

PIV Measurements in the Swirl-Plane of a Motored Light-Duty Diesel Engine

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ABSTRACT

Particle image velocimetry (PIV) is used to investigate the structure and evolution of the mean velocity field in the swirl (r - θ) plane of a motored, optically accessible diesel engine with a typical production combustion chamber geometry under motoring conditions (no fuel injection). Instantaneous velocities were measured at three swirl-plane heights (3 mm, 10 mm and 18 mm below the firedeck) and three swirl ratios (2.2, 3.5 and 4.5) over a range of crank angles in the compression and expansion strokes. The data allow for a direct analysis of the structures within the ensemble mean flow field, the in-cylinder swirl ratio, and the radial profile of the tangential velocity. At all three swirl ratios, the ensemble mean velocity field contains a single dominant swirl flow structure that is tilted with respect to the cylinder axis. The axis of this structure precesses about the cylinder axis in a manner that is largely insensitive to swirl ratio. Higher swirl ratios reduce the cycle-to-cycle variability in the orientation of this structure. The tangential velocity profiles are well represented by the Bessel function profile often used to initialize the tangential velocity in CFD codes. A value of 2.2 is recommended for the value of the parameter α which controls the shape of the Bessel function profile. During the compression stroke, the measured in-cylinder swirl ratios were similar to those determined from steady-state flow bench measurements. The swirl ratios decreased significantly after TDC, and continued to decline during the expansion stroke.

INTRODUCTION

In diesel engines, mixing between the injected fuel and the available oxygen is a complex process that plays an important role throughout the entire combustion process. Mixing requirements are further amplified in new low temperature

combustion (LTC) strategies, which often use high levels of charge dilution and tailored mixture stratification to control the combustion process. To increase mixing rates and turbulence intensities, modern high-speed direct-injection (HSDI) diesel engines use complex combustion chamber designs and impart a strong tangential velocity component, commonly referred to as swirl, to the in-cylinder flow. Swirl, defined as the organized rotation of the flow around the cylinder axis [1], is used by diesel engines to improve mixing by promoting the large-scale transport of fuel vapor within the combustion chamber and by generating strong, non solid-body-like velocity gradients (dv_{θ}/dr) that feed the production of turbulent kinetic energy [2].

The traditional method of characterizing the strength of the swirl flow in IC engines is with steady-state flow bench measurements. These measurements are obtained by sealing the head of the engine against a cylinder and setting up a constant mass flow rate through the head. The angular momentum, or swirl strength, induced in the flow as it passes through the head is then typically measured with an impulse torque meter or paddle wheel velocimeter placed within the cylinder, a few bore diameters below the head. The main drawback of these types of measurements is that they do not incorporate many of the complex flow processes present in an operating engine, including the motion and geometry of the piston and wave dynamics in the intake manifold. Moreover, only a global measurement of the flow swirl is obtained. Asymmetries in the swirl flow about the cylinder axis or other flow structural features are not captured.

Several previous studies have investigated the behavior of swirling flows in internal combustion (IC) engines with more sophisticated optical diagnostic techniques. Early work, reviewed by Miles [2], employed single point Laser Doppler Velocimetry (LDV) measurements to investigate the flow

structures and turbulence properties in a bowl-in-piston diesel engine with a re-entrant bowl. Typically, the induction process produces a single dominant flow structure in the r - θ plane, a vortex centered near the cylinder axis with the tangential velocity increasing with radial position until the effects of wall friction reduce the velocity near the cylinder bore.

The swirling flow structure is generally not uniform along the axis of the cylinder (z direction) and also changes throughout the engine cycle. Turbulent dissipation and friction from the cylinder walls, head, and piston surfaces cause the swirling flow to decrease in strength as the cycle progresses. However, the swirl flow is also modified by the changing geometry of the combustion chamber. As the piston approaches top-dead-center, a majority of the mass within the cylinder is forced into the bowl volume, reducing its moment of inertia. According to the conservation of angular momentum, this causes the flow to rotate at a higher angular velocity and tangential velocity gradients in the radial direction are vastly increased [1].

More recent studies have employed Particle Image Velocimetry (PIV) and other full-field techniques that capture flow structures in greater detail. These studies [3,4,5,6,7,8,9,10] focused on clarifying the structure of the mean velocity field as well as the smaller scale, turbulent motions. Of particular relevance to the present study, Reuss, *et al.* [3,4] and Nordgren *et al.* [5] used PIV measurements to compare measured in-cylinder swirl ratios with the estimates derived from flow bench testing, to examine the tilt, precessional movement, and cycle-to-cycle variability of the swirl structure, and to evaluate the decay of the swirl ratio as the cycle progressed. Reuss, *et al.* [3] have also examined the impact of the assumed initial conditions on simulated flow fields. However, each of these studies either used a flat optical piston geometry or a modified geometry that allows optical access that produces little-to-no distortion of the light emanating from the combustion chamber. The geometric details of production-like pistons can be expected to introduce asymmetries into the swirl flow, to impact the attenuation of flow swirl, and to alter the geometry and location of the swirl structure.

Although a few additional studies [11,12,13,14] have acquired velocity measurements in a diesel engine with production-like piston geometries, measurements were only acquired in the vertical, r - z plane of the combustion chamber and the structure of the swirl flow was not examined. Colban *et al.* [11] investigated the in-cylinder flow in the same light-duty diesel engine used in this study, which has a production-like optical piston with a re-entrant, bowl-in-piston geometry and piston-top valve cutouts. Colban *et al.* developed a methodology to correct for the optical distortions generated by the curved surfaces of the relatively complex piston geometry and were able to identify the main flow structures

in the r - z plane during both motored and fired operation. The current study expands on this previous work by investigating the velocity field in the r - θ plane of the same optically accessible diesel engine at several vertical positions and swirl ratios. A part of our motivation for doing so is to better understand the spatial distributions of UHC and CO reported by Ekoto *et al.* [15,16,17] in the same engine. The temporal evolution of these distributions suggests the presence of azimuthal asymmetries. Understanding the flow in the r - θ plane, even in the absence of fuel injection, is a first step toward explaining these asymmetries. In addition, comparison of the measurements of Ekoto *et al.* with computational fluid dynamics (CFD) based combustion simulations indicate that the fuel-air mixing processes in the piston bowl are not well captured.

A second motivation for this work is to obtain higher quality data characterizing the initial swirl distribution, and its pre-combustion evolution, for initializing and validating CFD simulations. To allow for efficient exploration of design alternatives, many engine combustion simulations begin sometime after intake valve closing (IVC) in what are referred to as closed-cycle simulations. These simulations require an initialization of the in-cylinder flow field. As an example, the CFD code KIVA [18], uses Bessel functions of the first kind (J_1 and J_2) to approximate the v_θ profile:

$$v_\theta(r,t) = \frac{\omega_c r_b R_s \alpha}{4 \cdot J_2(\alpha)} \cdot J_1\left(\alpha \frac{r}{r_b}\right) \quad (1)$$

In Eq. (1), r_b is the cylinder radius, α is the input parameter that defines the shape of the velocity profile, R_s is the swirl ratio, and ω_c is the angular rotation rate of the crankshaft. This functional form of the v_θ distribution is an exact solution to the transient, one-dimensional Navier-Stokes equations, as shown in Appendix A. Wahiduzzaman and Ferguson [19] performed LDV measurements in a constant volume cylindrical vessel filled using a rapid-acting intake valve and determined that their v_θ profiles were well characterized by the Bessel function in Eq. (1) with a value for α of 3.11. This is the suggested value of α for flow initialization in KIVA [18]. However, a least-squares fit to the data of Johnston, *et al.* [20] - obtained in an idealized side-valve engine - suggest a somewhat smaller value for α of 2.7. Van den Heuvel *et al.* [8] used Doppler Global Velocimetry (DGV) in the cylinder volume of a flow bench setup at several stationary intake valve positions, and PIV measurements in an optical engine of unspecified geometry, to investigate the behavior of the v_θ profiles generated by a modern diesel engine. These authors determined that the swirl flow generated by their head design was not well represented by the Bessel function in Eq. (1). Jointly, these studies indicate a need to better characterize the

tangential velocity distribution, particularly those produced by modern HSDI diesel engine port designs.

This study investigates the mean velocity field in the r - θ swirl plane of an optically accessible diesel engine that retains the geometry of a typical production diesel engine by acquiring PIV measurements at three vertical positions (z -locations) within the combustion chamber. Consistent with the motivation described above, the main goals of this study were to: (1) characterize the swirl structure within the cylinder, including asymmetries that might impact the subsequent combustion process or the most appropriate modeling approach, (2) understand the evolution of this structure with crank angle and how it is affected by changes in swirl ratio, and (3) investigate the form of the tangential velocity (v_θ) profile and determine appropriate values for the swirl input parameter α when a Bessel function based initialization is used. The data are also used to provide further support for the use of flow-bench derived swirl ratios to characterize the total in-cylinder angular momentum.

EXPERIMENTS

ENGINE

The engine used in this study is modified 1.9 L GM automotive diesel engine that provides extensive optical access to the combustion chamber. A schematic of the engine is provided in Fig. 1. Optical access to the side of the combustion chamber is achieved through 50 mm wide by 25 mm tall fused silica windows located at the top of the cylinder liner. The top portion of the Bowditch-style piston is made of fused silica and includes the re-entrant bowl and valve cutout geometries of typical production pistons. The only significant deviation from a production piston design is that the height of the top ring-land was extended to allow imaging within the bowl portion of the piston. The resulting crevice volume is roughly 3-4 times larger than a production diesel engine. Two gapless compression rings minimize blowby. An important feature of the engine is that the intake ports (one helical and one tangential) are fitted with swirl control vanes that allow the in-cylinder swirl ratio to be varied. Additional specifications for the optical engine are given in Table 1. This engine has been used in several previous studies [11,15,16,17,21,22,23,24] and was shown to reproduce the emissions and combustion behavior of a similar all-metal engine [24].

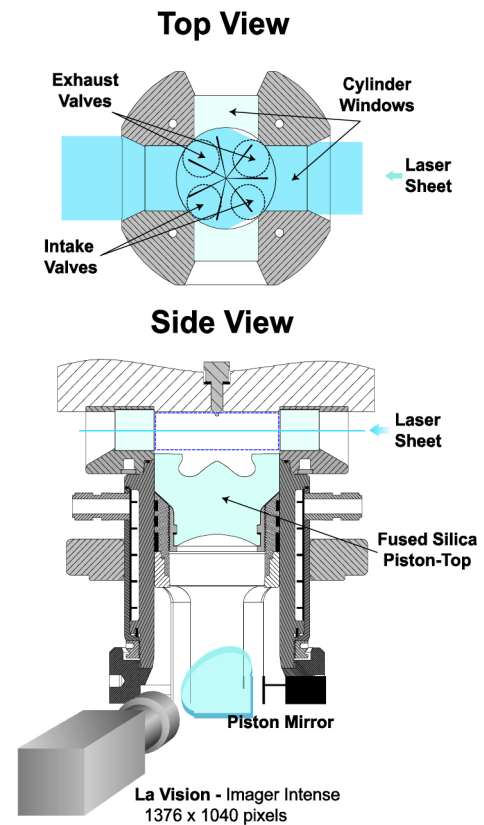


Figure 1. Cross-section of the optical engine and piston-bowl geometry, with schematic representation of the PIV imaging system.

Table 1. Engine specifications and valve timings.

Bore	82.0 mm
Stroke	90.4 mm
Displacement Volume	0.477 L
Geometric CR	16.7
Squish Height	0.78 mm
Intake Valves	2
Intake Valve Open	-359° CA
Inlet Valve Close	-152° CA
Exhaust Valves	2
Exhaust Valve Open	132° CA
Exhaust Valve Close	360° CA

*° CA are referenced to TDC of compression CA ° CA

PARTICLE IMAGE VELOCIMETRY (PIV) SETUP

In the PIV measurements, a Big Sky Nd:YAG laser system excited Mie scattering from small particles seeded into the intake flow. This system has two independent laser heads and a set of beam-combining optics to align the two output beams. The second harmonic output (532 nm) of the lasers was directed to the engine with a series of dichroic laser

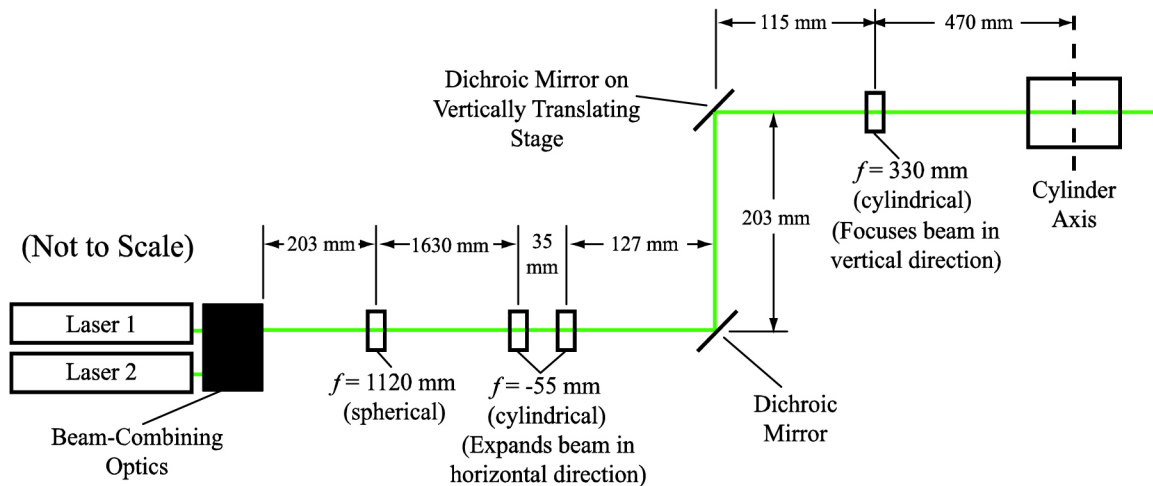


Figure 2. Layout of optical components used in PIV experiments.

mirrors. The final mirror was positioned on a translation stage that provided precise control of the vertical position of the laser sheet within the combustion chamber. The laser beam was expanded in the horizontal direction using two -55 mm focal-length cylindrical lenses positioned 35 mm apart and 915 mm from the center of the cylinder. The curvature of the cylinder windows (Fig. 1) further increases the divergence of the laser sheet in the horizontal direction, allowing a majority of the cylinder to be illuminated by the laser. The laser light was focused vertically in the center of the cylinder with a 1120 mm spherical lens and a 330 mm cylindrical lens spaced apart by 2110 mm, as shown in Fig. 2. The resulting laser-sheet thickness was approximately $700\text{ }\mu\text{m}$ at the center of the cylinder. The use of a two-lens system was required to position the laser sheet near the firedeck (bottom surface of the engine head) to facilitate measurements near top-dead-center (TDC). Reflections of the incident laser light that saturated the camera's CCD array were unavoidable at several of the measurements locations. Therefore, PIV measurements were not collected over a range of crank angles near TDC for each measurement height. Laser energies were limited to a maximum of approximately 30 mJ/pulse in all of the experiments to limit the intensity of the reflected laser light.

Particles manufactured by 3M (iM30K Glass Bubbles) were seeded into the intake stream using a modified TSI Model 3400 fluidized-bed seeder. The particles are made of borosilicate glass with a density (ρ_p) of 600 kg/m^3 and ranged in diameter (d_p) from 9 to $25\text{ }\mu\text{m}$. Assuming Stokes drag, the response time ($\tau_s = d_p^2 \rho_p / (18\mu)$) of these particles is $\sim 280\text{ }\mu\text{s}$ (3.5° CA at 1500 RPM) for a particle with a diameter of $18\text{ }\mu\text{m}$. This time scale is longer than estimates of the Kolmogorov time scale, but an order of magnitude smaller than the mean-flow and large-scale turbulent time scales of greatest relevance, derived from previous measurements in a light-duty diesel engine operating at a similar condition [2]. Therefore, the measurements acquired in this study cannot be

used to examine the turbulence properties of the in-cylinder flow and will only be applied to assess the mean-flow structures. The seeder was supplied with a 0.5 g/s flow of nitrogen to fluidize the bed of seed particles and bronze beads ($d = 50\text{ }\mu\text{m}$). Collisions with the bronze beads help prevent the seed particles from agglomerating, but cause the seed particles to become electrostatically charged and adhere to the fused silica piston and windows. Water vapor, added to the intake stream in the proportion typically found in EGR gases, helps to dissipate the static charge and significantly reduces window fouling.

Images of Mie scattered light from the particles were acquired with a LaVision Imager Intense CCD camera (1376×1040 pixels) equipped with a Tamron 28-135 mm zoom lens. Ambient light was filtered by a 532 nm narrow band-pass filter ($\text{FWHM} = 3\text{ nm}$). A magnification of ~ 0.05 was used for the images acquired in this study; however, the magnification varies slightly as the piston moves through the engine cycle (due to the lensing effect of the contoured piston window). This magnification results in an image resolution of $\sim 130\text{ }\mu\text{m/pixel}$. Two images, separated temporally by $15\text{ }\mu\text{s}$, were acquired to determine individual velocity fields. The exposure duration of the first image was $10\text{ }\mu\text{s}$. The electronic exposure duration of the second image is constrained to 90 ms by the readout duration of the first image. To minimize the influence of stray light on the second image, the exposure was limited to 3 ms using a mechanical shutter.

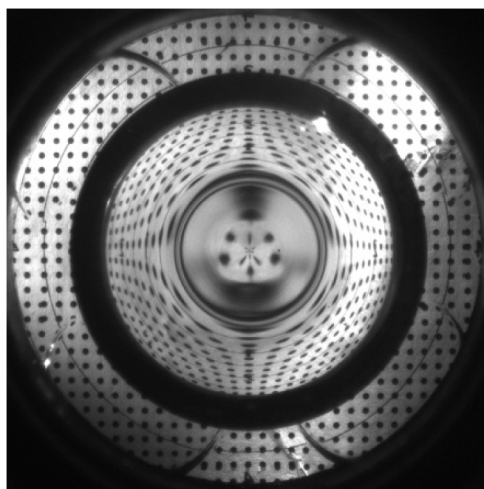
DATA PROCESSING

Image Processing

The background signal was first subtracted from the data images by calculating a mean background image. Initially, two different methods of determining the background signal were examined: (1) the mean of 100 images acquired with the laser in operation, but without seeding the in-cylinder flow

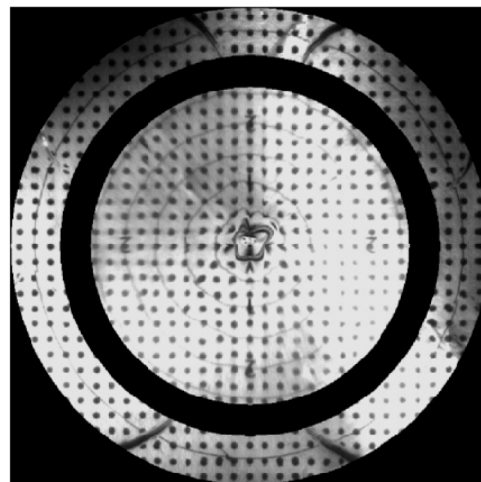
(2) the mean of the PIV data images acquired with the laser in operation and the in-cylinder flow seeded. It was determined that the mean of the PIV images is more effective at eliminating the background signal. This is attributed to the fact that when seed particles are introduced into the flow, laser light is scattered in all directions, illuminating the surfaces of the combustion chamber and increasing the background level. However, since the seed particles are never observed in the same exact location, the individual particle signals are diminished in the averaging process, while the background signal survives. Because the overall scattered light levels are higher, the mean PIV data image provides a more appropriate representation of the true background signal.

The optical distortions generated by the curved surfaces of the fused silica piston were corrected by applying piece-wise linear transformations to the distorted images. The transformations were derived from images of a calibration target obtained at each crank angle and measurement height at which velocity measurements were acquired. Examples of a distorted calibration target image, and a corrected target image, are shown in [Fig. 3](#).



(a)

Figure 3. (a) - Image of calibration target before distortion correction and (b) after distortion correction. The piston is located at 50°bTDC. Spacing between points on the target is 2.5 mm. A portion of the bowl region was blackened (black ring separating the bowl and squish regions) to eliminate regions where the acquired signal saturated the camera.



(b)

Figure 3 (cont.). (a) - Image of calibration target before distortion correction and (b) after distortion correction. The piston is located at 50°bTDC. Spacing between points on the target is 2.5 mm. A portion of the bowl region was blackened (black ring separating the bowl and squish regions) to eliminate regions where the acquired signal saturated the camera.

The distortion-correction transformation was determined by mapping the grid points in a distorted calibration target image to the known position of the points on the target using a data processing code independently developed by the authors. Near the bowl wall, the piston surface has a high level of curvature. This highly curved region causes two main problems: (1) light incident on this region causes the camera to saturate at a majority of the measurement locations and (2) the optical distortion is greatest in this region and is much more difficult to properly correct. Consequently, this region was artificially blackened in the data images to eliminate regions where the acquired signal was saturated. Distinct piecewise-linear transformations were generated for the bowl region (inside the blackened ring) and for the squish region (outside the blackened ring). The transformations were then used to correct all of the data images at the crank angle of interest.

[Figure 3](#) demonstrates that the distortion correction works well in the squish portion of the combustion chamber, but is less effective near the center of the cylinder. Errors generated by the imperfect distortion correction process are difficult to fully quantify, but are estimated to be less than approximately 0.5 mm in absolute position. Error in the local image scale, which is used to convert particle displacement to velocity, is estimated to be small at all but the central locations of the cylinder, where the velocities are small. It is also important to note that only one distortion transformation was determined for each pair of PIV images. The 15 μ s delay between images

causes the second image to be acquired at a slightly different crank angle than the location of the calibration image. This delay potentially introduces additional error in the magnitude of the computed velocity. These shortcomings of the distortion correction process are not expected to significantly influence the conclusions made in this study.

Velocity Field Processing

The particle images were preprocessed by normalizing the intensity of the particles with a min/max filter. This filter determines the minimum and maximum intensity in a 5 pixel square window, calculates a median intensity, and normalizes the original intensity to this value. The velocity vectors were computed by cross-correlation using an iterative multi-pass algorithm within the DaVis FlowMaster version 7.1.1.35 software package. The interrogation window size was reduced from 64×64 pixels in the first pass to 32×32 pixels with 50% overlap in the second pass, resulting in a spatial resolution for the velocity measurements of 2 mm.

The in-cylinder swirl ratio was calculated for each instantaneous velocity field following the procedure used in [5], which is briefly summarized here. To determine the swirl ratio, an effective rotation rate must be assigned to the in-cylinder flow. This effective rotation rate corresponds to the angular velocity of a flow in solid-body rotation that has the same total angular momentum as the measured velocity field. The total angular momentum in the measured swirl plane was estimated as the sum of the angular momentum of each velocity vector as shown in Eq. (2).

$$M_{v1} = \sum_{i=1}^n r_i v_{\theta,i} \quad (2)$$

The angular momentum of a solid body rotating with an angular velocity, ω , is given in Eq. (3)

$$M_{v2} = \sum_{i=1}^n r_i \cdot (r_i \cdot \omega) = \omega \sum_{i=1}^n r_i^2 \quad (3)$$

Setting the total angular momentum in Eqs. (2) and (3) equal, the rotation rate of the swirl flow can be determined by Eq. (4). Finally, the swirl ratio, R_s , of each instantaneous velocity field was calculated using Eq. (5). The swirl ratio is the ratio of the angular velocity ω to the angular velocity of the engine ($2\pi N$), where N is the rotational speed of the engine.

$$\omega = \frac{M_{v1}}{\sum_{i=1}^n r_i^2} \quad (4)$$

$$R_s = \frac{\omega}{2\pi N} \quad (5)$$

Spurious vectors, mainly generated from reflections of the incident laser light in regions of high bowl curvature, act to skew the in-cylinder swirl ratios towards smaller values. To remove these erroneous vectors, several discrimination criteria were established. First, a maximum velocity threshold of 40 m/s was placed on the calculated velocity field. This velocity threshold was determined by approximating the in-cylinder flow as a solid body rotation with a swirl ratio of ~ 6 . The maximum swirl ratio used in this study is 4.5 and, therefore, a swirl ratio of 6 provides an estimate of the maximum velocity. At a radial position of 41 mm (at the bore wall), the flow velocity for a solid body rotation corresponding to a swirl ratio of 6 is ~ 40 m/s. Secondly, vectors were eliminated if their velocity deviated from the ensemble mean value by greater than twice the mean piston speed (fluctuating velocity $\sim$ half the mean piston speed in most engine flows). The erroneous vectors were reduced in number, but not completely eliminated by this conditional sampling process. However, the number of remaining erroneous vectors is small when compared to the number of high quality vectors. The presence of the remaining erroneous vectors does not have a significant effect on the calculated swirl ratio.

Additional data processing was incorporated in the analysis of the tangential velocity profiles in an effort to further reduce the number of erroneous velocity vectors. The mean tangential velocity and root-mean-square velocity were initially calculated as a function of radial position for each ensemble mean-velocity field. Vectors that deviated from the mean velocity by more than twice the root-mean-square velocity were removed. Furthermore, a significant number of erroneous vectors near the cylinder wall were discarded. The elimination of these data points allow for a better estimate of the tangential velocity profile and a more accurate calculation of the parameters in Eq. (1) that provide the best fit to this profile.

OPERATING CONDITIONS

The main specifications of the single, motored operating condition used in this study are provided in Table 2. As mentioned previously, a major motivation for this study is to examine flow-related factors that can cause asymmetries in measured UHC and CO distributions, and that may impact our ability to accurately model the fuel-air mixing processes. The low speed (1500 RPM), slightly boosted ($P_{in} = 1.5$ bar), highly dilute (10% O₂ mole fraction) condition selected represents a low-load, partially premixed compression ignition (PPCI) operating condition investigated extensively in previous studies [15,16,17, 21,22,23,24]. Under fired operation, it provides an adequate combustion efficiency

