

A Detailed Study of In-Cylinder Flow and Turbulence using PIV

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ABSTRACT

In-cylinder velocity measurements were acquired in a two-valve, single-cylinder research engine to study the bulk fluid motion and small-scale turbulence. Different port geometries (two), different port orientations (two) and both shrouded and non-shrouded intake valves were tested to vary the intake-generated flow. Tests were performed at engine speeds of 300, 600, 900 and 1200 RPM with an atmospheric intake pressure. Prior to testing on the engine, the different head configurations were tested on a steady flow bench. Particle image velocimetry data were taken on a single plane, parallel to the piston surface, in the engine using both a low magnification to characterize the large-scale flow phenomena, and a high magnification to characterize the turbulence field. The low-magnification results showed that the swirl center location was relatively insensitive to engine speed, but did change position throughout the cycle. The solid-body rotation rate was found to be comparable to the steady-flow bench swirl ratio for swirl-dominated flows, and the decay in the rotation rate during the cycle was measured. The high magnification data (acquired at TDC) were investigated using both ensemble- and spatial-averaging to define the mean flow field, and all of the results from the spatial-average method were investigated as a function of cutoff frequency. The turbulence intensity was found to be linear with engine speed, and the ratio of turbulence intensity to mean piston speed was found to be a unique function of cutoff frequency. Two -point spatial correlation functions were computed, and used to evaluate 8 of the possible 27 integral length scales. The longitudinal length scales were found to range from 5-8 mm, were insensitive to direction of separation, and were on average twice the lateral length scale, indicating a high level of isotropy in the flow. Turbulent kinetic energy spectra were calculated, and were found to show an extended inertial subrange for the higher engine

speeds; the spectra were fit well by the model spectrum of Pope. Lower engine speeds and the use of high cutoff frequencies in the spatial-averaging method were found to reduce the presence of the inertial subrange, and may result in a low Reynolds number condition where the turbulence is not fully developed and scale separation is not achieved. The spectral analysis provided length scales (L_{11}) estimates that were approximately $2.5\times$ smaller than those from the correlation results.

INTRODUCTION

The in-cylinder flow of an internal combustion engine strongly affects the combustion process. It has long been understood that the large-scale flow field directly influences the level of turbulence throughout the compression stroke of piston engines [1,2,3,4,5,6]. Studies have shown that there is a direct and nearly linear relationship between the in-cylinder turbulence and the turbulent flame speed in spark-ignition engines [7, 8], which directly impacts engine performance and emissions. It is, therefore, of great importance to understand the bulk flow field and the resulting turbulence, and ultimately their effect on the combustion process.

There have been a number of studies that have investigated the in-cylinder flow of internal combustion engines. In-cylinder velocity measurements have been made using hot-wire anemometry (HWA), laser Doppler velocimetry (LDV), and particle image velocimetry (PIV) to determine the relation between the engine speed and the turbulence [1, 2, 9,10,11,12,13,14,15,16,17]. These studies have shown that the turbulence intensity is nearly a linear function of the engine (mean piston) speed, *i.e.* there is a proportional increase in the turbulence intensity as the engine speed increases. The geometry of the engine and the details of the intake system combine to determine the speed-scaling proportionality constant.

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The characterization of the turbulence requires more than just the turbulence intensity. Integral length scale measurements, which give an estimate of the size of turbulent eddies in the flow, have been made in several studies. Many of the earlier investigations used Taylor's hypothesis to indirectly calculate integral length scales [9, 21,22,23]. Time-varying velocity data were collected and Taylor's hypothesis used to relate the time scales to length scales. However, the conditions for which Taylor's hypothesis is valid are generally not satisfied in the non-stationary flow field of an internal combustion engine. Direct measurements of the integral length scales have been performed using multi-point LDV and PIV measurements [18,19,20, 25, 26]. Longitudinal length scales were found to decrease during the compression stroke, where a minimum was reached very near TDC. In one study [19] an increase in the longitudinal length scale was observed with an increase in engine swirl ratio, while [27] observed an opposite trend. Fraser and Bracco [20, 25] measured the longitudinal length scale at a number of engine crank angles and normalized the data by the engine clearance height. A maximum value of about 0.2 near TDC was observed [20], but this depended on the data analysis method used in calculating the mean flow velocity [25]. Li *et al.* [24] measured integral length scales at five different points in the cylinder using PIV data. Their results showed that the integral length scales varied in magnitude over the flow field and also varied when calculated using different components of the fluctuating velocity.

PIV has increasingly been used in the analysis of in-cylinder flows [24, 27,28,29,30,31,32,33,34,35]. Typically, a Reynolds decomposition of the flowfield is performed to separate the mean velocity from the fluctuating component. One of the challenges of data analysis in engine flows is to define an appropriate mean velocity. The method used to calculate the mean velocity field directly affects the subsequent analysis. Many data acquisition systems are only fast enough to take a single set of PIV images per engine cycle. One method of calculating the mean velocity field then is to acquire the velocity data at a set crank angle for a large enough number of engine cycles. The data are then averaged over all the cycles at each point to find an ensemble average [27, 31, 32]. Another method of calculating the mean velocity field at a specified crank angle for each cycle is to perform a cycle-resolved analysis [24, 28, 29]. The two dimensional Fourier transform is performed on the velocity field of an individual cycle, a filter is applied using an appropriate cutoff lengthscale, and the inverse Fourier transform is performed resulting in a filtered mean velocity field for that cycle. Sufficiently fast data acquisition systems allow for a time-resolved mean velocity field to be calculated for an individual cycle [33], as was performed with single-point LDV measurements. Filtering methods not using a Reynolds decomposition are also reported in the literature [34].

A variety of parameters of interest have been studied using PIV. In engines with predominant swirling flows, the incylinder swirl has been found to decrease approaching TDC, but the peak swirl occurs at crank angles depending on engine operating conditions and measurement location [28, 31,35]. A precession of the swirl center with crank angle has also been reported [28, 35]. Li *et al.* [35] showed that the swirl center precession during the compression stroke was very similar at two different engine speeds. Energy and dissipation spectra have also been reported in the literature [27, 34, 36].

The motivation for the current study is to investigate size- and speed-scaling effects in geometrically similar engines; the authors have designed two geometrically scaled, optically accessible research engines. In the present paper, results are only presented from the larger scaled engine. The purpose of this paper is to establish a data processing methodology for PIV data that characterizes the turbulent flow field as fully as possible. This characterization includes the large-scale bulk motion as well as the small-scale turbulence properties. In order to reduce biases associated with limited spatial resolution in the turbulence statistics, a higher image magnification was used to investigate the turbulence data; the bulk flow characterization was performed with a lower magnification to allow full coverage of the visible portion of the cylinder. The small-scale turbulence properties were investigated using various methods and the results analyzed to investigate the relative agreement of each method. The engine features a modular intake port that allows multiple port geometries to be tested with the same engine configuration; two different intake port geometries have been designed and tested. The ports can be rotated about the valve axis to vary the swirl or tumble generated. The combustion chamber resembles as closely as possible a right cylinder to aid in data analysis and to make the results amenable to computations.

EXPERIMENTAL APPARATUS

OPTICAL ENGINE

Experiments were performed in an optically accessible, Bowditch-type single cylinder research engine, shown in Fig. 1. The engine head features one intake and one exhaust valve. The valves were ground so that they are flush with the (flat) engine head deck when closed. The upper portion of the cylinder was replaced with a ring window; the piston surface is flat and contains a centered window that permits optical access to 53% of the cylinder bore. The design intent of the engine cylinder was to as closely approximate a right-cylinder volume as possible. Experiments were performed at engine speeds of 300, 600, 900, and 1200 rpm with a manifold pressure of ~100 kPa. The intake air temperature was controlled to 323 K.

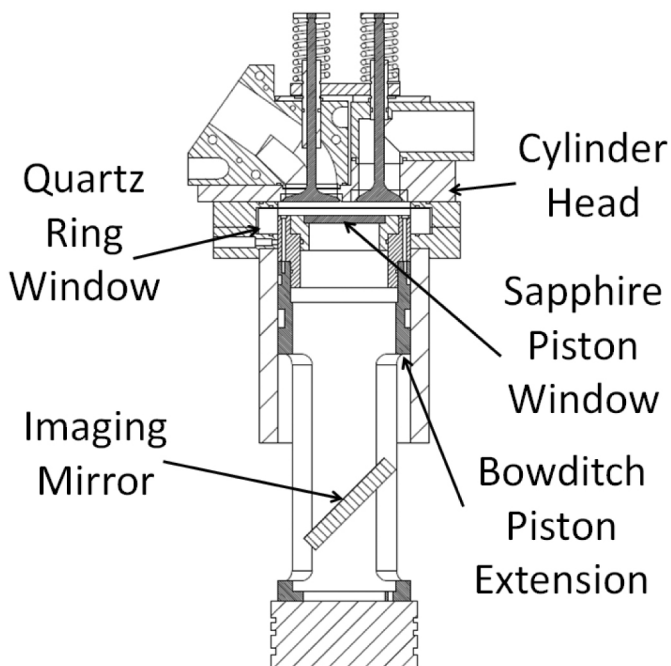


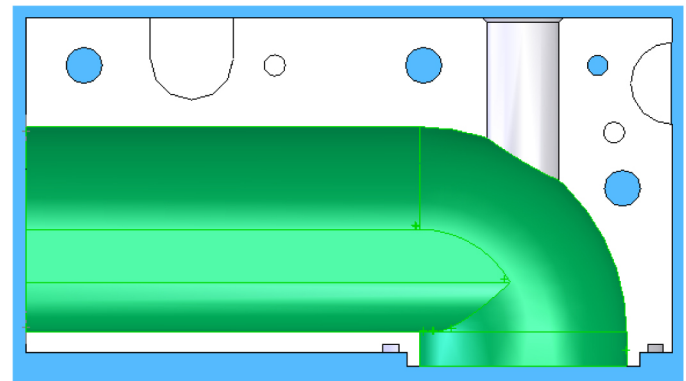
Figure 1. Diagram of optically accessible IC engine.

Both regular (non-shrouded) and shrouded intake valves were used. Throughout this paper the non-shrouded valve will be abbreviated as NV, and the shrouded valve as SV. The shrouded valve featured a 180-degree cylindrical shroud. The intake and exhaust lift profiles were identical. A single camshaft drives two identical rocker arms that act on the valves. Relevant dimensions of the engine are detailed in Table 1.

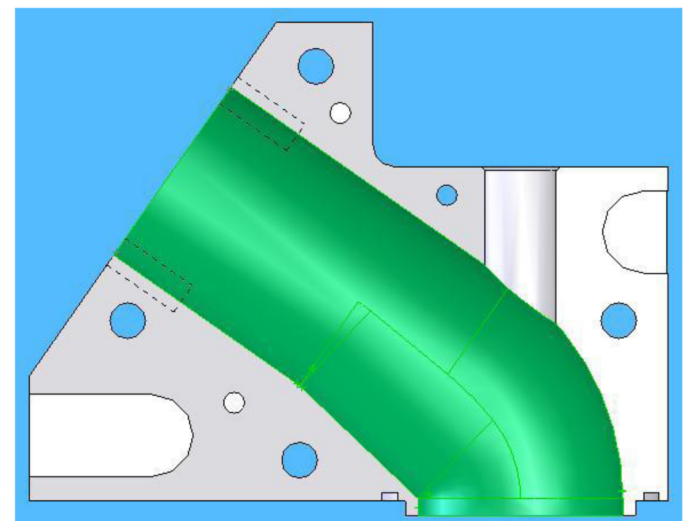
Two modular intake ports, designed to be similar in geometry to that used by small engine manufacturers, were investigated. Figure 2 shows the solid model cross-section for the two ports, termed the utility (Fig. 2a) and performance (Fig. 2b) ports, referred to hereafter as UP and PP, respectively. The intake port can be rotated about the valve axis to generate either a swirl- or tumble-dominated flow. Figure 3 illustrates the two intake port orientations tested for this study. The shrouded valve was used only with the 0-degree orientation while the non-shrouded valve was used with both 0- and 90-degree port orientations.

Table 1. Engine geometry, TDC refers to end of compression.

Compression Ratio	10.0
Bore	82.0 mm
Stroke	76.0 mm
Connecting Rod Length	144.8 mm
Intake Valve Diameter	35.0 mm
Exhaust Valve Diameter	28.0 mm
Intake Valve Open	420 bTDC
Exhaust Valve Open	90 aTDC
Intake Valve Close	90 bTDC
Exhaust Valve Close	420 aTDC
Intake and Exhaust Valve Maximum Lift	7.9 mm

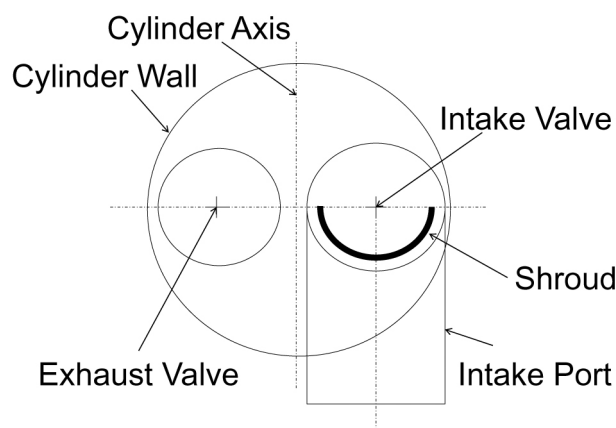


(a)

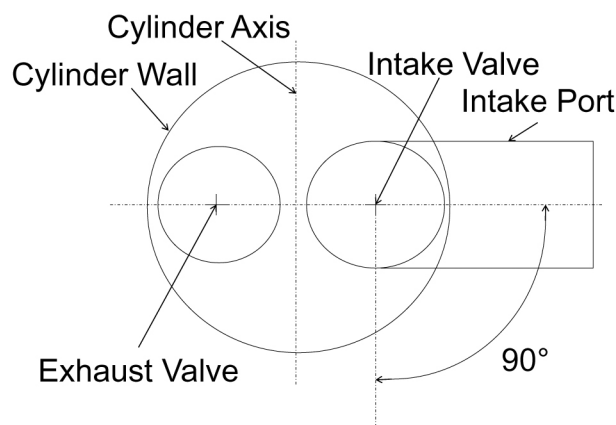


(b)

Figure 2. Solid model of (a) utility port (UP) and (b) performance port (PP).



(a)



(b)

Figure 3. Top view of engine showing (a) 0-degree and (b) 90-degree port orientations with respect to engine cylinder.

OPTICAL SYSTEM

The second harmonic (532 nm) of an Ekspla model NL303D dual Nd:YAG laser was used to generate the laser excitation for the PIV experiments. The two beams were made collinear and directed towards the optical engine. The pulse energy of each laser beam was held at 230 mJ, and a laser sheet was generated using a 50 mm focal length cylindrical lens and a 500 mm focal length spherical lens separated by 20 mm. The laser sheet was located 4.6 mm below the cylinder head deck. A Princeton Instruments MicroMAX interline transfer CCD camera was used to acquire particle images. A large field-of-view (FOV) was imaged using a 105 mm f/2.8 AF Micro Nikkor lens, and a high-magnification FOV was imaged using a 200 mm f/4 Micro Nikkor lens. The camera has a 1300×1030 pixel CCD array with 6.7 μm square pixels. The

read-out rate of the camera was such that at 600 rpm an image pair was acquired every other engine cycle.

Olive oil droplets were generated using a TSI 9306A six jet atomizer. The particles were introduced into a surge tank upstream of the intake runner to ensure sufficient mixing. The seeding density was adjusted to allow at least ten particle pairs for the smallest PIV interrogation window [37] of 32×32 pixels. The temporal separation of laser pulses ranged from 10 to 160 μs and was set such that the maximum in-plane particle displacement would not exceed a quarter of the interrogation window. For the large FOV, an image magnification of 0.15 was obtained and a FOV of 57.4 mm by 45.3 mm was achieved. For the high-magnification FOV, an image magnification of 0.49 was obtained and a FOV of 17.5 mm by 14.0 mm was achieved.

The PIV images were analyzed using the TSI Insight3G software. The software calculated velocity vectors by utilizing a multiple-pass algorithm with 50% overlap and an FFT correlation engine. Velocity vectors were validated by comparing the ratio of the two highest correlation peaks to a set threshold value. Vectors passing the test are termed first-choice vectors, while vectors failing the test were replaced by a median vector of its nearest neighbors. The initial interrogation grid was 64×64 pixels and the final interrogation grid was 32×32 pixels, which resulted in velocity vectors spaced on a 16 pixel grid. Velocity vectors were thus spaced 708 μm for the large FOV and 217 μm for the high-magnification FOV, corresponding to interrogation zones 1.42 and 0.43 mm, respectively.

STEADY FLOW CHARACTERIZATION

Flow rate and swirl data were measured for all intake port and valve combinations using a SuperFlow 600 flow bench at the industry-standard pressure drop of 6.97 kPa across the test section. The density of the air was calculated from temperature and humidity data acquired using a Mannix model J411-TH Digital Hygro Thermometer, and the barometric pressure was measured using a Heise model CM dial pressure gauge.

A swirl adapter fixture, having a height-to-bore ratio of 1.5, was installed between the cylinder head and the flow bench for flow rate testing and between the cylinder head and swirl meter for swirl testing. The intake valve lift was adjusted using a modified micrometer that mounted to the engine head. Intake horns, having radii of curvature large enough to minimize the pressure drop at the inlet to the intake ports, were connected to the entrance of the intake ports.

An impulse-type swirl meter was used to measure swirl. The meter uses a Transducer Techniques RTS-5 torque sensor. A tubular polycarbonate honeycomb flow straightener with cell

size 6.4 mm and height 9.1 mm was secured to the torque sensor with a metal shaft. A Daytronic model 3270 strain gage conditioner/indicator provided the excitation voltage for the torque sensor and a Lab VIEW data acquisition system recorded the instantaneous voltage at a rate of 10Hz. Data were collected and averaged over a 30 second period to obtain the mean voltage (torque).

The flow parameters used to characterize the engine heads are the flow coefficient, C_f , the swirl coefficient, C_s , and the swirl ratio, R_s . The flow and swirl coefficients are measured at discrete valve lifts. The flow coefficient is a measure of the actual mass flow rate to a theoretical mass flow rate and is defined as

$$C_f = \frac{\dot{m}}{\rho V_B A_v} \quad (1)$$

where $\dot{m}$ is the air mass flow rate, ρ is the air density, A_v is the valve inner seat area, and V_B is the Bernoulli velocity given by

$$V_B = \sqrt{\frac{2\Delta P}{\rho}} \quad (2)$$

where ΔP is the pressure drop across the test section. The flow coefficients are plotted against the engine crank angle degrees in Fig. 4. Comparing the ports with the non-shrouded valve, the performance port achieves higher flow coefficients at the crank angles corresponding to the range of highest valve lift. This result was expected since the performance port was designed to enhance the flow coefficient in this range. The flow coefficients of the ports with the shrouded valve compared to the non-shrouded valve are less since the flow path is partially restricted by the shroud. A mass-weighted average flow coefficient, $C_{f,avg}$, was obtained by performing a mass-weighted average over the engine crank angle profile, defined as

$$C_{f,avg} = \frac{\sum_{\theta} C_f \dot{m}}{\sum_{\theta} \dot{m}} \quad (3)$$

The swirl coefficient, C_s , is a characteristic non-dimensional rotation rate and is calculated for impulse-type meters using

$$C_s = \frac{8T}{\dot{m} V_B B} \quad (4)$$

where B is the cylinder bore and T is the torque measured by the meter. Figure 5 gives the swirl coefficients plotted against the non-dimensional valve lift, l/D , where l is the valve lift and D is the valve inner seat diameter. The shrouded valve directionally restricts the flow into the cylinder, tending to enhance the swirl compared to the non-shrouded valve, resulting in greater swirl coefficients. The swirl ratio, R_s , is a convenient single metric that takes into account the flow and swirl coefficients over the entire lift profile of the engine. The swirl ratio is calculated as

$$R_s = \frac{\pi \eta_v^2 B S}{4 A_v} \frac{\int_{\theta_{IVO}}^{\theta_{IVC}} C_f C_s d\theta}{\left(\int_{\theta_{IVO}}^{\theta_{IVC}} C_f d\theta \right)^2} \quad (5)$$

where η_v is the volumetric efficiency (assumed equal to 1 for all calculations), θ_{IVO} and θ_{IVC} are the crank angle, in radians, at intake valve open and intake valve close, respectively, and S is the engine stroke.

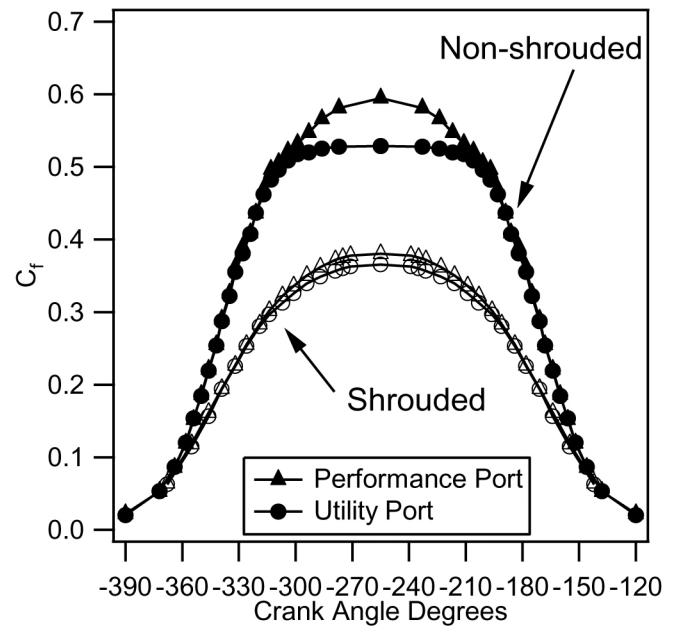


Figure 4. Flow coefficient for different port geometry and valve type at the 0-degree orientation. 0 crank angle degrees refers to TDC at end of compression.

The mass-weighted average flow coefficients and swirl ratios are given in Table 2. A two-sided, 95% uncertainty, u_{95} , was calculated for the flow and swirl coefficients at four separate valve lifts by performing a repeatability test with a sample size of 11 trials. The uncertainty for the flow coefficients ranged from 0.001 to 0.008 and for the swirl coefficients

Table 2. Steady flow characterization results. Note: -CCW swirl, +CW swirl when viewed from top of engine.

Valve	Port	Orientation	$C_{f,avg} \pm u_{95}$	$R_s \pm u_{95}$
Shrouded	Utility	0-degree	0.293 ± 0.001	-3.214 ± 0.032
	Performance	0-degree	0.303 ± 0.001	-3.058 ± 0.030
Non-shrouded	Utility	0-degree	0.446 ± 0.001	-0.251 ± 0.009
		90-degree	0.430 ± 0.001	0.128 ± 0.009
	Performance	0-degree	0.481 ± 0.003	-0.234 ± 0.008
		90-degree	0.499 ± 0.003	0.121 ± 0.008

ranged from 0.009 to 0.014. The uncertainty for the mass-weighted average flow coefficients and swirl ratios was then found using the uncertainty propagation method from [40]. The swirl ratios of the ports in the 0-degree orientation with the shrouded valve are roughly 13 times greater than with the non-shrouded valve.

the compression stroke during different cycles. For each condition, 50 cycles were acquired. Figure 6(a) contains the ensemble average and Fig. 6(c) & (d) contain two individual cycles for a given condition. The ensemble average was calculated by averaging the velocity vectors at each location (x,y) as

$$\bar{U}_{E,i}(x,y) = \frac{1}{N_c} \sum_{i=1}^{N_c} U_i(x,y), \quad i=1,2 \quad (6)$$

where U_i is the instantaneous velocity, N_c is the number of cycles, and $i=1,2$ refers to the components of the velocity in the x - and y -directions, respectively. Figure 6(a) contains a white box indicating the relative location of the high-magnification FOV and a purple box indicating the bounds of the swirl center figures in the upcoming section. The percentage of first-choice vectors for all conditions ranged from 78-95%.

The ports in the 0-degree orientation (see Fig. 3) resulted in a bulk swirling motion in the plane of measurement. The data acquired with the ports in the 0-degree orientation were analyzed to determine the location of the swirl center and rate of angular rotation. The method used to calculate the swirl center and angular velocity is as follows. The ensemble average velocity field of the PIV data were calculated for a given condition over 50 cycles. An algorithm was developed that assumed a solid body rotation and calculated the location and angular velocity magnitude, Ω , that minimized the sum squared error of the velocity difference between the solid body and ensemble average velocity fields. The average sum squared error per velocity vector for a given ensemble average velocity field ranged from 0.12 to 6.63 m^2/s^2 with a median of 0.47 m^2/s^2 . Figure 6(b) shows the solid body velocity field calculated for Fig. 6(a), where the calculated swirl center is shown as a yellow cross in Fig. 6(a). As can be seen, the yellow cross is not exactly on the swirl center in Fig. 6(a). This is attributed to the non-solid body rotation of

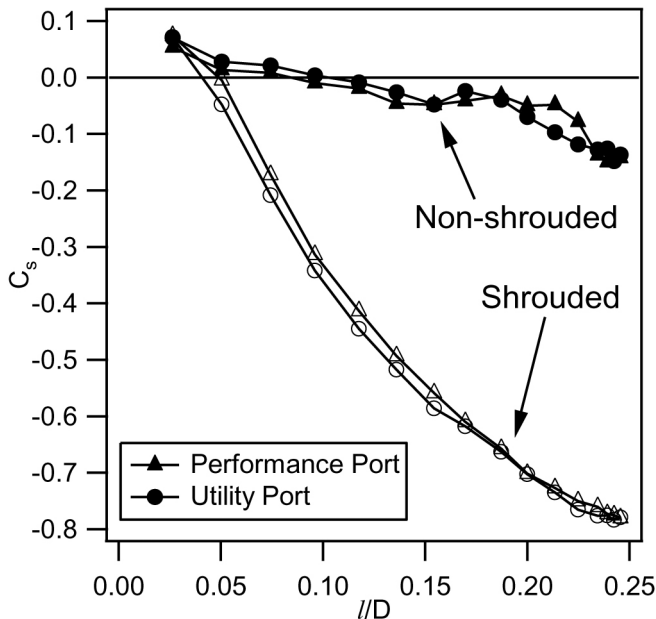


Figure 5. Swirl coefficient for different port geometry and valve type at the 0-degree orientation.

RESULTS

LOW MAGNIFICATION PIV RESULTS

The large FOV (low magnification) PIV data were acquired to observe the bulk in-cylinder fluid motion. The camera magnification was chosen so that the entire visible cross-section area of the cylinder could be imaged. Images were acquired at crank angles of 90 bTDC, 45 bTDC, and TDC of

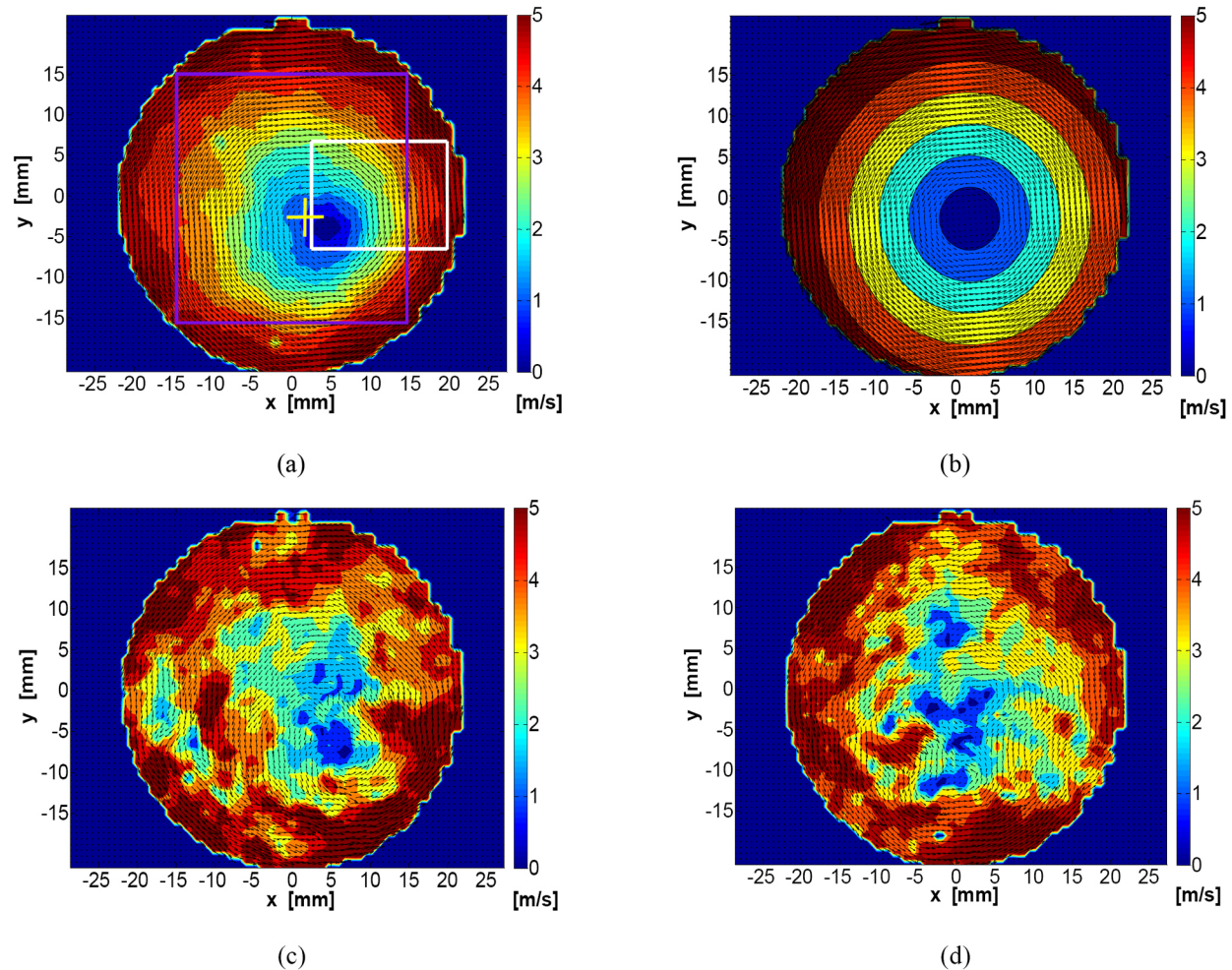
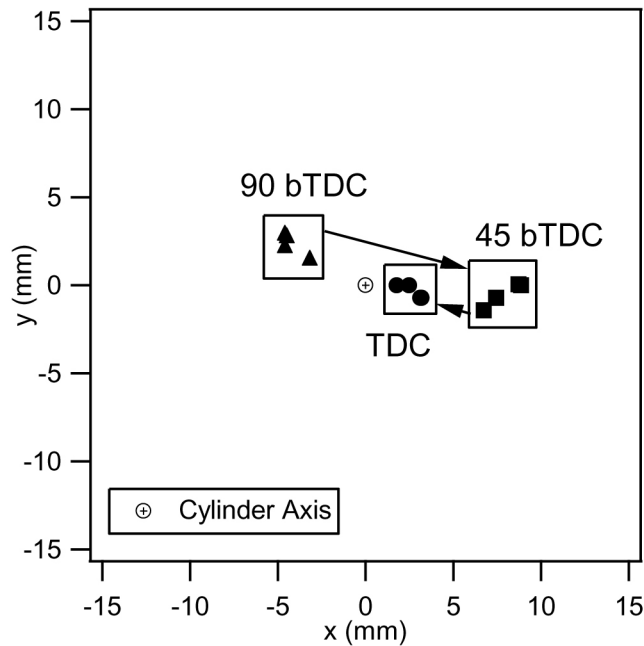


Figure 6. Large FOV velocity fields at TDC for the utility port with the shrouded valve at 600 rpm. (a) Ensemble average with white box indicating high-magnification FOV, (b) calculated solid body, and (c)-(d) two randomly chosen instantaneous velocity fields.

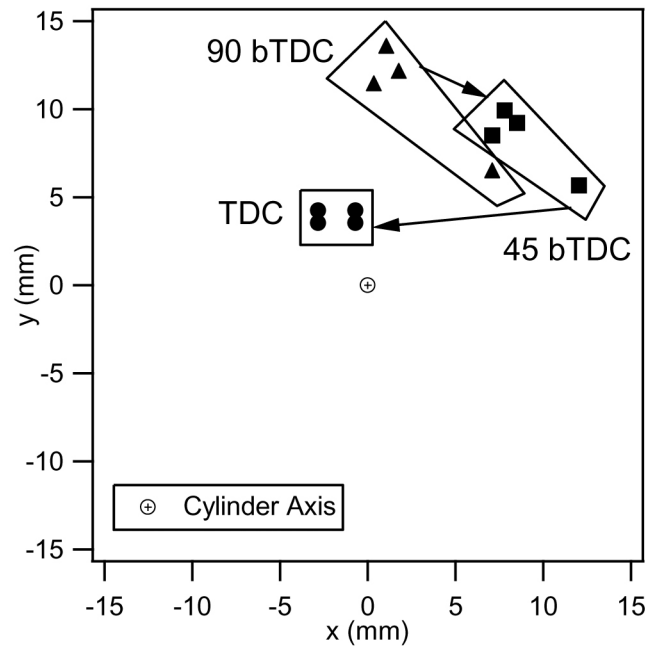
the ensemble average velocity field (in fact, the instantaneous velocity fields, Figs. 6(c)-(d), have no well defined swirl center). The algorithm, which tries to fit a solid body rotation, gives a good estimate of the angular velocity of the ensemble average velocity field. While the swirl center may not exactly match the ensemble average velocity image, this method is more descriptive of the flow field as a whole.

Figure 7 shows the swirl center locations for both ports with each valve. The location of the swirl centers at a given crank angle did not vary much over the range of engine speeds, therefore, Fig. 7 omits individual labels of the engine speed and for clarity boxes the data at a given crank angle. The swirl center locations of the ports with the shrouded valve tend to be better grouped together at each crank angle compared to the data with the non-shrouded valve. Also, the swirl centers at TDC for all port configurations are located nearest to the cylinder axis. Figure 8 shows the flow angular velocity normalized by the engine rotation rate at each crank

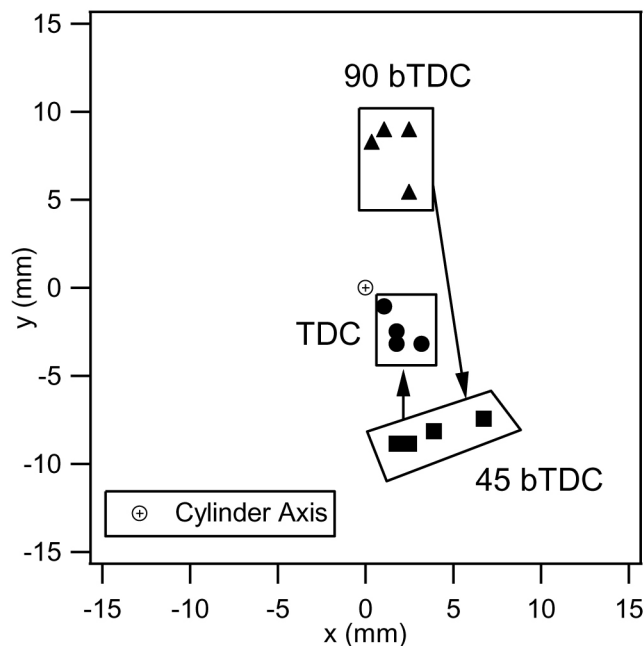
angle. Normalizing the data by the engine rotation rate (RPM) nearly collapses the four speeds onto one curve. The shrouded valve for either port results in about a fourfold increase in angular velocity over the non-shrouded valve. A decreasing trend can be seen for the angular velocity as the piston approaches TDC for all conditions, which is attributable to viscous losses at the wall. Figure 9 plots the normalized angular velocity at TDC against the swirl ratios found from the steady-flow testing (see Table 2). The normalized angular velocity of the ports with the shrouded valve are on average 19% higher than the swirl ratio values while the ports with the non-shrouded valve are about three times higher than predicted by the steady swirl ratio.



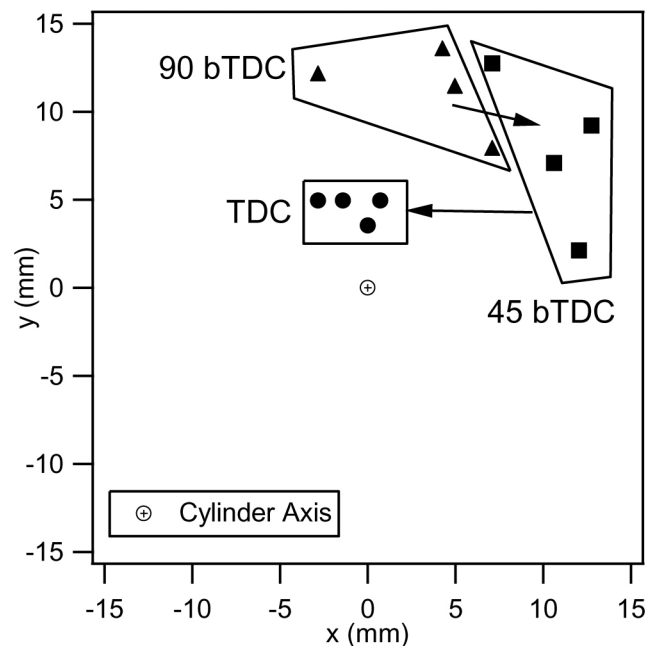
(a)



(b)



(c)



(d)

Figure 7. Swirl center locations at 90 bTDC, 45 bTDC, and TDC and at 300, 600, 900, and 1200 rpm, ports in 0-degree orientation. Performance port: (a) shrouded valve, (b) non-shrouded valve. Utility port: (c) shrouded valve, (d) non-shrouded valve.

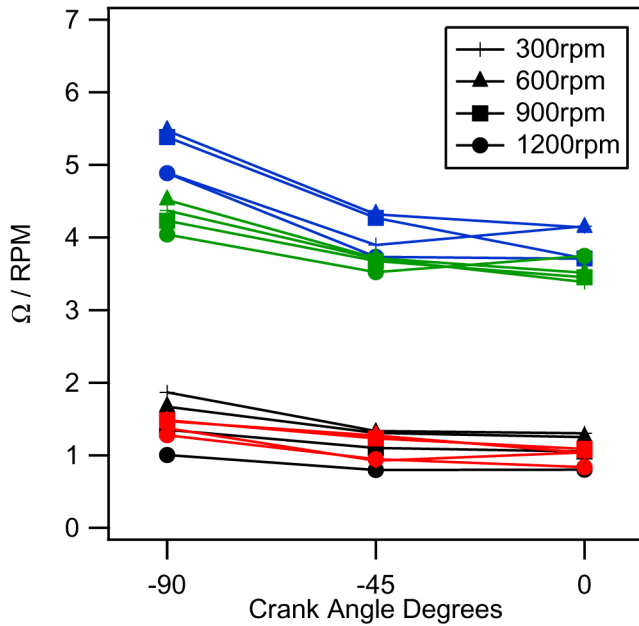


Figure 8. Normalized angular velocity vs. crank angle, ports in 0-degree orientation. Red: performance port, non-shrouded valve. Black: utility port, non-shrouded valve. Green: performance port, shrouded valve. Blue: utility port, shrouded valve.

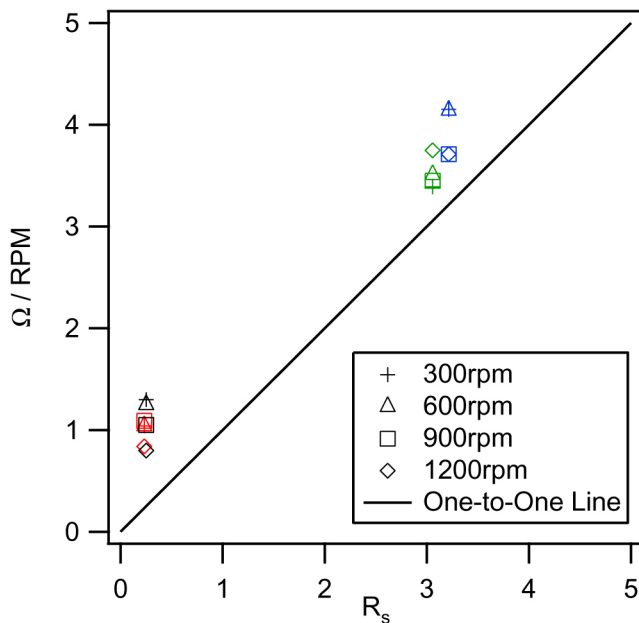


Figure 9. Normalized angular velocity at TDC vs. swirl ratio, ports in 0-degree orientation. Red: performance port, non-shrouded valve. Black: utility port, non-shrouded valve. Green: performance port, shrouded valve. Blue: utility port, shrouded valve.

HIGH MAGNIFICATION PIV RESULTS

Turbulence Intensity

The high-magnification FOV PIV data were acquired to observe and calculate the small-scale fluid turbulence. The location of the high-magnification FOV relative to the large FOV was shown in Fig. 6(a). Images were acquired only at TDC of the compression stroke. For each condition, a set of 200 cycles were captured. The percentage of first-choice vectors for all conditions ranged from 89-99%. Data were analyzed for all port orientations and engine running conditions.

Two methods were used to calculate the *mean* velocity field. As mentioned previously, one of the challenges of data analysis in engine flows is to define an appropriate mean velocity. Therefore, it was sought to perform the analysis using the methods appropriate for the spatially resolved, multiple-realization data available. In the first method, an ensemble average velocity field was calculated for each engine condition over the 200 cycles, see Eq. 6. In the second method, a spatial-average velocity field was calculated based on a cutoff lengthscale for each individual cycle. The analysis computes the average velocity in the *x*- and *y*-directions separately. A two-dimensional fast-Fourier transform (FFT) was performed on the velocity field. The two-dimensional spectrum was multiplied by a Fermi-Dirac soft cutoff filter [25], which was calculated as

$$\frac{1}{1 + \exp((\kappa_{xy} - \kappa_c)/(0.1\kappa_c))} \quad (7)$$

where κ_{xy} is the two-dimensional wavenumber defined as

$$\kappa_{xy} = \sqrt{\kappa_x^2 + \kappa_y^2} \quad (8)$$

and κ_c is the cutoff wavenumber defined as

$$\kappa_c = \frac{2\pi}{L_c} \quad (9)$$