

Characterization of Flow Asymmetry During the Compression Stroke Using Swirl-Plane PIV in a Light-Duty Optical Diesel Engine with the Re-entrant Piston Bowl Geometry

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ABSTRACT

Flow field asymmetry can lead to an asymmetric mixture preparation in Diesel engines. To understand the evolution of this asymmetry, it is necessary to characterize the in-cylinder flow over the full compression stroke. Moreover, since bowl-in-piston cylinder geometries can substantially impact the in-cylinder flow, characterization of these flows requires the use of geometrically correct pistons. In this work, the flow has been visualized via a transparent piston top with a realistic bowl geometry, which causes severe experimental difficulties due to the spatial and temporal variation of the optical distortion. An advanced optical distortion correction method is described to allow reliable particle image velocimetry (PIV) measurements through the full compression stroke.

Based on the ensemble-averaged velocity results, flow asymmetry characterized by the swirl center offset and the associated tilting of the vortex axis is quantified. The observed vertical tilting of swirl center axis is similar for tested swirl ratios (2.2 and 3.5), indicating that the details of the intake flows are not of primary importance to the late-compression mean flow asymmetry. Instead, the geometry of the piston pip likely impacts the flow asymmetry.

The PIV results also confirm the numerically simulated flow asymmetry in the early and late compression stroke: at BDC, the swirl center is located closer to the exhaust valves for swirl-planes farther away from the fire deck; near TDC, the swirl center is located closer to the intake valves for swirl-planes farther away from the fire deck. It is evident from experimentally determined velocity fields that the transition between these two asymmetries has a different path for various swirl ratios, suggesting the influence of intake port flows.

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INTRODUCTION

In direct-injection Diesel engines, proper mixing of the injected fuel and available oxygen within a short amount of time is of great importance to achieve high fuel efficiency and low emissions. Fuel-air mixing plays an essential role in the combustion and pollutant formation processes [1, 2], and is strongly influenced by the in-cylinder flow field, particularly when the injected fuel quantity is small.

Particle image velocimetry (PIV) is an optical diagnostic technique for velocity measurements. PIV measurements conducted inside optical engines [3, 4, 5, 6, 7, 8, 9] and constant volume vessels [10, 11] have provided significant insights about transport of pollutants [5, 12] and growth of boundary layer phenomena [13, 14]. Insights gained through PIV measurements have also been useful for the

development of conceptual models for transient entrainment rates [15, 16]. In order to provide practical guidelines to improve engine design, more efforts are needed to understand the physics governing the evolution of the observed in-cylinder flow fields and correlate them with engine design parameters.

In swirl-supported Diesel engines, a solid-body-like rotational flow (swirl [17]) around the cylinder axis is generated during the intake and compression strokes. Swirl is designed to promote large-scale transport of fuel vapor and air during injection and also enhance the late-cycle mixing. Furthermore, mean flow swirl center precession feeds the generation of turbulent kinetic energy required to bring about small-scale mixing [1]. The swirl flow in a conventional four-valve head Diesel engine is often achieved with two different intake port geometries: one tangentially oriented ports designed to direct the intake flow with a strong tangential velocity component;

and the other helical-shaped port is designed to create an intake flow that is more spatially uniform with minimal induction losses. In the variable swirl design, each intake port, near the entrance of the cylinder head, is fitted with a butterfly valve with a number of calibrated throttling positions. The swirl magnitudes are set by adjusting the positions of the valves. The calibration of the swirl magnitudes is based on global swirl measurements taken on steady-state flow benches which do not fully incorporate the complex flow processes present in an operating engine. These processes are also impacted by the piston bowl geometry, piston motion and the wave dynamics in the intake manifold. Moreover, the steady-state swirl calibration does not include information on the spatial structure of the flow, which is also important for accurate simulation prediction.

Previous studies have employed PIV and other full-field techniques inside internal combustion engines to capture the flow structures in greater detail. Reuss [4] compared ensemble-averaged velocity distributions with three dimensional CFD simulation results provided by GMTEC (an in-house code) and KIVA; excessive angular mean-momentum dissipation of the model prediction was observed. Nordgren [18] compared PIV results with AVL FIRE CFD to investigate swirl ratio evolution and swirl center precession. Deslandes [19] compared PIV results with two piston geometries: flat and bowl-in-piston, in order to investigate the geometry effect on swirl center location, swirl intensity distribution, and turbulent kinetic energy. Cosadia [6] coupled time-resolved PIV with phase-invariant POD decomposition to analyze cyclic flow variability. For all of these projects, pistons with either flat [4, 18, 19] or simplified geometries [6, 20] were employed to minimize optical distortion. However, as described by Miles [1], both measurements and numerical simulation confirm that the different piston bowl geometries result in different near-TDC turbulence structures. Therefore, it is desirable to use a production-like, re-entrant piston bowl geometry to achieve the realistic boundary conditions for in-cylinder flow studies.

Early single point Laser Doppler Velocimetry (LDV) measurements found that the only mean flow structures surviving well into the compression stroke are swirl and tumble [17]. In Diesel engines, swirl is of more importance than tumble. The position of swirl center often rotates around the cylinder axis. The bowl-in-piston geometry usually promotes the centering of swirl and reduces its spatial variation during the compression stroke. The radial distribution of the swirl velocity exhibits solid-body-like structure except near the outer radii, where a flattened radial profile is present. Planar PIV techniques have expanded the measurement spatial resolution with respect to the single point LDV technique. However, as the compensation for the optical distortion in the swirl-plane is more challenging than that in the tumble (vertical) plane, planar PIV has not been applied in engines with realistic bowl geometries [5, 21, 22, 23, 24] until the recent work done by Petersen [25].

The dewarping methodology applied by Petersen [25] was based on frames of a calibration target with uniformly-spacing points that was placed in the measuring plane; distorted target frames were obtained through the piston bottom at various crank angle positions. By registering the distorted image points to the undistorted physical coordinates, piece-wise linear transformations were obtained to correct the optical distortions generated by the curved bowl surfaces. With

swirl-plane PIV conducted from -50°aTDC until -25°aTDC for three different steady-state swirl ratios (2.2, 3.5 and 4.5), results obtained confirmed the swirl velocity radial distribution measured by LDV, and great similarity of the flow asymmetry has been found near the firing TDC with various swirl ratios: swirl centers tend to be located closer to the intake side for swirl-planes located farther away from the fire deck. This asymmetry has been confirmed with numerical simulation as well [16]. Interestingly, the simulated streamlines at BDC indicate that swirl centers are located closer to the exhaust side in the swirl-planes farther away from the fire deck. The simulated swirl asymmetry at BDC is logically expected from the intake port orientation. In order to demonstrate that the expected asymmetry at BDC is real, and that the inclination of the swirl axis changes between BDC to the late compression stroke, the swirl-plane flow fields need to be characterized throughout the full compression stroke.

For swirl-plane PIV, reliable velocity measurements are hindered by optical distortions induced by the piston geometry, particularly when the distance from the piston to the measurement plane is large; i.e., when the piston is close to BDC. Artificial radial velocities resulting from the time-changing optical distortion, severe distortion, and limited spatial resolution must be accounted for and resolved. With the swirl-plane PIV setup optimized for these experiments and a refined velocity re-construction method for the time-changing optical distortion correction, the bore-scale flow topology metrics such as swirl center motion and swirl center axis tilting can be analyzed throughout the full compression stroke.

The current study is an extension of Petersen's previous swirl-plane PIV investigations to cover the full compression stroke. The main motivation is to complete a full mapping of swirl azimuthal asymmetry for the purpose of validating CFD predictions and defining best practices for engine CFD. The experimental assessment of flow asymmetries is also expected to help deciding the optimum injector location, so as to minimize the asymmetric mixture preparation inside light-duty, swirl-supported Diesel engines.

EXPERIMENTAL APPARATUS AND OPTICAL DIAGNOSTICS

Engine and Operating Conditions

The single-cylinder optical research engine used in this study is built with the cylinder head from a General Motors four cylinder, 1.9 L light-duty engine. The intake ports (one helical and one tangential) are each fitted with swirl control vanes that allow the in-cylinder swirl ratio to be varied between 1.5 and 4.5. The engine specifications are listed in [Table 1](#).

Optical access is provided by a classic Bowditch piston extension with a fused-silica piston top. The re-entrant bowl and valve cut-outs are geometrically accurate representations of a production-intent piston design. The placement of a mirror inside the piston assembly at 45° relative to the line-of-sight provides a view of nearly the entire 82 mm-diameter bore. The cylinder liner is composed of two parts: the lower liner has a drop-down design which facilitates optical alignment and cleaning when the engine is not in operation, the upper liner is stationary, and is sealed against the fire deck and the lower liner with

O-rings on both surfaces while the engine is in operation. Fused-silica windows (50.7 mm × 25 mm) in the upper cylinder liner provide laser and imaging access all the way to the fire deck (Figure 1).

Table 1. Engine Geometry and Valve Timings

Bore / Stroke	82.0 / 90.4 [mm]
Displacement Volume	0.477 [cm ³]
Cycle	4-stroke
CR (Geometric)	16.7
Squish Height	0.78 [mm]
Intake/Exhaust Valves	2/2
Intake Valve Open (IVO)	-359 [°CA]
Intake Valve Close (IVC)	-152 [°CA]
Exhaust Valve Open (EVO)	132 [°CA]
Exhaust Valve Close (EVC)	360 [°CA]

0°CA are referenced to the firing TDC.

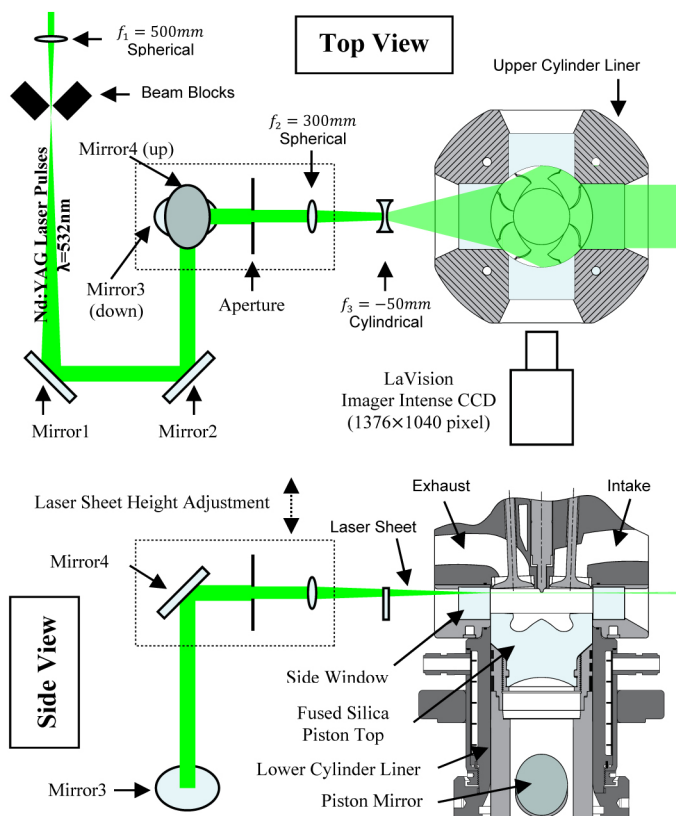


Figure 1. Schematic of the optical setup (not to scale) for swirl-plane PIV measurements with slit imaging approach. The laser sheet is parallel to the piston top, and is shown 3mm below the fire deck. The piston position shown is -50°aTDC.

All the PIV data are acquired while the engine is motored with no fuel injection. Two steady-state swirl ratios of 2.2 and 3.5 are tested in this study. The port throttling configuration is calibrated from the previous steady-state flow bench measurements [26, 27]. The detailed specifications of the motored operating condition are shown in Table 2. This low speed (1500 RPM), slightly boosted (1.5 bar intake pressure), highly diluted (10% O₂ mole fraction) operation is usually used in a low-load, partially premixed compression ignition (PPCI)

condition studied in this group. With the constant total mass flow rate of 8.936 g/s, the cycle-averaged motored in-cylinder pressure trace exhibits good repeatability for two swirl ratios in Figure 2.

Table 2. Engine Operating Conditions

Engine Speed	1500 [RPM]
Intake Pressure	1.5 [bar]
Intake Temperature	99 [°C]
Steady-State Swirl Ratio ($R_{s,steady}$)	2.2 3.5
Oxygen Mole Fraction	10%
Air Mass Fraction / Mass Flow Rate	0.46 / 4.113 [g/s]
N ₂ Mass Fraction / Mass Flow Rate	0.40 / 3.538 [g/s]
CO ₂ Mass Fraction / Mass Flow Rate	0.13 / 1.185 [g/s]
H ₂ O Mass Fraction / Mass Flow Rate	0.01 / 0.1 [g/s]

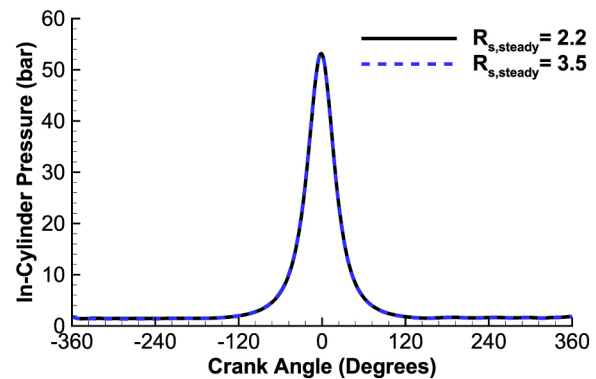


Figure 2. Cycle-averaged motored in-cylinder pressure trace under two steady-state swirl ratios ($R_{s,steady}=2.2$ and 3.5).

Particle Image Velocimetry

Figure 1 shows the experimental setup for the planar PIV imaging. This section is focused on the three most important aspects of the PIV experiments: laser sheet formation, particle seeding, and imaging configuration.

Laser Sheet Formation

The laser used in this experiment is a dual-head pulsed Nd:YAG laser system (Big Sky Ultra CFR 200). The second harmonic output (532 nm) is formed into a sheet and directed into the combustion chamber with four high reflectivity laser line mirrors. For planar PIV experiments, it is very important to have a thin laser sheet so as to minimize the perspective error [28]. However, generating a uniformly thin laser sheet with enough fluence in the 82 × 82 mm field of view is difficult. The laser “thin wedge” formed by the common lens combination (equivalent to one cylindrical concave lens and one spherical convex lens) may achieve a very small laser waist region, but this “wedge” will have a much thicker cross section before and after the minimum waist region. This is due to the relatively large laser beam quality factor ($M^2 \approx 25$). Therefore, a “slit imaging” approach is implemented for this experiment. As shown in Figure 1, a 1” diameter spherical convex lens ($f_1 = 500$ mm) is used to focus the beam at the laser exit. A slit composed of two beam blocks is positioned 400 mm away from the convex lens. The 2 mm wide slit is imaged into the combustion chamber center through a 1” diameter spherical convex lens ($f_2 = 300$ mm) with a magnification ratio of approximately 1/8. Since the negative focal length of the entrance

side window is not short enough to horizontally expand the slit image to illuminate the whole combustion chamber, a cylindrical biconcave lens ($f_3 = -50$ mm) is positioned 110 mm from the imaging convex lens ($f_2 = 300$ mm) to expand the laser horizontally before it encounters the entrance window. All the vertical surfaces inside the chamber (except the entrance and the exit windows) are air-brush painted with absorptive black paint so as to decrease background scattering. With the "slit imaging" approach, the minimum sheet thickness in the chamber center is around 450 μm and the thickness increases to around 800 μm at 40 mm away from the center. The flash lamp and Q-switch are externally triggered from the LaVision timing and synchronization unit (PTU) which is synchronized with engine speed at 1500 rpm. The laser is fired once every 5 cycles at a rate of 2.5 Hz. The Q-Switch delay from the flash lamp trigger is tuned to achieve similar energy for both laser pulses. With the laser operating in full power mode, the energy reaching the chamber is approximately 20 mJ/pulse.

Particle Tracer Fidelity Check and Seeding Process

The tracer used in this experiment is porous SiO_2 powder (Seika Corporation, B-6C) with a true density (ρ_p) of 600 kg/m^3 and an average particle diameter (d_p) of 2.0 μm . The time constant of this particle is calculated assuming Stokes drag:

$$\tau = \frac{\rho_p d_p^2}{18\nu\rho_f} \quad (1)$$

Since the mass flow rate is kept constant during engine operation (Table 2) and the valve overlap of this engine is very small (Table 1), the charge density (ρ_f) can be estimated with crank-angle-resolved chamber volume. The charge kinematic viscosity (ν) is calculated according to Sutherland's formula:

$$\nu = \nu_0 \left(\frac{T_0 + C}{T + C} \right) \left(\frac{T}{T_0} \right)^{3/2} \quad (2)$$

$T_0 = 273.15$ K, ν_0 is the reference kinematic viscosity at temperature T_0 . C is Sutherland's constant [29]. The charge temperature (T) during the compression stroke after IVC is calculated with an isentropic relation, and the temperature dependent heat capacity ratio γ is computed for the mixture composition given in Table 2.

The ratio of particle density to charge density ($\bar{\rho} = \frac{\rho_p}{\rho_f}$) ranges from 25 to 426 throughout the whole engine cycle, and the particle frequency response function [30] can be approximated as a simple first-order low-pass filter with time constant τ [31]. Therefore, the cutoff frequency (f_{cutoff}) is:

$$f_{\text{cutoff}} = \frac{1}{2\pi\tau} \quad (3)$$

In order to perform the particle tracer fidelity check, a representative mean flow time scale must be obtained. The time scale of this flow is determined using a commercial CFD software package

CONVERGE™ [32] which solves the Reynolds-Averaged Navier-Stokes (RANS) equations. Turbulence is accounted by means of the ReNormalization Group (RNG) closure for the two equation $k-\epsilon$ model. A second-order-accurate spatial discretization scheme and a fully-implicit, first-order-accurate time integration scheme is used for the governing conservation equations. The largest computational grid size is 0.004 m, the smallest is 0.0005 m, and the total cell count is about 1 million. A pdf of $\sqrt{S_{ij}S_{ij}}$ constructed from the values of all the cells in the bulk of the cylinder (with distances greater than 1 mm from the walls) is generated every 5 CAD from -360°aTDC until 120°aTDC . The following figure shows that the particle cutoff frequency (f_{cutoff}) during the compression stroke is at least 50 times higher than the volume-averaged maximum strain rate ϵ , and the Stokes number $S_k = \epsilon/f_{\text{cutoff}} < 0.1$ from -280°aTDC through the full compression stroke. Therefore, the particles used in this study yield in an acceptable flow tracing accuracy with errors below 1% [33, 34].

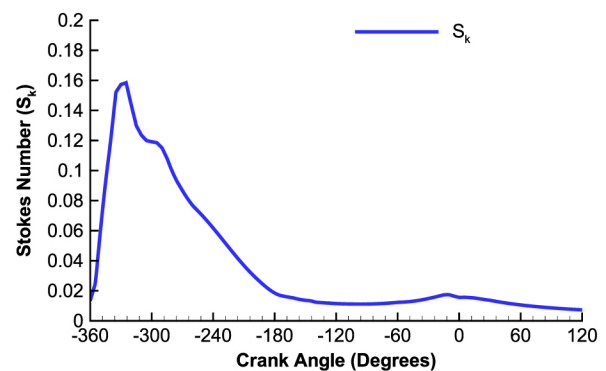


Figure 3. Particle Tracer Fidelity Check.

Bronze beads (TSI, $d = 100 \mu\text{m}$) are used to mix with dry tracer particles so as to avoid particle agglomeration. The mixture is placed in a modified TSI Model 3400 fluidized bed seeder fed with a 0.5 g/s flow of nitrogen. The tracer particles are fed into the engine intake upstream of the surge tank. Water vapor with a flow rate of 0.1 g/s is added to the intake stream (in the proportion typically found in EGR gases), but upstream of the particles feeding location in order to help dissipate the static charges on the particles caused by the collision with bronze beads. This procedure has been found to prevent particles from sticking to the windows and piston and thus degrading the image quality. Since the 2 μm porous SiO_2 powder is very hard, to avoid excessive liner wear and high motoring torque requirements, the engine is stopped and cleaned between each test point in order to clean particles from the piston and the side windows. In order to keep the engine thermal condition consistent for all the test points, the experiments are arranged in an 8 minute schedule: engine is motored for 3 minutes and stopped and cleaned for 5 minutes. One minute after the engine starts, when the desired operating condition (Table 2) is stabilized, nitrogen flow is re-directed through the fluidized bed seeder and the tracer particles are fed into the engine intake stream. Considering the transient behavior of particle density (time lag for particle's presence in the chamber), the PIV sampling duration is limited to one minute in which 150 pairs of particle images within 750 engine cycles at 2.5 Hz are recorded. The engine is allowed to motor for an additional minute to purge seeding particles before it is stopped and cleaned.

Imaging Configuration

Three swirl-planes are chosen to quantify the flow asymmetry (Figure 4). For each plane, PIV measurements are performed at 15 CAD intervals starting at BDC and ending just before the laser sheet begins to intersect the top of the piston (Table 3). For each steady-state swirl ratio, crank angle position, and plane, 150 pairs of particle frames are acquired with a monochromatic CCD camera (LaVision Imager Intense, 1376×1040 pixels, 12-bit image depth) in double-frame mode. The manual-focus zoom lens (Zoom-NIKKOR 35-200mm) is adjusted to compromise the wide range of piston positions. Since the PIV frames are taken through the piston, the magnification factor of this optical setup is very small. The actual particle size has little impact on its image size, which is mainly dominated by the diffraction-limited spot diameter (a function of the magnification factor, camera aperture opening, and laser wavelength) [35]. According to the general rule of PIV design: the recommended particle image size is 2~3 times the physical pixel size [36]. Therefore, the camera aperture is set to f/11 yielding in a particle image size of 2.5 times the physical pixel size. The temporal interval between each pair of frames is mainly chosen to have in-plane and out-of-plane displacement less than the $\frac{1}{4}$ of interrogation window size (PIV $\frac{1}{4}$ rule [37]). The exposure time of the first image is 10 μ s and the exposure duration of the second image is normally constrained by the readout duration of the previous frame to 90msec. A mechanical shutter (Vincent Associates, UNIBLITZ® VS14S2T0-17) is used to limit the 2nd exposure duration to around 3ms and decrease background interference on the second image.

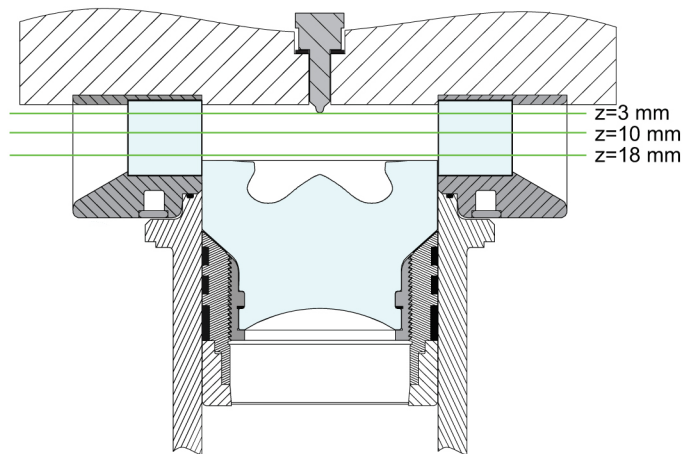


Figure 4. Illustration of swirl-plane vertical positions for PIV measurements. (the piston is shown for a crank angle of -50° aTDC).

Data Processing

Image pre-processing, optical distortion correction and post-processing are all important to compute reliable velocity fields with this re-entrant bowl geometry. This section briefly describes these three processes. The main purpose is to address their importance and present the advantages and disadvantages of the approaches used in this work.

Table 3. Test Matrix and PIV Configurations

Engine steady-state swirl ratio $R_{s,steady}$	2.2, 3.5
Swirl-plane vertical position (below fire deck) z [mm]	3, 10 and 18
Crank angle position of PIV measurement [$^\circ$ aTDC]	-180,-165,-150,-135,-120,-105,-90,-75,-60,-45,-30,-25 (z=3 mm) -180,-165,-150,-135,-120,-105,-90,-75,-60 (z=10 mm) -180,-165,-150,-135,-120,-105,-90,-75 (z=18 mm)
Particle	2 μ m porous SiO ₂ powder
Laser repetition rate [Hz]	2.5
Number of sampled cycles	150
Temporal interval between two frames (Δt) [μ sec]	5 ($R_{s,steady}=2.2$, except -180° aTDC) 7 ($R_{s,steady}=2.2$, -180° aTDC) 4 ($R_{s,steady}=3.5$, except -180° aTDC) 5 ($R_{s,steady}=3.5$, -180° aTDC)
Camera aperture opening	f/11
CCD pixel size [μ m]	6.45
Magnification	A function of radial position in measuring plane and CAD. Around 0.5 for squish zone, and 0.1-0.4 in bowl region.
Image resolution after dewarping α [μ m/pixel]	80.91 (z=3mm) 80.39 (z=10mm) 78.99 (z=18mm)
Field of view [mm^2]	82 $\times$ 82

Pre-processing: Background Subtraction

Background subtraction is very important for PIV experiments conducted in an enclosure (pressurized combustion chamber). The background subtraction for particle feature extraction purposes must account for several spatially and temporally-variable sources of interference:

1. Temporally-changing seed density over different engine cycles.
2. Spatial variations in the distribution of laser fluence in the swirl plane.
3. The difference in laser energy between two adjacent pulses and pulse-to-pulse energy fluctuations.
4. Temporally and spatially varying background intensities due to sources (2) and (3).
5. The crank angle dependent piston lensing effect and the coupling between scattered light from particles, background intensity, and the laser sheet itself.

Moreover, the piston with complex geometry exhibits worse signal degradation than the simplified geometry. As a consequence, both the measurement resolution and signal-to-noise ratio degrade. The degraded signal-to-noise ratio in this experiment will further enhance the sensitivity to the previously mentioned five interferences. To compensate, the laser pulse energy used in this test is higher than all the documented pulse energies for planar PIV in an enclosure with simplified geometry [10].

Due to the transient behavior of particle density: the presence of particles inside the combustion chamber for each test exhibits about 12 seconds delay from the enabling of the fluidized bed (the same time as start of laser) and the background signal level ramps up quickly and then slowly decays. Subtraction of a constant background image (either an image taken without seeding or the average of all PIV frames in a given measurement) produces unsatisfactory results

in cycles with background intensity levels lower than the subtracted background. In this case, particle signals are attenuated more than necessary and the signal-to-noise ratio is degraded further.

Therefore, a two-step particle feature extraction algorithm is adopted as follows:

1. For each frame, every pixel inside chamber is corrected with its local laser pulse energy ratio.
2. Calculate the adaptive background and perform subtraction.

The laser energy ratio between two pulses $\gamma_{\text{laser1}}^{\text{laser2}}$ can be approximated as:

$$\gamma_{\text{laser1}}^{\text{laser2}}(i, j) = \frac{\sum_{k=1}^{150} I_1(i, j, k)}{\sum_{k=1}^{150} I_2(i, j, k)} \quad (4)$$

Where i, j are pixel indices, k is engine cycle index. For each pair of frames, pixel counts in the two PIV image frames are scaled as follows:

If $\gamma_{\text{laser1}}^{\text{laser2}}(i, j) < 1$,

$$I_{\text{scaled1}}(i, j, k) = \frac{I_1(i, j, k)}{\gamma_{\text{laser1}}^{\text{laser2}}(i, j)} \quad (5)$$

$$I_{\text{scaled2}}(i, j, k) = I_2(i, j, k) \quad (6)$$

If $\gamma_{\text{laser1}}^{\text{laser2}}(i, j) > 1$,

$$I_{\text{scaled1}}(i, j, k) = I_1(i, j, k) \quad (7)$$

$$I_{\text{scaled2}}(i, j, k) = I_2(i, j, k) * \gamma_{\text{laser1}}^{\text{laser2}}(i, j) \quad (8)$$

In order to avoid subtracting signal contributed by the particles, for each pair of frames (or engine cycle k) and at each pixel position (i, j), the minimum value is kept for composing the matrix $I_{\text{min}}(i, j, k)$ for adaptive background calculation. Then for each pixel position (i, j), a linear regression of $I_{\text{min}}(i, j, k)$ with respect to the PIV sampling cycle number k is performed.

$$I_{\text{fit}}(i, j, k) = A(i, j) \cdot k + B(i, j) \quad (9)$$

$A(i, j)$ and $B(i, j)$ are coefficients resulted from the linear regression. As a result, the background $I_{\text{bg}}(i, j, k)$ for each cycle has an offset to the intensity level at the 1st PIV sampling cycle in which no particles are present.

$$I_{\text{bg}}(i, j, k) = I_{\text{fit}}(i, j, k) - [I_{\text{fit}}(i, j, 1) - I_{\text{min}}(i, j, 1)] \quad (10)$$

Finally, the background subtracted frames are calculated as:

$$I_{\text{net1}}(i, j, k) = I_{\text{scaled1}}(i, j, k) - I_{\text{bg}}(i, j, k) \quad (11)$$

$$I_{\text{net2}}(i, j, k) = I_{\text{scaled2}}(i, j, k) - I_{\text{bg}}(i, j, k) \quad (12)$$

Velocity Field Re-Construction Approach

The commercial software DaVis 8.1.4 is used to perform the cross-correlation between each pair of frames yielding in an instantaneous velocity field every 5th engine cycle. The biggest challenge for this experiment is the optical distortion brought by the realistic piston geometry. A previously reported velocity field re-construction approach [25] performs the frame dewarping prior to cross-correlation calculation. This approach has several drawbacks:

1. Since the image transformation is calculated statically for a given crank angle, the same transformation is applied to both frames; however, piston motion creates an artificial radial velocity that is not taken into account.
2. Image registration is not only extremely tedious, but can also create large errors due to subjective identification of the image points when the optical distortion is heavy.
3. As the piston approaches BDC, the exacerbated optical distortion will have little effect on particle image size (dominated by the diffraction-limited spot diameter) which will still cover 2~3 physical pixels. When the distortion correction is performed, each pixels are moved into their new undistorted locations. The image resolution is artificially increased with the created gaps between the dewarped pixels that are filled in by the interpolation scheme, which “stretches” the particle images radially into “lines”. This will induce considerable uncertainty in the radial velocity.

The first two drawbacks have been overcome through the use of an analytical representation of the optical distortion as a function of pixel radial position and relative piston position (the distance between the laser plane and the piston top). This representation was derived using a combination of an analytical ray tracing routine [38] and an empirical perspective ratio calibration (details are provided in [Appendix A](#)). However, the last drawback - the resolution problem - remains the biggest limitation for reliable PIV measurements near BDC. To simplify further discussion, the distortion correction method described in [Appendix A](#) is referred to as “Method A” for the rest of paragraph.

An alternative velocity field re-construction approach - referred to as “Method B” - has been developed to address the resolution problem.

Instead of performing cross-correlation on dewarped background-subtracted frames, this operation is conducted on each pair of undewarped frames (background-subtracted still) with a much smaller interrogation window size ([Table 4](#)). Then, using the calibrated analytical transformation for optical distortion, the interrogation window positions are translated into the undistorted domain, and the radial and tangential velocities are corrected accordingly. The details of this method are presented in [Appendix B](#).