structure, velocity gradients, particle size and number, and even background light noise. These errors may be increased by an order of magnitude in computing turbulence parameters. Consequently, the smoothing technique which applies convolution of the two-component vector with an axisymmetric Gaussian kernel was used to attenuate the small-scale velocity fluctuation caused by these errors [27]. The Gaussian Low-Pass Smoothing Filter which takes the central pixel more into account than those outer pixels is:

$$\begin{pmatrix} 1 & 2 & 1 \\ 2 & 4 & 2 \\ 1 & 2 & 1 \end{pmatrix} / 16$$

3 DESCRIPTION OF FLOW PARAMETERS

3.1 Mean Velocity and Fluctuating Velocity

The two-dimensional instantaneous velocity vector $\vec{U}(x, y, \theta, i)$ in the cycle i at θ crank angle (CA) at the flow field in the vertical planes with the coordinates of (x, y) can be decomposed into its phase-averaged mean velocity vector $\vec{U}_{PA}(x, y, \theta)$, superimposed by fluctuating velocity $\vec{u}(x, y, \theta, i)$.

$$\vec{U}(x, y, \theta, i) = \vec{U}_{PA}(x, y, \theta) + \vec{u}(x, y, \theta, i) \quad (1)$$

where the phase-averaged mean velocity was calculated over the total cycles, N , by

$$\vec{U}_{PA}(x, y, \theta) = \frac{1}{N} \sum_{i=1}^N \vec{U}(x, y, \theta, i) \quad (2)$$

3.2 Total Energy and Fluctuation Kinetic Energy of Velocity

The total energy and fluctuating kinetic energy of velocity at point (x, y) and θ °CA in the cycle i were defined as follows:

$$E(x, y, \theta, i) = \frac{1}{2} (U^2(x, y, \theta, i) + V^2(x, y, \theta, i)) \quad (3)$$

$$k(x, y, \theta, i) = \frac{1}{2} (u^2(x, y, \theta, i) + v^2(x, y, \theta, i)) \quad (4)$$

where $U(x, y, \theta, i)$ and $V(x, y, \theta, i)$ were the components of $\vec{U}(x, y, \theta, i)$ in the x and y dimensions, respectively; $u(x, y, \theta, i)$ and $v(x, y, \theta, i)$ were components of the fluctuating velocity $\vec{u}(x, y, \theta, i)$ in the x and y dimensions, respectively.

The spatial averaged total energy $E_{SA}(\theta, i)$ and fluctuation kinetic energy $k_{SA}(\theta, i)$ were counted as follows:

$$E_{SA}(\theta, i) = \frac{1}{N_x N_y} \sum_{(x, y)} E(x, y, \theta, i) \quad (5)$$

$$k_{SA}(\theta, i) = \frac{1}{N_x N_y} \sum_{(x, y)} k(x, y, \theta, i) \quad (6)$$

where N_x and N_y are measured points at x and y directions, respectively.

Then $E_A(\theta)$ and $k_A(\theta)$ as the phase averaged $E_{SA}(\theta, i)$ and $k_{SA}(\theta, i)$ in total N engine cycles, respectively, are defined as:

$$E_A(\theta) = \frac{1}{N} \sum_{i=1} E_{SA}(\theta, i) \quad (7)$$

$$k_A(\theta) = \frac{1}{N} \sum_{i=1} k_{SA}(\theta, i) \quad (8)$$

2.3 Tumble Ratio

The spatial averaged tumble ratio TR at θ °CA in the cycle i could be calculated from the two dimensional velocity fields as follows:

$$TR(\theta, i) = \frac{\sum_{(x,y)} \vec{r}(x, y, \theta, i) \times \vec{U}(x, y, \theta, i)}{\omega_c \sum_{(x,y)} \vec{r}^2(x, y, \theta, i)} \quad (9)$$

where ω_c was the engine angular speed and $\vec{r}(x, y, \theta, i)$ was the distance between the velocity vector and the tumble centre. To be noted, the coordinate system origin used here is the tumble centre in the measured plane.

When the piston moved close to the BDC (Bottom Dead Centre) and no obvious vortex centre could be found in the measuring area, the vortex centre was actually decided by assuming that the flow field had symmetrical structure around the cylinder centre. Although there might be significant error for some cases, this assumption could help to provide at least qualitative information of tumble ratio when the piston moved close to the BDC.

4 RESULTS AND DISCUSSION

4.1 Effect of MVL on Tumble Flow Field

Figure 8(a) and (b) show the tumble flow fields of Plane 1 and Plane 2 under MVLs of 1.7 mm, 4.0 mm and 6.8 mm at crank angle of 138°CA at which intake valves are fully opened. Plane 1 and Plane 2 are as

demonstrated in Figure 7. It needs to be noted those flow fields are phase-averaged of 100 continuous cycles. One of main features in the figure is that the maximum flow velocity decreases obviously in both planes with the increase of MVL. This may be because the adequate intake air amount for higher MVL during the early intake stage resulted in very small in-cylinder vacuum and low intake flow velocity. For the flow field, it can be found in Figure 8(a) that with the MVL of 1.7 mm, two oppositely rotating tumble structures exit, and this is presented more clearly in Figure 8(b). For the MVL of 4.0 mm, two tumble structures can still be seen. But it becomes only an obvious tumble vortex for the MVL of 6.8 mm. This suggests there is a strong interaction between the intake air flow and the cylinder wall under the MVLs of 1.7 mm and 4.0 mm. When the MVL increases to 6.8 mm, the take air flows toward more to the piston top direction. As a result, this forms the single rotation direction structure, with a large scale structure.

Although the whole range of the in-cylinder tumble flow can't be imaged because of the limit of measurement area, the result from the current measurable region shows that a large scale with the same rotating orientation is easier to be formed under high MVL, while small scale counter- clockwise rotating flows exist because of the interaction and obstruction among the flows moving from different sides of the cylinder wall under low MVL condition.

Figure 9(a) and (b) show the Phase-averaged flow fields in two measurement planes with MVLs of 1.7 mm, 4.0 mm and 6.8 mm when intake valves completely closed (238 °CA). It can be found in Figure 9(a) that the distortion and break happen on the tumble structure owing to the compression. For the MVL of 1.7 mm, the clockwise tumble structure on the left side of the cylinder has decayed. The flow motions at two planes present similar flow structure and a clockwise large scale tumble structure is dominant in the flow field, and the maximum velocity is approximately 7 m/s. For the MVL of 6.8 mm, it can be found the tumble structure has broken in the measurement plane and no obvious vortex can be found. At Plane 2 (directly under valves) in Figure 9(b), velocity trends have apparently become weak while lower MVLs still have high velocity vectors.

When the piston moves to near the end of the compression stroke, as shown in Figure 10 (at 310 °CA), the obvious big vortexes for the MVLs of 1.7 mm and 4 mm have disappeared in the measurement area. But as shown in Figure 10(a), the flow field of the MVL of 6.8 mm presents a big scale vortex which did not appear in the measurement area at 138 °CA and 228 °CA. This suggests that the main tumble flow energy of the MVL of 6.8 mm may exit in the cylinder bottom area which can't be seen at 138 °CA and 238 °CA. This should be due to more vertical intake flow velocity for the MVL of 6.8 mm.

4.2 Effect of MVL on Tumble Ratio

In Figure 11(a) and (b), it shows the variations of tumble ratio as the function of crank angle. As is shown

in Figure 11(a), between the crank angle of 138 °CA and 180 °CA which is during the intake stroke, the counter-clockwise rotating flow takes the predominant place in the measurement plane for the MVL of 1.7 mm, then high positive tumble ratio values are produced. For MVLs of 4.0 mm and 6.8 mm, lower tumble ratio with negative values are presented due to the intake air flows' domination at clockwise direction. Between 130 °CA and 220 °CA, the MVL of 4.0 mm and the MVL of 6.8mm produced very similar tumble ratio (direction and value), but during the compression stroke, the tumble ratio of the MVL of 4.0 mm decayed quickly, while the tumble of 6.8 mm MVL kept the direction and strength very well. This should still come from the reason as mentioned in the last section – the more tumble flow energy of the MVL of 6.8 mm stays just above the piston top. Similarly the tumble ratio of 1.7 mm MVL becomes weak very quickly during the compression stroke due to unapparent vortex structure, though most of velocity vectors are still big in the measurement area.

Figure 11(b) reveals that, a counter-clockwise flow structure dominate the flow for the MVL of 1.7mm at the early intake stroke, as the piston moves to the BDC, clockwise tumble structure predominate in the plane near the BDC (160 °CA to 180 °CA) until the end of the compression of the compression stroke. For the MVL of 6.8 mm, as the air intake in the early stage of the intake stroke has made the in-cylinder vacuum so small, the intake flow velocity is very low and this results in the very low tumble ratio (in the measurement area).

From Figure 11(a), it indicates at the end of compression stroke, the modulus of the tumble ratio with the MVL of 6.8mm can be up to 0.6 and is much larger than that of other two MVLs. For GDI engines which have fuel injection around this time, this is especially beneficial for fuel-air mixing. This suggests propitious in-cylinder tumble motion can be provided for GDI fuel-air mixing when higher valve lift is operated.

4.3 Effect of MVL on Fluctuation Kinetic Energy

In this section, the fluctuating of the velocity was examined through the kinetic distribution using equations (5) and (6). When the kinetic energy of the flow was calculated using equation (3) and (4), the variation of the in-cylinder density was not taken into account due to the difficulty to obtain the density distribution in cylinder. In the other hand, the actual spatial variation of in-cylinder density is normally small enough to be ignored, in particular for gasoline engines.

As shown in Figure 12(a) and (b), those are the velocity kinetic energy variations as function of the crank angle for the MVL of 1.7 mm and 6.8 mm. For each MVL, shown results comprise the total kinetic energy and the total fluctuation kinetic energy. Compared to the MVL of 1.7 mm, both total kinetic energy and total fluctuation kinetic energy of 6.8 mm MVL have much small values at all crank angle

positions, though as demonstrated in last section, the MVL of 6.8 mm produced a higher tumble ratio than the MVL of 1.7 mm at the compression end.

From results of both the MVL of 1.7 mm and 6.8 mm, it can be found that the fluctuation kinetic energy is dominant in the total velocity energy. The proportion of fluctuation kinetic energy to the total velocity energy is higher than that acquired by Druault et al [28]. Some of the reason is that the engine speed was set lower in the present study, which may enhance the velocity fluctuation [29]. Moreover, the energies of the flow fields were enhanced largely as the MVL was decreased, which can be seen by comparing the same kind of energy curves at the different MVL conditions. The total energy at lower MVL conditions was about 4 times more than those at the higher MVL conditions and the crank angle at which the peak reached was advanced by about 30 °CA.

Shown in Figure 13(a) and (b), they are the relative standard deviation of spatial averaged total and fluctuation kinetic energy versus crank angle, which can be used to demonstrate the cycle-to-cycle variation of each kinetic energy. The standard deviation values were normalized with the mean spatial averaged kinetic energy. For the MVL of 6.8 mm (as shown in Figure 13(b)), the curves of each kinetic energy decreased slightly on the first half of the induction stroke (70-138 °CA), then increased in the compression stroke. At the middle of the compression stroke, the curves reach the second highest peak at about 220 °CA. Finally, after a little descent, the curves reach the highest peak at 310 °CA. This may be because in this stage the tumble flow broke up into smaller vortices, which caused the large-scale fluctuation becoming violent.

By comparing the deviation curves under low MVL (1.7 mm, in Figure 13(a)) with high MVL (6.8 mm, in Figure 13(b)), the same tendency can be observed. But the value of relative standard deviation for low MVL condition before 238 °CA is 0.07 higher than higher MVL condition. The deviation of total fluctuation kinetic energy became larger significantly and deviated from the curve of total kinetic energy during the induction stroke. Moreover, the deviation of total kinetic energy had a large increase at 310 °CA by about 0.08 than that at the higher MVL condition. The change of deviation curves meant that the cycle-to-cycle fluctuation was enlarged as the MVL lowered.

Figure 14 (a) and (b) show the spatial distribution of the fluctuating kinetic energy under three MVLs at two different planes. It can be found in Figure 14(a) that, for the MVL of 1.7 mm and 4.0 mm, the fluctuating kinetic energy is larger than that of 6.8 mm MVL. The spatial distribution shows some heterogeneity. For the MVL of 1.7 mm, the maximum of the fluctuating kinetic energy is located near the centre of the cylinder, while that of 4.0 mm MVL is near the intake-exhaust valve side, which may be caused by the influence from the obstruction of the boundary. For all three MVLs at Plane 2 (under

intake-exhaust valves), the spatial distribution of fluctuating kinetic energy is more symmetrical and the portion with larger value is located under the intake valve.

4.4 Cycle-to-Cycle Variation of Large-Scale Flow Structure

From practical measurement results, it has been found the tumble motion was rather unstable not only temporally but also spatially. Those may be due to dynamically changing boundaries imposed by the moving piston and valves. Even if the phase-averaged velocity showed clearly tumble motion under some condition, the tumble centres could only be seen in certain cycles from the low pass filtered velocity fields (as shown in Figure 15). In addition, it can be seen in Figure 15, the flow field was rather different cycle by cycle, even for the continuous cycles. In order to examine the variation characteristics of the tumble centre which should be important for influencing the cycle-to-cycle variations of gasoline engines, the tumble centre will be detected in the following section and its variation will be analyzed.

Because of the limitation of the cylindrical window, the tumble centre may be located out of the measurement range for some cycle, when the piston is close to the BDC. In the study of the cycle-to cycle variation of the tumble centre, only flow fields at 250, 280 and 310 °CA which are during in the second half of the compression stroke were selected to detect the tumble centre for each cycle. With the aim to improve the accuracy of the tumble centre detection and shortening the data processing time, the predictor-corrector scheme as introduced in [30, 31] was employed.

Two predictor methods for finding the region containing the potential vortex centres are streamlines method and structured grid method, while the corrector method is the pattern matching algorithm using the Hamel-Oseen model which is for checking velocity locating in the reduced region (obtained through the predictor approach). The predictor-corrector has numerous benefits as it combines the advantage of the pointed-based methods, which are computationally less expensive with the more accurate but time consuming scale based methods.

For the purpose of detecting tumble centre precisely, the low pass filtered velocity fields were used to avoid the interference of small scale fluctuating velocities. In Figure 16, it shows the results of tumble centre detection in symmetric plane at 280 °CA for certain cycles. The largest area marked by the red '×' in the velocity field is the region containing the potential tumble centre, which is detected by the predictor method. In this area, the corrected result marked by the green '+' is obtained by the corrector method. Finally the point with the minimum velocity was considered as the tumble centre marked by the blue '•'.

After the method for detecting the tumble centre was developed and the tumble centres were successfully examined those flow fields, the cycle-to-cycle variation of tumble centre was analyzed. For presenting the result, the box-and-whisker plot was employed. In descriptive statistics, a box-and-whisker plot is a convenient way of graphically depicting groups of numerical data through their summaries: 10th percentile, 25th percentile, median, 75th percentile, 90th percentile, interquartile range and interdecile range, etc (as shown in Figure 17). Box-and-whisker plots display the differences among populations without making any assumptions of the underlying statistical distribution. The spacings between the different parts of the box help indicate the degree of dispersion (spread) and skewness in the data, and identify outliers. For the statistics data of velocity fields, the interdecile range was used to demonstrate the cycle-to-cycle variation of the tumble centre and circulation, in the present research.

As indicated in Figure 16, the interdeciles of x-coordinate were extended as the piston moving up (from 250, to 280 and to 310 °CA), for MVLs of 1.7 mm, 4.0 mm and 6.8 mm, respectively. For the MVL of 6.8 mm, the interdecile of x-coordinate at 250 °CA was dispersed in [-11.0, 16.0] (expressed in millimeters) and extended slightly with the increase of crank angle. For lower MVLs, the extension of the x-coordinate interdeciles become larger, compared to the MVL of 6.8 mm (Figure 18 (a) and (b)). It can also be seen that interdeciles of y-coordinate were narrower than that of x-coordinate and less affected by the MVL variation. This suggests that the interdeciles of the tumble centres related to the horizontal direction (y coordinate) were more dramatic than that related to the vertical direction (x coordinate) with the variation of MVL. This may be attributed to the positive effect of the piston motion on the re-centring of the tumble and thus on the limitation of the instantaneous tumble centre variability.

5 CONCLUSIONS

The investigation into in-cylinder tumble flow under reduced Maximum Valve Lifts (MVL) has been carried out on an optical test engine. PIV and relevant data analysis methodology have been applied for examining tumble flow field, tumble ratio variation and fluctuating kinetic energy distribution. Furthermore, a numerical analyzing method for detecting vortex centre has been developed and with those results of vortex centre distribution, the cycle-to-cycle variation of the in-cylinder flow has been explored. The following conclusions have been draw from the present study.

- Results of phase-averaged flow fields demonstrate that higher MVLs can produce more vertical flows which turn more toward to the piston top and finally are more possible to form big scale tumble flow structure.

- Although the result of lower MVLS show a higher tumble ratio when the measuring range can't cover most cylinder area, higher MVLS substantially produce higher tumble ratios which can be confirmed when most cylinder area lies in the measuring range with the piston's moving close to TDC. The big scale vortex structures produced by higher MVLS made main contribution for producing high tumble ratios of higher MVLS.
- In terms of kinetic energy, lower MVLS result in higher values due to higher velocity vectors, though most time they did not form regular big vortex structure. Based on higher total kinetic energy under lower MVL condition, higher fluctuating energy is presented under lower MVL conditions. The result would in the one hand be helpful for better air-fuel mixing and flame propagation, but in the other hand would enhance cyclic variation.
- With appropriate numerical method, the vortex centres under different MVL condition were detected and the cycle-to-cycle variation of flow structure was explored. Results suggest the main reason for the increase of the cyclic variation is due to the weakened tumble vortex. Those can be found with lower MVLS under which it is found to maintain big scale tumble structures to the compression end is difficult. The results also suggest that for GDI engine with VVL operation, it is essential to enlarge possible tumble or swirl flow for reducing the cyclic variation for the interest of the stability of ignition and combustion.

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Figures

Figure 1 Impact of reduced MVL on the discharge coefficient

Figure 2 Optical engine set-up and PIV system for tumble flow measurements

Figure 3 Optical measurement area for investigating tumble flow

Figure 4 Valve lift profiles of VVL

Figure 5 In-cylinder pressures

Figure 6 Impact of reduced MVL on the discharge coefficient

Figure 7 Optical measurement planes for investigating tumble flow

(a) at Plane 1

(b) at Plane 2

Figure 8 Phase-averaged flow fields at 138 °CA (during the intake stroke) for MVL 1.7 mm, 4.0 mm and 6.8 mm, (a) at Plane 1 and (b) at Plane 2

(a) at Plane 1

(b) at Plane 2

Figure 9 Phase-averaged flow fields at 238°C (during the compression stroke) for MVL 1.7 mm, 4.0 mm and 6.8 mm, (a) at Plane 1 and (b) at Plane 2

Figure 10 Phase-averaged flow fields at 310°CA (near the compression end) for MVL 1.7 mm, 4.0 mm and 6.8 mm, (a) at Plane 1 and (b) at Plane 2

Figure 11 Tumble ratio variations as function of crank angle, (a) at Plane 1 and (b) at Plane 2

(a) MVL=1.7 mm

(b) MVL=6.8 mm

Figure 12 Averaged fluctuation kinetic energy of 100 cycles at Plane 1 (FKE – Fluctuation Kinetic Energy, KE – Kinetic Energy)

(a) MVL=1.7 mm

(b) MVL=6.8 mm

Figure 13 Relative standard deviation of spatial averaged total and fluctuation kinetic energy of 100 cycles at Plane 1 (FKE – Fluctuation Kinetic Energy, KE – Kinetic Energy)

MVL=1.7 mm

MVL=4.0 mm

MVL=6.8 mm

(a) at Plane 1

(b) at Plane 2

Figure 14 Fluctuating kinetic energy distribution (at 310°C) with three different MVLs, (a) at Plane 1 and (b) at Plane 2

(a) Phase-averaged velocity field of 100 cycles
Max velocity in the field is 3.6594 m/s

(b) Low-pass filtered velocity field (4th cycle)
Max velocity in the field is 6.3502 m/s

(c) Low-pass filtered velocity field (5th cycle)
Max velocity in the field is 6.618 m/s

(d) Low-pass filtered velocity field (6th cycle)
Max velocity in the field is 6.63 m/s

Figure 15 Cycle-to-cycle variation of tumble Flow fields (at Plane 1) at 280°CA with $MVL=6.8 \text{ mm}$

(a) MVL = 1.7 mm (13th cycle)
Max velocity in the field is 12.3395 m/s

(b) MVL = 6.8 mm (9th cycle)
Max velocity in the field is 5.1097 m/s

Figure 16 Tumble centre detected for low-pass filtered velocity fields of 100 cycles at 280°C (at Plane 1)

Figure 17 Description of Box-and-Whisker plot presented in the following figures

(a) MVL=1.7 mm

(b) MVL=4.0 mm

(c) MVL=6.8 mm

Figure 18 Cycle-to-cycle variation of tumble centre in x and y coordinate for low-pass filtered velocity fields and tumble ratio for instantaneous velocity fields of 100 cycles at Plane 1

Tables

Table 1 Main parameters of FOX350 Optical engine

Valve number	4
Bore x Stroke (mm x mm)	78.7 x 66.0
Compression ratio	6.7:1
Combustion chamber	Pent-roof shape
Intake valve diameter (mm)	30.0
Exhaust valve diameter (mm)	26.0
Valve timings	EVO: 74 °CA BBDC EVC: 26 °CA ATDC IVO: 22 °CA BTDC IVC: 68 °CA ABDC